Isı Bilimi ve Tekniği Dergisi, 32, 2, 121-131, 2012
J. of Thermal Science and Technology
©2012 TIBTD Printed in Turkey
ISSN 1300-3615

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

Nihal UĞURLUBİLEK
Department of Mechanical Engineering, Faculty of Engineering, Eskisehir Osmangazi University,
26480 Eskisehir, Turkey, e-mail: nihalu@ogu.edu.tr

(Geliş Tarihi: 18. 01. 2011, Kabul Tarihi: 17. 04. 2011)

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^o and 0.2 m, respectively. The Navier- Stokes and energy equations have been solved using Simple algorithm and Standart k-ε 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 occured 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 ekseni 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^o 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 $\left[=\dfrac{d_h q_w}{k(T_w - T_b)}\right]$
P	turbulent kinetic energy product
ΔP	pressure drop [N/m2]
Pr	Prandtl number$\left[=\mu C_p / k\right]$
q	heat flux [*W/m*^{2}]
Re	Reynolds number $\left[=\dfrac{d_h \rho \bar{V}}{\mu}\right]$
R_3	the performance criterion
T	temperature [*K*]
u,v,w	velocities in *x*, *y*, and *z* directions [*m/s*]4
$\vec{V}$	velocity vector

Greek letters

η	thermal performance factor
ε	turbulent dissipation rate
ν_t	turbulent viscosity
ρ	Density [*kg/m*^{3}]
$\sigma_\varepsilon, \sigma_k, C_1, C_2$	model coefficients
μ	dinamic viscosity [*kg/ms*]
$\vec{\tau}$	stress tensor

Subscripts

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 develope 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 acustic 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 algoritm 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:

$$\nabla . \vec{V} = 0 \tag{1}$$

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

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

Energy equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\left(1+\frac{\alpha_t}{\alpha}\right)\nabla^2 T \tag{3}$$

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

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

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

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

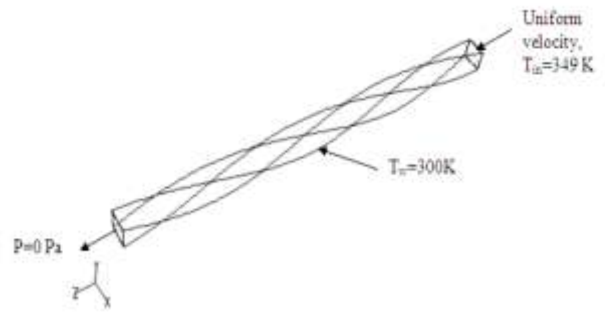


Figure 1. The geometry and boundary conditions.

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=349K.

ρ (kg/m^3)	μ (kg/ms)	C_p (J/kgK)	k (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

$$Re=\frac{d_h\rho\bar{V}}{\mu} \tag{6}$$

The friction factor

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

and Nusselt number

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

where the fluid bulk temperature, T_b, has been calculated

$$T_b=\frac{1}{\bar{V}A_c}\iint_A \bar{V}T\,dA_c \tag{9}$$

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

$$q_w=\frac{1}{A_w}\iint_A q\,dA_w \tag{10}$$

and T_w is wall temperarure with area averaging.

$$T_w=\frac{1}{A_w}\iint_A T\,dA_w \tag{11}$$

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).

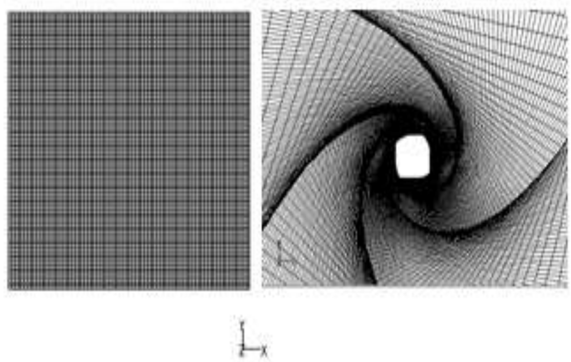


Figure 2. The grid structure of the inlet and the channel wall (view from the inside).

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.

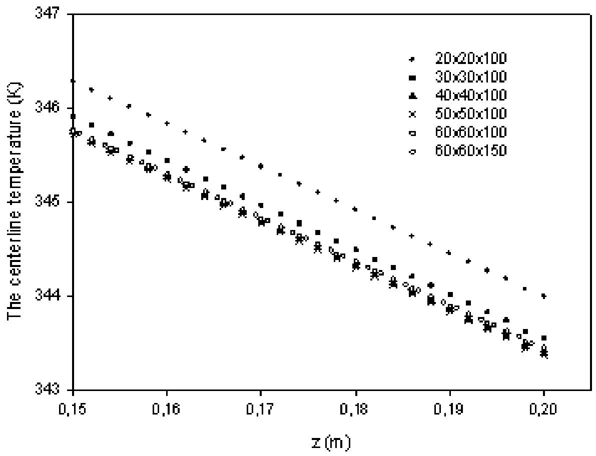


Figure 3. Grid independency study.

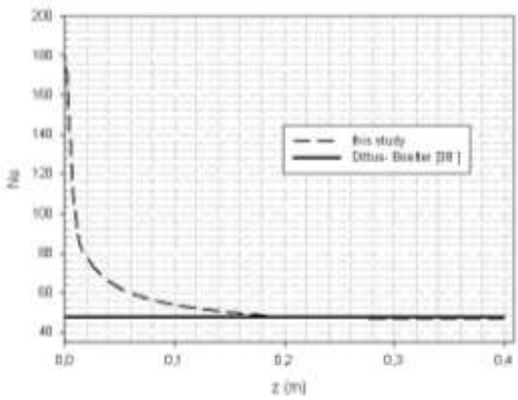
For this study, the length to width ratio of the smooth square duct (*L/H*) is considered as 40. The Reynolds number is accepted as 10000.

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 approache 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).

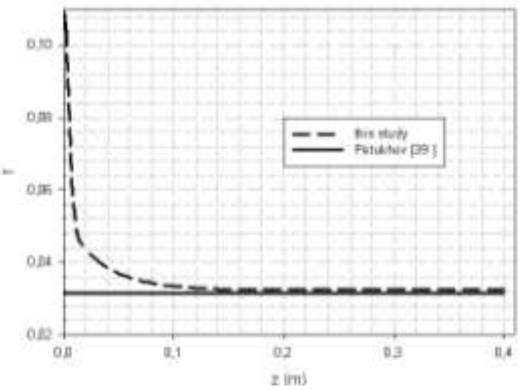
RESULTS AND DISCUSSION

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^{o} 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

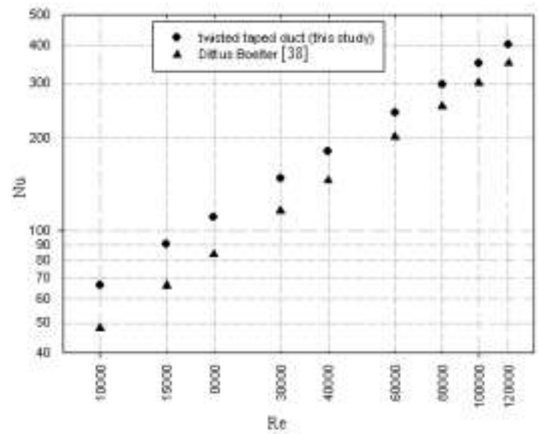


(a)

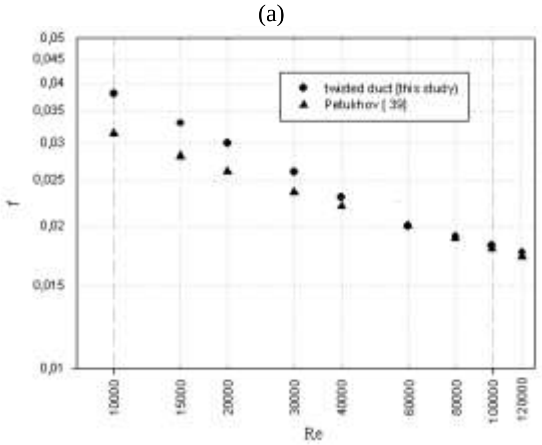


(b)

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 temperarture 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 showed 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.

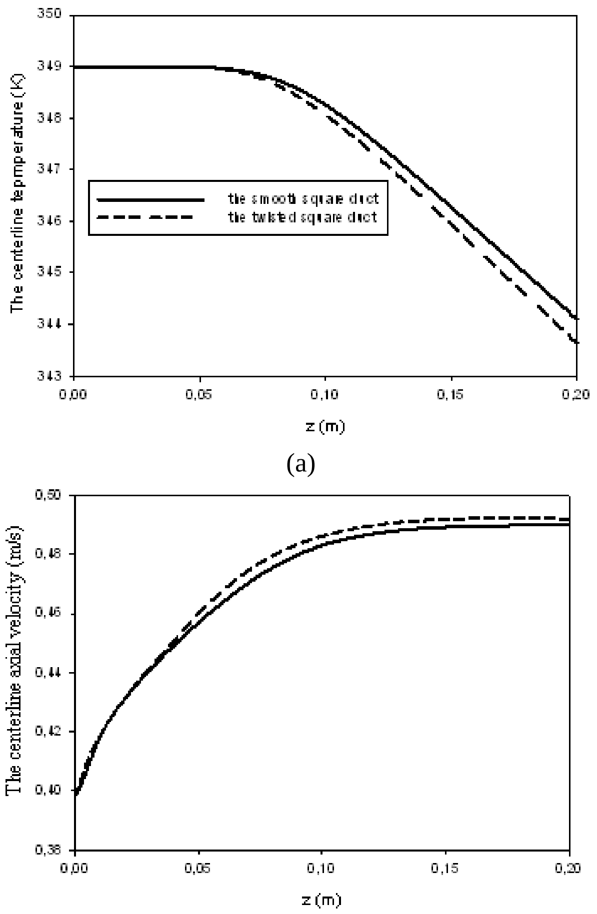


Figure 6. The centerline temperature (a) and axial velocity (b) plots for Re=10000.

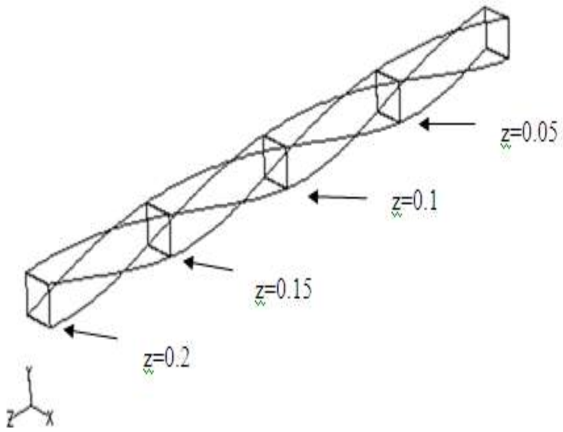


Figure 7. The locations of the geometry along the z direction.

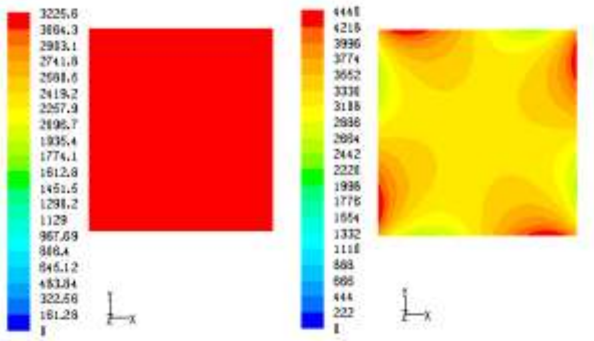


Figure 8. Pressure at the inlet for two cases for Re=100000.

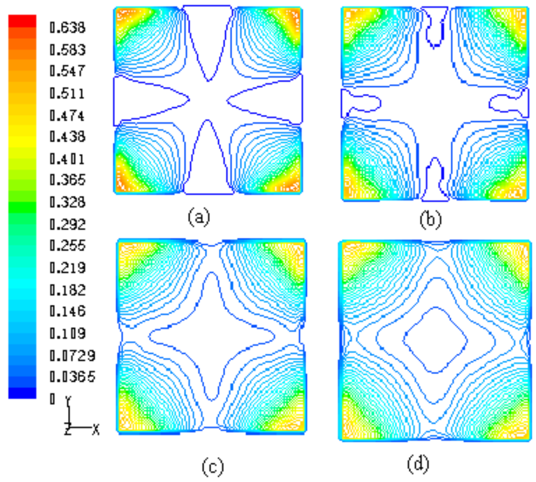


Figure 9. Tangential velocity contours (m/s) at z=0.05m(a), z=0.1 m(b), z=0.15m(c) and z=0.2m(d) for Re=100000.

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 occured 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 occures 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,

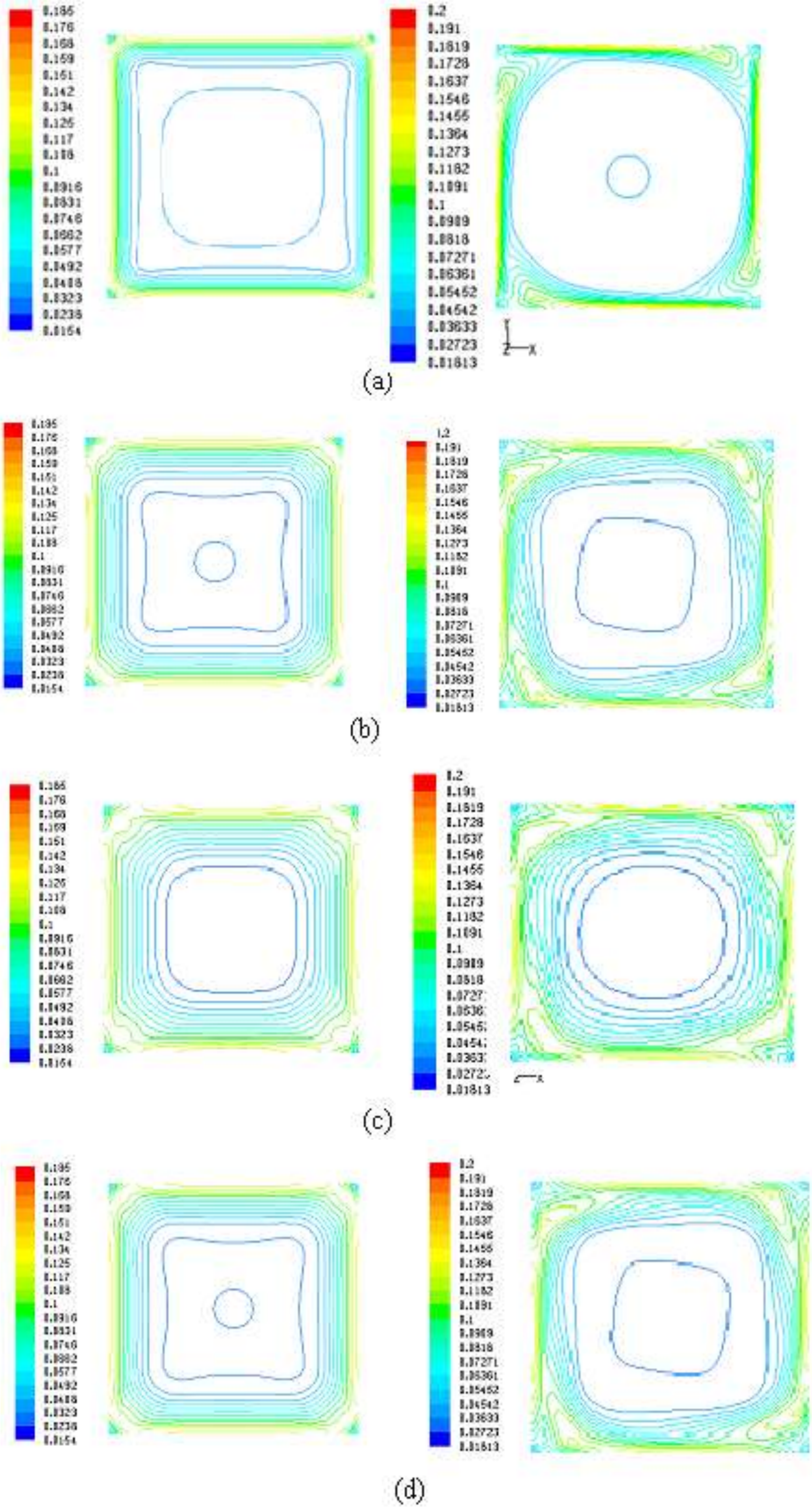


Figure 10. Turbulence kinetic energy contours of the smooth and twisted square ducts at (a) z=0.05, (b), (b) z=0.1, (c) z=0.15, (d) z=0.2 m (*Re*=100000).

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)

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

Therefore, performance evaluation factor, η, can be written as (Promwonge,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 (η) 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.

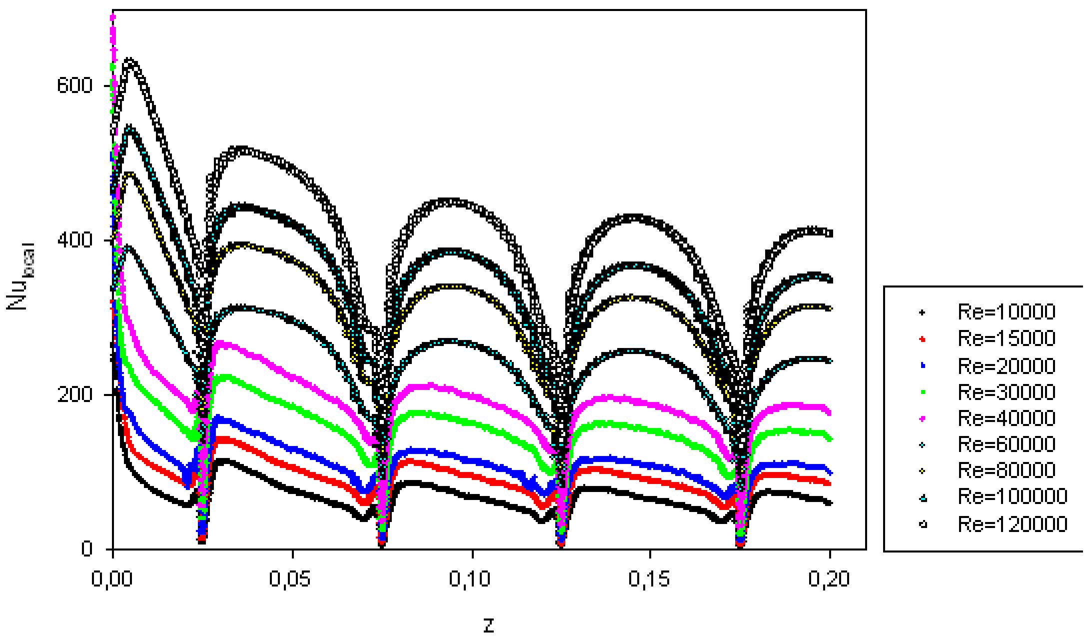


Figure 11. The local Nusselt number of the twsited square duct.

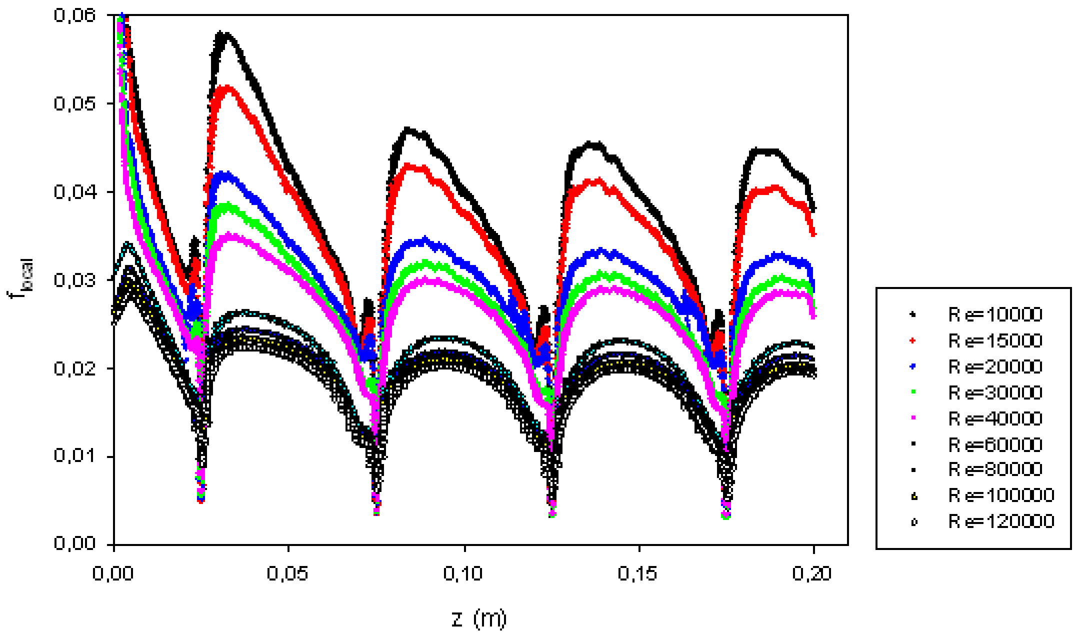


Figure 12. The local friction factor of the twsited square duct.

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 (μ) of the working fluid used at the compared studies is bigger than those this study. Because of the larger viscosity, the secondary flow occures smaller than those this work. It can be observed from this result that the secondary flow occures less intensive with increasing Prandtl number. The reason of this case is that the incerasing 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-ε 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^{o} 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.

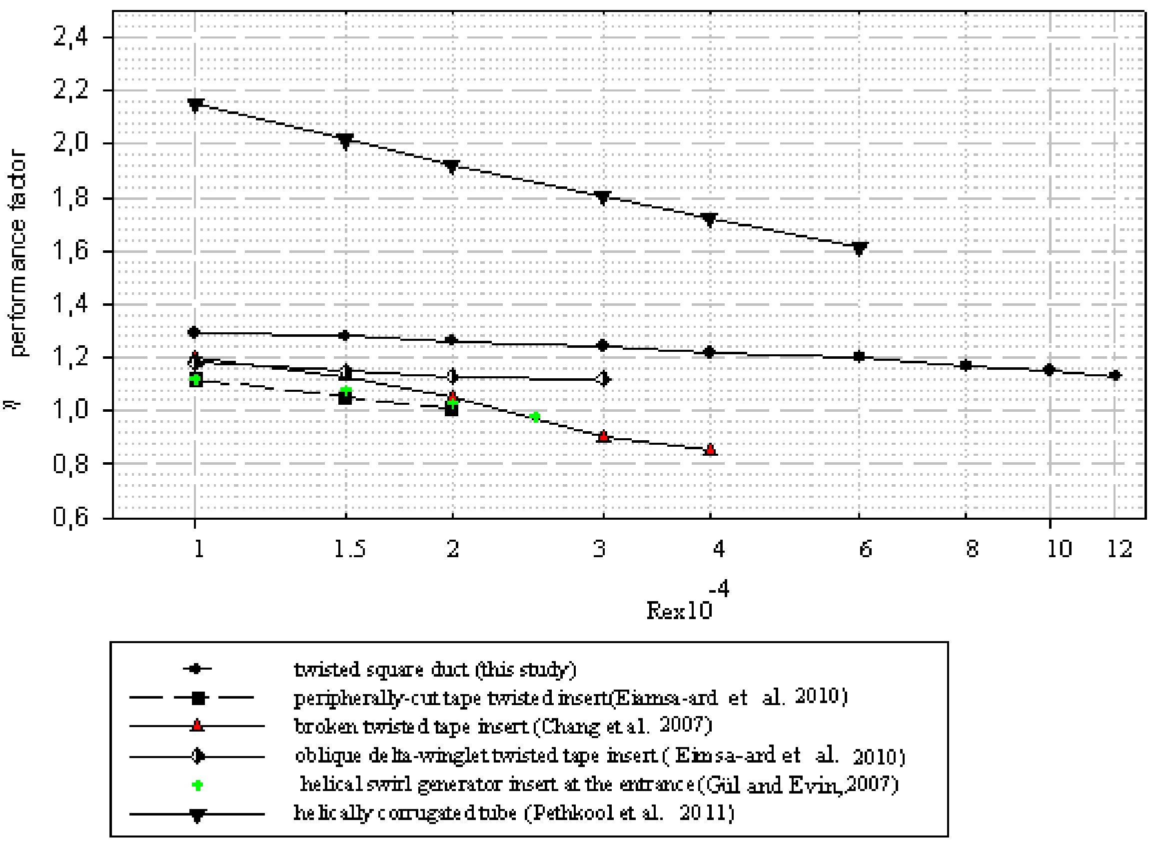
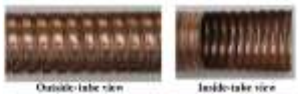


Figure 13. Comparison of the performance factors

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 ^oC)		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 ^oC)		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 ≤*Re*≤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 ^oC) 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.

ACKNOWLEDGMENTS

The author is indebted to anonymous reviewers for their extremely careful and constructive reviews, which significantly improved the quality of the paper.

REFERENCES

Asmantas, L.A., Nemira, M.A., Trilikauskas, V.V., Coefficients of heat transfer and hydraulic drag of a twisted oval tube, *Heat Transfer-Soviet Research*, 17 4, 103 -109, 1985.

Berger, S.A., Talbot, L., Yao, L.S., Flow in a curved pipe, *Annu. Rev. Fluid Mech*., 15. 410–512, 1983.

Berger, S.A., Flow and heat transfer in curved pipes and tbes, *AIAA Paper*, 91-0030, 1991.

Bergles, A.E., Blumenkrantz, A.R., Taborek, J., Performance evaluation criteria for enhanced heat transfer surfaces, *J. Heat Transf.*, 2, 239–243, 1974.

Bergles, A.E., Enhancement of heat transfer, 6th *Int. Heat Transfer Conf.*, Toronto, 89–108, 1978.

Bishara, F., Jog, M.A., Manglik, R.M., Computational simulation of swirl enhanced fl ow and heat transfer in a twisted oval tube, *Journal of Heat Transfer*, 1318, 1 page, 2009.

Bolinder, C.J., The effects of torsion on the bifurcation structure of laminar flow in a helical square duct, *Trans. ASME J. Fluid Engrg*., 117, 242–248, 1995.

Chang, S. W., Yang, T. L., Liou, J. S., Heat transfer and pressure drop in tube with broken twisted tape insert, *Experimental Thermal and Fluid Science*, 32, 489–501, 2007.

Chen, Y., Chen, H., Zhang, B., Hsieh, H. T., Fluid flow and convective heat transfer in a rotating helical square duct, *International Journal of Thermal Sciences,* 45 10, 1008–1020, 2006.

Chen, W.H., Jan, R., The torsion effect on fully developed laminar flow in helical square channels, *Trans. ASME J. Fluids Engrg*., 98. 41–48, 1993.

Chompookham,T., Thianpong, C., Kwankaomeng, S.,

Promvonge, P., Heat transfer augmentation in a wedge-ribbed channel using winglet vortex generators, *International Communications in Heat and Mass Transfer,* 372 163–169, 2010.

Dean, W.R., The streamline motion of fluid in a curved pipe, *Philosophy Magazine*, 5 673–695, 1928.

Dittus, F. W., Boelter, L. M. K., Universty of California, Berkeley, *Publication on Engineering*, 2, 443, 1930.

Dzyubenko, B.V., Yakimenko, R.I., Efficiency of heat transfer surfaces using the method of effective parameters, *Heat Transfer Research* 32 7, 8, 447- 454, 2001.

Eiamsa-ard S., Seemawute P., Wongcharee K., Influences of peripherally-cut twisted tape insert on heat transfer and thermal performance characteristics in laminar and turbulent tube flows, *Experimental Thermal and Fluid Science*, 34, 711–719, 2010.

Eiamsa-ard, S., Wongcharee, K., Eiamsa-ard , P., Thianpong , C., Heat transfer enhancement in a tube using delta-winglet twisted tape inserts, *Applied Thermal Engineering,* 30, 310–318, 2010.

Fluent 6.1.22 Users Guide, Fluent Incorporated. Centerra Resource Park, Lebanon, USA, 2001.

Gao, X.N., Zou, H.C., Wang, D.Y., Lu, Y.S., Heat transfer and flow resistance properties in twisted tube oblate with large twist ratio, Huanan Logong Daxue/*Journal of South China University of Technology Natural Science*, 36 11, 17–26, 2008.

Garcia, A., Vicente, P.G.,Viedma,A., Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers, *Int. J. Heat Mass Transf.,* 48. 4640–4651, 2005.

Gül, H. , Evin, D., Heat transfer enhancement in circular tubes using helical swirl generator insert at the entrance, *International Journal of Thermal Sciences*, 46, 1297–1303, 2007.

Gunes, S., Ozceyhan V., Buyukalaca, O., Heat transfer enhancement in a tube with equilateral triangle cross sectioned coiled wire inserts, *Experimental Thermal and Fluid Science*, 34 6, 684-691, 2010.

Hwang, J.J., Lai, D.Y., Three-dimensional laminar flow in a rotating multiple pass square channel with sharp 180-deg turns, *J. Fluids Engineering*, 120 488–495, 1998.

Ito, H., Flow in curved pipe, *Japan Soc. Mech. Engrg. Int. J.*, 30, 543–552, 1987.

Kahraman, N., Sekmen, U. Çeper, B., Akansu S. O., Numerical investigation of effect of turbulator on heat transfer in pipe flows, *J. of Thermal Science and Technology*, 28 2, 51-59, 2008.

Kotorynski, W.P., Steady laminar flow through a twisted pipe of elliptical cross-section, *Computers &Fluids*, 144, 433-444,1986.

Kucuk, H., Asan H., Forced convection heat transfer in eccentric curved annular square ducts, *J. of Thermal Science and Technology*, 29 1, 67-78, 2009.

Launder, B. E. and Spalding, D. B.. *Lectures in Mathematical Models of Turbulence*. Academic Press. London. England. 1972.

Liebenberg, L., Meyer, J.P., In-tube passive heat transfer enhancement in the process industry, *Appl. Therm. Eng*., 27 16 2713–2726, 2007.

Liu, S., Masliyah, J.H., Axially invariant laminar flow in helical pipes with a finite pitch, *J. Fluid Mech*. 251, 315–353, 1993.

Nandakumar, K., Masliyah, J.H., Swirling flow and heat transfer in coiled and twisted pipe, *Adv. Transport Processes*, 4, 49–112, 1986.

Nobari, M.R.H. , Nousha, A., Damangir, E., A numerical investigation of flow and heat transfer in rotating U-shaped square ducts, *International Journal of Thermal Sciences*, 48, 590–601, 2009.

Norouzi, M., Kayhani, M.H., Shu, C., Nobari, M.R.H., Flow of second-order fluid in a curved duct with square cross-section, *J. Non-Newtonian Fluid Mech.*, 165, 323–339, 2010.

Papa, F., Vaidyanathan, K., Keith, T.J., Dewitt, K.J., Numerical computations of flow in rotating ducts with strong curvature, *Int. J. Numerical Methods for Heat & Fluid Flow*, 10 5 541–557, 2000.

Papa, F., Keith Jr, T.G., Dewitt, K. J., Vaidyanathan, K., Numerical calculation of developing laminar flow in rotating ducts with a 180-deg bend, *Int. J. Numerical Methods for Heat & Fluid Flow*, 12 7 780–799, 2002.

Patankar, S.V., *Numerical Heat Transfer and Fluid Flow*. 79-138. Hemisphere, McGraw-Hill,Washington DC., 1980.

Pethkool, S., Eiamsa-ard , S., Kwankaomeng , S., Promvonge, P., Turbulent heat transfer enhancement in a heat exchanger using helically corrugated tube, *International Communications in Heat and Mass Transfer*, 38, 340–347, 2011.

Petukhov, B. S., *Advances Heat Transfer*, 6, 1970.

Pozrikidis, C., Stokes flow through a twisted tube, *J. Fluid Mech.*, Cambridge University Press, 567, 261–280, 2006.

Promvonge, P., Thianpong, C., Thermal performance assessment of turbulent channel flow over different shaped ribs, *International Communications in Heat and Mass Transfer*, 35 10, 1327–1334, 2008.

Qin, Z., Pletcher, R. H., Large eddy simulation of turbulent heat transfer in a rotating square duct, *International Journal of Heat and Fluid Flow*, 27, 371–390, 2006.

Scott W. J., Oran M. T., Hassan A., Chaotic advection by laminar flow in a twisted pipe, *Journal of Fluid Mechanics*, 209, 335-357,1989.

Seigel, L.G., The effect of turbulence promoters on heat transfer coefficients for water flowing in horizontal tubes, *ASHVE J*. 111–114, 1946.

Shah R.K., London A.L., Laminar flow forced convection in ducts, in: *Advances in Heat Transfer*, Academic Press, New York, 1978.

Shome, B., Numerical investigation of flow and heat transfer in twisted circular-sector ducts, *Numerical Heat Transfer; Part A: Applications*, 45 10, 977–995, 2004.

Si, Q., Xia, Q., Liang, L.H., Li, D.X.,Investigation of heat transfer and flow resistance on twisted tube heat exchanger, *Journal of Chemical Industry and Engineering* China, 46 5, 601- 608, 1995.

Sripattanapipat, S., Promvonge, P., Numerical analysis of laminar heat transfer in a channel with diamond-shaped baffles, *International Communications in Heat and Mass Transfer*, 36 1 32–38, 2009.

Thomson, D.L., Bayazitoglu, Y., Meade, A.J., Low Dean number convective heat transfer in helical ducts of rectangular cross section, *Trans. ASME J. Heat Transfer*, 120. 85–91, 1998.

Tuttle, E.R., Laminar flow in twisted pipes, *J. Fluid Mech*. 219, 545– 570, 1990.

Wang, L.B., Tao, W.Q., Wang, Q.W., He, Y.L., Experimental and numerical study of turbulent heat transfer in twisted square ducts, *Journal of Heat Transfer-Transactıons of the Asme*, Volume. 123 5, 868-877, 2001.

Webb,R.L., *Principles of Enhanced Heat Transfer*, John Wiley and Sons, NY, 1994.

Yang, W. M., Analysis of fluid flow and heat transfer in twisted circular-sector ducts, *Numer. Heat Transfer*, 10, 45–61, 1986.

Yang, L., ,Li, Zh. X., Numerical analysis of laminar flow and heat transfer in twisted elliptic tubes, *Engineering Mechanics*, 20 5, 144- 148, 2003.

Zabielski, L., Mestel, A.J., Steady flow in a helically symmetric pipe, *J. Fluid Mech*., 370, 297–320, 1998.

Zachár, A., Analysis of coiled-tube heat exchangers to improve heat transfer rate with spirally corrugated wall, *International Journal of Heat and Mass Transfer*, 53, 3928–3939, 2010.

Zhang, X.X., Wei, G.H.,. Sang, Zh. F, Experimental research of heat transfer and flow friction properties in twisted tube heat exchanger, *Chemical Engineering*, 35 2, 17 -20, 2007.

Nihal UGURLUBILEK was born in Bandırma-Turkey in 1971. She graduated from the department of Mechanical Engineering at Anadolu University in 1992. She pursued her MSc (1997) and PhD (2007) studies at Eskişehir Osmangazi University. She had been assigned assistant of Mechanical Engineering in 1994 and since then she has been working at the same institution. Her main research interests are convective heat transfer, fluid mechanics, heat transfer enhancement techniques, CFD, boiling and condence. She is a member of MMO.