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Laminar Flow Heat Transfer Studies in a Twisted Square Duct for Constant Wall Temperature

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***Abstract:** Three dimensional analysis of steady fully developed laminar flow inside twisted duct of square cross section flow area is studied for Reynolds number range of 100 – 2000 using a commercially available software. Twist ratio used are 2.5, 5, 10 and 20. Heat transfer results are generated for a uniform wall temperature case for Prandtl number range of 0.7 to 20. Maximum values for the product of friction factor and Reynolds number is observed for a twist ratio of 2.5 and Reynold number of 2000. Maximum Nusselt number is observed for the same values along with Prandtl number of 20. Correlations for friction factor and Nusselt number are developed involving swirl parameter. Local distribution of friction factor ratio and Nusselt number across a cross-section is presented. Heat transfer enhancement for Reynold number of 2000 and Prandtl number of 0.7 for twist ratio of 2.5, 5, 10, and 20 are 33 %, 18 %, 10 % and 7 % respectively. The results are significant because it will contribute to development of compact heat exchanger.*
***Keywords:** Twisted duct, laminar flow, twist ratio, Periodic flow, Heat transfer Enhancement*

NOMENCLATURE

A	Heat transfer area, m^2
C	Specific heat of fluid, J/kg-K
D	Side of Square, m
d	Hydraulic diameter of duct, m
E	Enhancement factor for twisted tube
f	Friction factor $= \dfrac{\tau}{\left(\dfrac{\rho V_m^2}{2}\right)}$
$\bar{f}$	Average friction factor for cross section $= \dfrac{\tau_m}{\left(\dfrac{\rho V_m^2}{2}\right)}$
H	Twist ratio, dimensionless $= \dfrac{S}{d}$
h	Heat transfer coefficient, W/m^2- K
k	Thermal conductivity of fluid, W/m-K
L	Periodic length, m

Nu	Nusselt Number, dimensionless = $\dfrac{q\ \ d}{(T_w - T_b)\,k}$
$\overline{Nu}$	Average Nusselt Number for cross section, dimensionless = $\dfrac{q_m\ d}{(T_w - T_b)\,k}$
Pr	Prandtl number , dimensionless = $\dfrac{\mu C}{k}$
q	Wall heat flux , W/m^2
Re	Reynolds number = $\dfrac{\rho V_m d}{\mu}$
S	Pitch of the twisted tube, m
Sw	Swirl parameter, dimensionless = $\dfrac{Re}{\sqrt{H}} A_r$
T	Temperature, K
V	Flow velocity (Twisted Tube), m/s

Greek Letters

μ	Dynamic viscosity , N-s / m^2
ρ	Density of fluid , kg / m^3
τ	Wall shear stress, N/m^2

Subscript

a	Area averaged
b	Bulk temperature , fluid
i	Inlet
m	Mean
r	Area ratio
st	Straight duct
T	Constant wall temperature boundary condition
t	Thermal
tw	Twisted tube
w	Wall

Abbreviations

CFD	Computational fluid dynamics

I. INTRODUCTION

Heat transfer enhancement techniques can be classified either as passive, which require no direct application of external power, or as active, which require external power. The effectiveness of both types of techniques is strongly dependent on the mode of heat transfer. This may range from single-phase free convection to dispersed-flow film boiling. Twisted duct works on the principle of passive enhancement technique. One of the ways of heat transfer enhancement is to create swirl in the flow.

In ducts, heat transfer augmentation by means of secondary flow is obtained either by means of an insert (e.g. twisted tape) in straight duct or by using a twisted duct. Twisted duct used in heat exchanger not only enhances heat transfer inside the ducts but also outside in duct to duct space. Thus, the mixing of the fluid is significantly improved. Swirl induced flow inside the duct is expected to enhance the heat transfer coefficient by an amount similar to that of twisted tape or turbulator inserts. Todd (1977) investigated a general problem of twisted tube.

He simplified the Navier-Stokes equations under large twist ratio assumption in the rotating coordinates which were solved by a regular perturbation method. Author defined the large twist ratio as ratio when change in twist angle (θ) of cross section of duct with respect to direction of flow (z) is very small ($d\theta/dz \ll 1$). He derived a fourth order partial differential equation for the stream function and showed that it is identical to the equation for the small transverse displacement of a clamped elastic plate under a constant loading. Although, this analysis is applicable for a pipe of any cross section with proper boundary conditions, its validity is limited only to a large twist ratio elliptic pipe.

Masliyah and Nandakumar (1981a and 1981b) numerically studied the fully developed steady laminar flow through twisted square ducts with rotation coordinates system. Axial conduction in fluids was neglected to preserve the two dimensional nature of the problem. The temperature along the periphery is assumed to be constant for each wall. However, this constant temperature might be different for each of the four walls.

The swirling motion enhanced the heat transfer for a twist ratio of 2.5 and Reynolds number range of 1-1000. Similar expected enhancement was not observed for other twist ratios. Later Xu and Fan (1986) pointed out discrepancy in the viscous dissipation term adopted by Masilyah and Nandakumar (1981a and 1981b).

Chang *et al.* (1988) studied laminar flow in a twisted elliptic tube for large twist ratios (H = 21, 53, 106) using finite difference method. The effect of twist ratio and aspect ratio of ellipse is investigated with respect to their role in determining the axial and circumferential velocities and streamline patterns. Bishara (2010) numerically studied laminar, periodically fully developed single phase flow of a Newtonian fluid in helically twisted tube with constant wall temperature boundary condition for Re range of 10-1000 and Pr of 3.

Elliptical tubes with aspect ratio 0.3, 0.5 and 0.7 and twist ratio of 6, 9 and 12 were considered. Twisted elliptical tube showed considerable heat transfer enhancement compared to straight tube. Yang *et al.* (2011) experimentally evaluated performance of five twisted elliptical tubes. Aspect ratio (major diameter / minor diameter) of elliptical tubes used was in the range 1.49 to 2.15 and twist ratio range covered was 17.4 – 32.8. Water was used as the working fluid for Re range of 600 – 55000 covering laminar, transition and turbulent regime. They concluded that for twisted duct flow remains laminar for Re $\leq$ 2300. In twisted tube heat transfer enhancement is higher for laminar regime compared to transition and turbulent flow regime.

Literature review suggests that heat transfer and flow resistance charactertics of fluid flow inside twisted duct was studied. Numerical studies done on twisted elliptical tube are for Re range of 10 – 1000 and Pr range of 3 – 5. Twist ratios employed are in range of 6 – 106.

There is not much literature available on twisted square duct. Numerical solutions for entire laminar flow regime are not available. Prandtl number range covered is narrow.

Twist ratio used are high which will not contribute much to heat transfer enhancement studies. Literature does not address mechanism of heat transfer enhancement. Studies are discrete and there is a need for an equation to predict friction factor and heat transfer solution for a wide range of Reynold number and Prandtl number.

In the present study, an effort is made to study the pressure drop and heat transfer in a twisted square duct for a Reynolds number ranging from 100-2000 and Prandtl number ranging from 0.7 - 20. FLUENT is used as the CFD tool to generate the fluid flow and heat transfer distribution in a square twisted duct for a range of twist ratios (2.5 – 20). Heat transfer enhancement mechanism is explained by providing local distribution of friction factor and Nusselt number in duct cross section. Objective is to generate correlation to predict friction factor and Nusselt number for entire range of parameters studied.

II. COMMON TERMINOLOGY OF TWISTED SQUARE DUCT

Common terminologies associated with twisted duct have been highlighted in Fig. 1 are as follows

1) *Pitch (S):* The distance between two consecutive points along the length of tube where the orientation of tube cross section exactly coincides to each other. Cross section rotates by 360° along one pitch distance.
2) *Twist Ratio (H):* It is the geometrical parameter used to describe flow through twisted ducts. It is defined as the ratio of the pitch (S) to hydraulic diameter (d) of the duct.

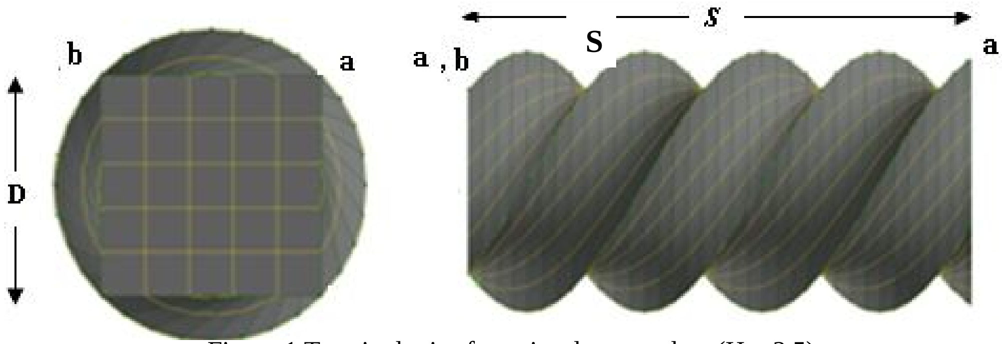

Figure 1 Terminologies for twisted square duct (H = 2.5)

III. NUMERICAL METHODOLOGY

FLUENT software has the ability to provide solution of fluid flow and heat transfer by using periodic flow concept. Periodic flow conditions exist when the flow pattern repeats over some length L, with a constant pressure drop across each repeating module along the flow direction. In the case of twisted duct, as the geometry of duct repeats, a suitable modelling length (S/4) of twisted duct was considered and periodic flow concept was utilized to obtain the solution. As for periodic flows, such problems can be analysed by restricting the numerical model to a single module of periodic length. This permits the use of a scaled model and reduces the computational time. Based on the experimental investigation of Yang *et al.* (2011) flow is considered to be laminar for studied Re range of 100 – 2000. Prandtl numbers considered are 0.7, 5, 10, 15 and 20. Twist ratio considered are 2.5, 5, 10 and 20. Three dimensional models are used in the analysis with hexahedron mesh elements. Peripherally and axially uniform wall conditions boundary conditions are used. Body forces due to gravity are neglected. Solutions are obtained for steady state, incompressible, hydrodynamically and thermally developed flow using FLUENT (version 6.2.16).

Common calculation methodology adopted in the present study is one case (H = 2.5, Re = 2000 and Pr = 0.7) of solution is presented. Twisted square duct (H = 2.5) of periodic Length (S/4) is meshed with hexahedron elements. Three meshes with elements 40000, 60000 and 80000 are generated. Figure 2 shows meshed model for twist ratio 2.5 with 80000 elements. Fluid flow and heat transfer solution is obtained in FLUENT as per the parameter listed in Table 1. Wall temperature (T_w), bulk temperature (T_b), heat flux (q_w) and shear stress (τ_w) data is obtained from FLUENT at different location along the length of the duct. Friction factor can be obtained from wall shear stress as

$$f = \frac{\tau_w}{\left(\frac{\rho V_m^2}{2}\right)} \tag{1}$$

Heat transfer coefficient (h) and Nusselt number (Nu) was calculated at each location along the length of tube and around wall

$$h = \frac{q}{(T_w - T_b)} \tag{2}$$

$$Nu = \frac{h.d}{k} \tag{3}$$

Average fluid flow and Nusselt number results are provided in Table 2. Difference in results between mesh 2 and mesh 3 is less than 0.4 %. Hence, the results of mesh 3 with 80000 elements was considered as the results for case of H = 2.5, Re = 2000 and Pr = 0.7. In similar way, results are obtained for twist ratio 5, 10, and 20 using mesh with hexahedron element 120000, 140000, and 180000 respectively. Machine used for computation is Intel core 2 Duo processor with speed of 2.6 GHz.

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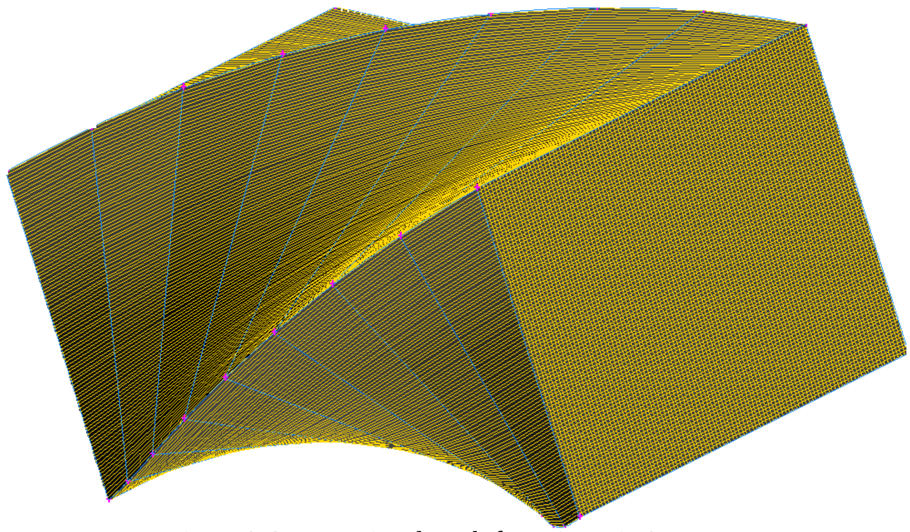

Figure 2 Computational mesh for twist ratio 2.5

Table 1 Parameters used in FLUENT

Models	Solver	Segregated
	Formulation	Implicit
	Space	3D
	Time	Steady
	Viscous model	Laminar
Boundary conditions	Inlet	Periodic
	Outlet	Periodic
	Wall	Constant temperature
	Periodic type	Translational
	Periodicity condition	Mass flow specified
Discretization	Pressure	Second order
	Momentum	Second order upwind
	Energy	Second order upwind
	Pressure – velocity coupling	Simple
Convergence	Continuity	10^{-7}
	Velocity	10^{-6}
	Energy	10^{-7}

Table 2 Grid Independence results for H = 2.5, Re = 2000 and Pr = 0.7

Models	Element	$\bar{f}$ Re	$\overline{Nu_T}$
Mesh 1	40000	48.22	13.47
Mesh 2	60000	47.77	12.52
Mesh 3	80000	47.71	12.49

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IV. VALIDATION WITH LITERATURE

CFD solution for fully developed flow in a twisted square duct for (H = 2.5) at Re = 709 was obtained. The results are compared with Masliyah and Nandakumar (1981a) and presented in Table 3. The results compare reasonably well with the literature.

Table 3 Comparison of solution with literature

Re	Present	Masilyah & Nandakumar (1981a)
	$\bar{f}.Re$	$\bar{f}.Re$
709	25.7	26.08

A. *Average Friction Factor*

f.Re values obtained for different twist ratios are presented in Fig. 3. It is observed that f.Re values for twisted duct are functions of Re and H. Maximum value of f.Re is 47.7 obtained for a twist ratio of 2.5 at Re = 2000. Numerical data for f.Re can be correlated in terms of swirl parameter Sw for a twisted duct as shown in Fig. 4. Sw is a modified form of swirl parameter used by Manglik and Bergles (1993) for twisted tape. Correlation (based on 25 data points) obtained is given by Eq. (4) and predict values within 5 % of actual numerical values.

$$f.Re = 0.0217Sw + 14.2 \tag{4}$$

$(100 \le Re \le 2000)$ and $(2.5 \le H \le 20)$

Where Sw is defined by Eq.(5)

$$Sw = \frac{Re}{\sqrt{H}} A_r \tag{5}$$

Where A_r is heat transfer area ratio of twisted tube to its untwisted counterpart having same flow cross section area and same flow length and is defined by Eq.(6)

$$A_r = \frac{A_{st}}{A_{tw}} \tag{6}$$

$$A_{st} = 4Sd_h \tag{7}$$

$$A_{tw} = 4\pi d^2_h \left[0.5\left(1+\frac{H^2}{\pi^2}\right) + \frac{H^2}{2\pi^2}\ln\left\{\frac{\pi}{H} + \left(1+\frac{\pi^2}{H^2}\right)^{\frac{1}{2}}\right\}\right] \quad (8)$$

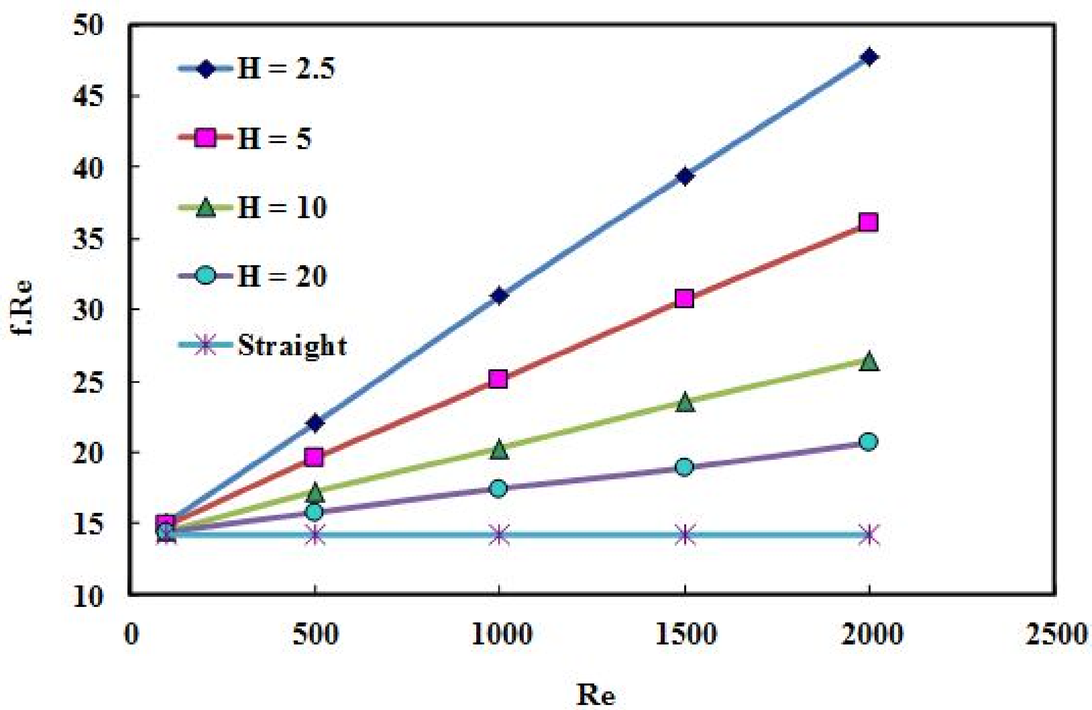

Figure 3 Variations of average f.Re with Re for different twist ratio

A_r obtained from Eq. (6) is represented by Fig.5. It is observed that the area ratio decreases with the increase in the twist ratio. Hence, not much advantage is obtained by employing twisted duct having H greater than 20 for heat transfer enhancement. 6

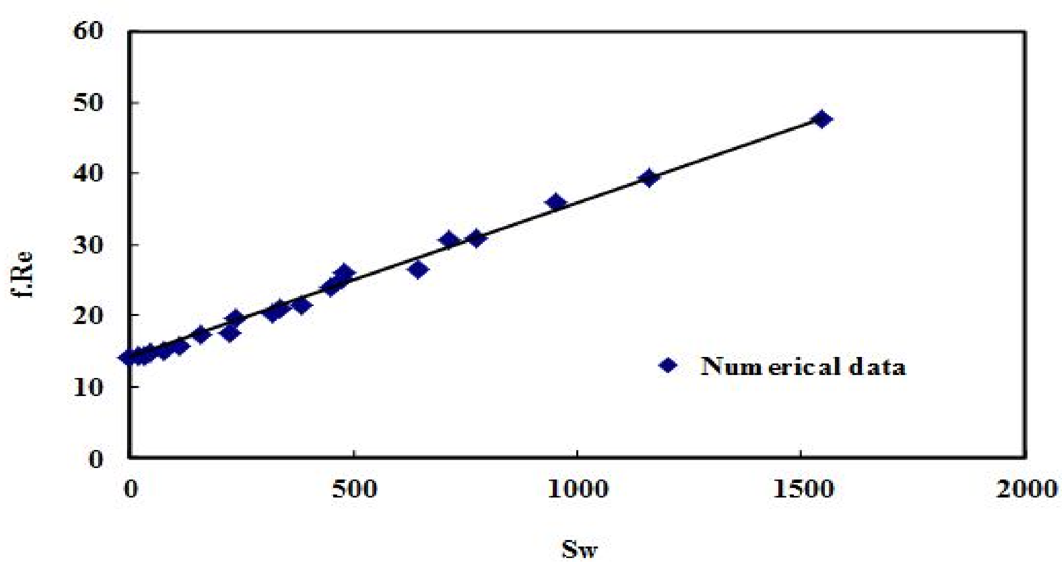

Figure 4 Variations of average f.Re with swirl parameter

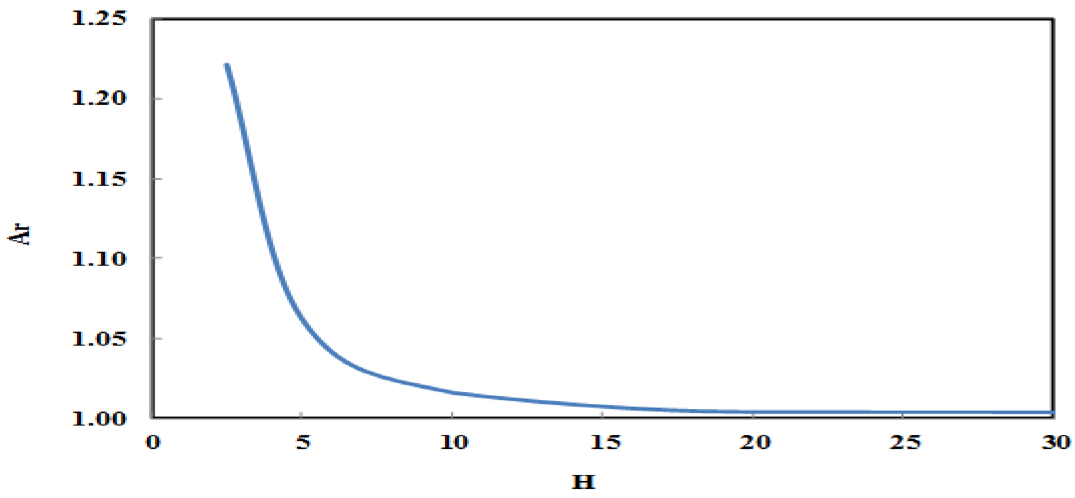

Figure 5 Effect of twist ratio on heat transfer area

B. Average Heat Transfer

Average $\overline{Nu}_T$ values are calculated for uniform wall temperature boundary condition. Pr values considered are 0.7, 5, 10, 15 and 20. Figure 6, 7, 8, 9 and 10 show $\overline{Nu}_T$ values for Pr of 0.7, 5, 10, 15 and 20 respectively. Maximum $\overline{Nu}_T$ value is 92.5 obtained for H = 2.5 at Re = 2000. Numerical data (based on 125 data points) of $\overline{Nu}_T$ at different Pr was correlated. Equation 9 predicts Nusselt number values within 20 % accuracy of numerical data. Accuracy of prediction can be improved within 8 % by using polynomial fit given by Eq.10 obtained from Fig.11.

$$\overline{Nu}_T = 0.002\, Sw^{0.825}\, Pr^{0.845} + 2.8 \qquad (9)$$

$$\overline{Nu}_T = 2.8 + 0.0092\, B - 10^{7}\, B^{2} \qquad (10)$$

$(100 \le Re \le 2000)$, $(2.5 \le H \le 20)$ and $(0.7 \le Pr \le 20)$ where B is

$$B = Sw\ \ Pr^{0.7}$$

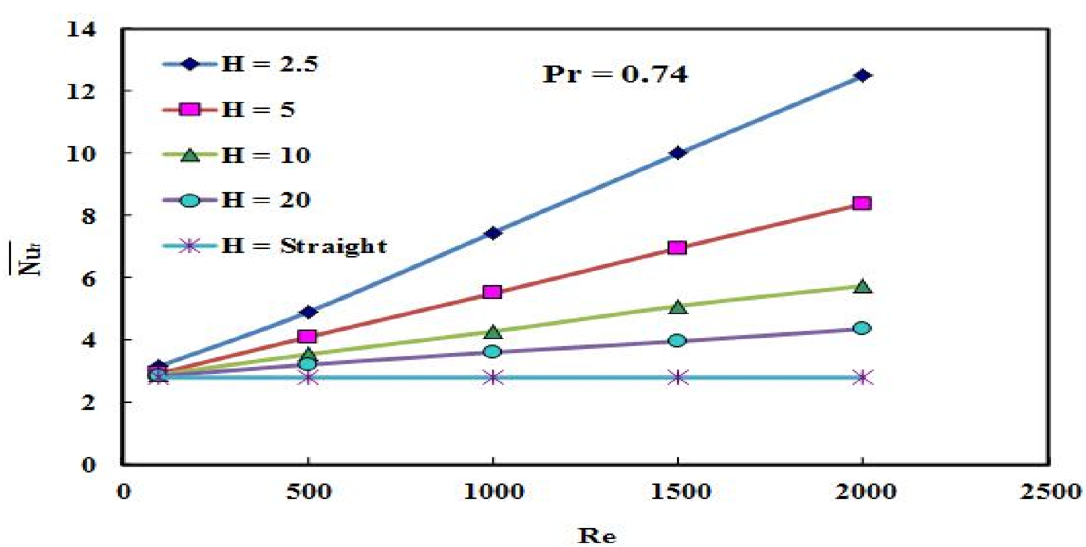

Figure 6 Heat transfer solution for Pr = 0.7

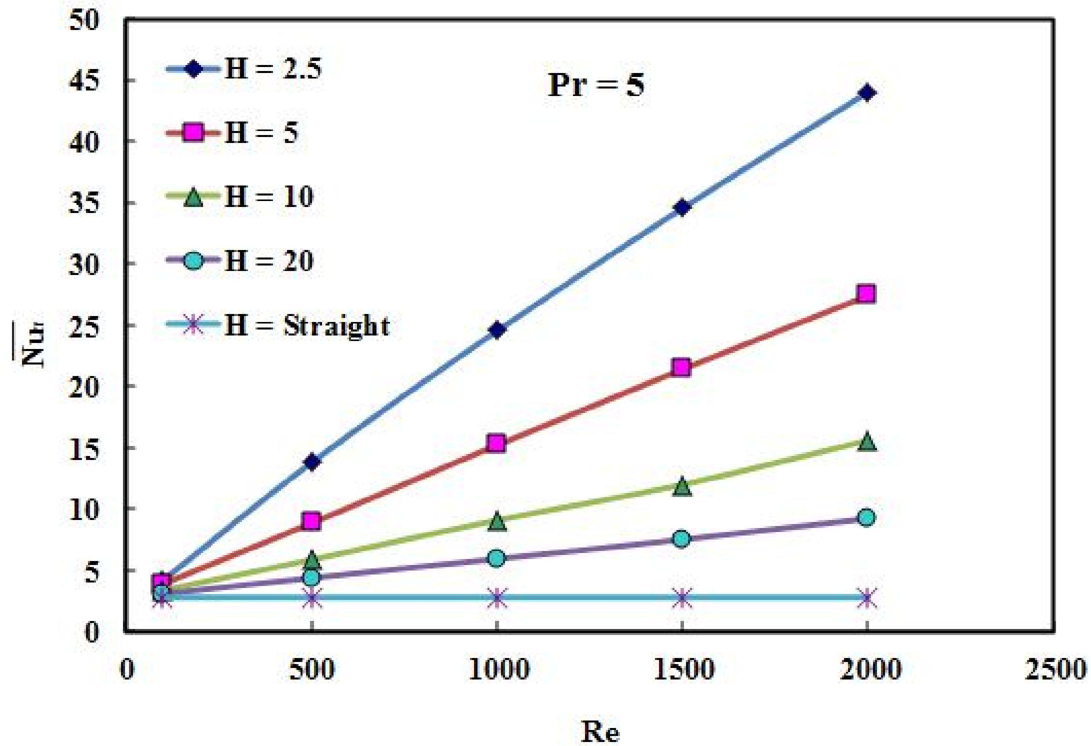

Figure 7 Heat transfer solution for Pr = 5

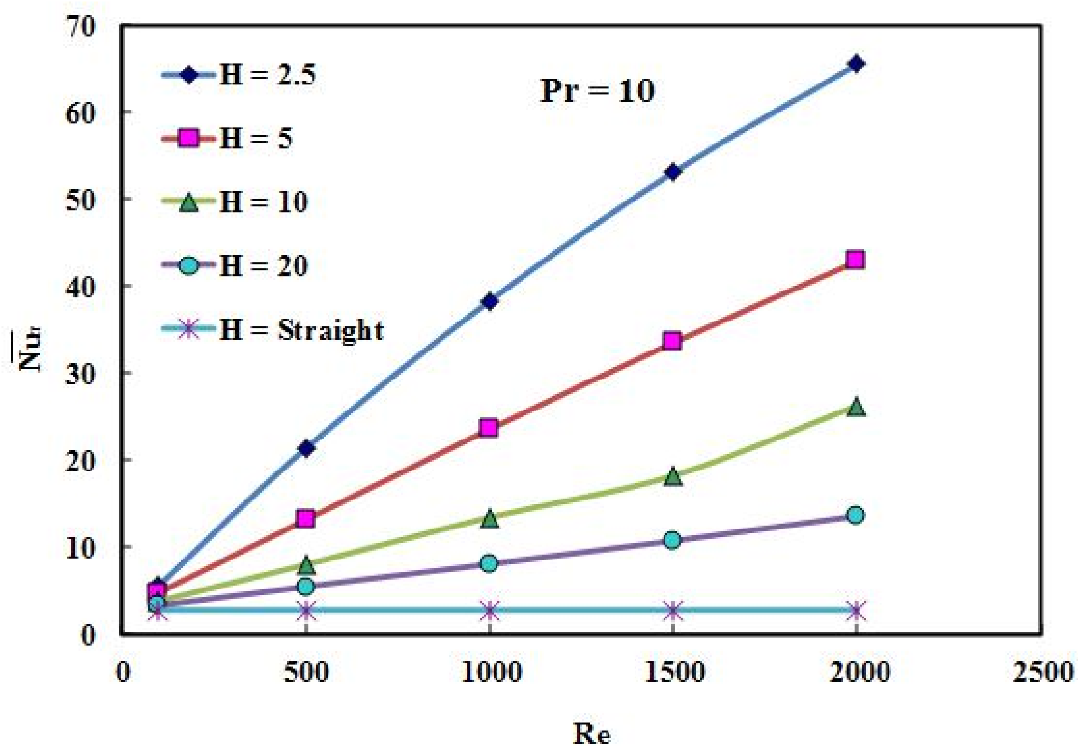

Figure 8 Heat transfer solution for Pr = 10

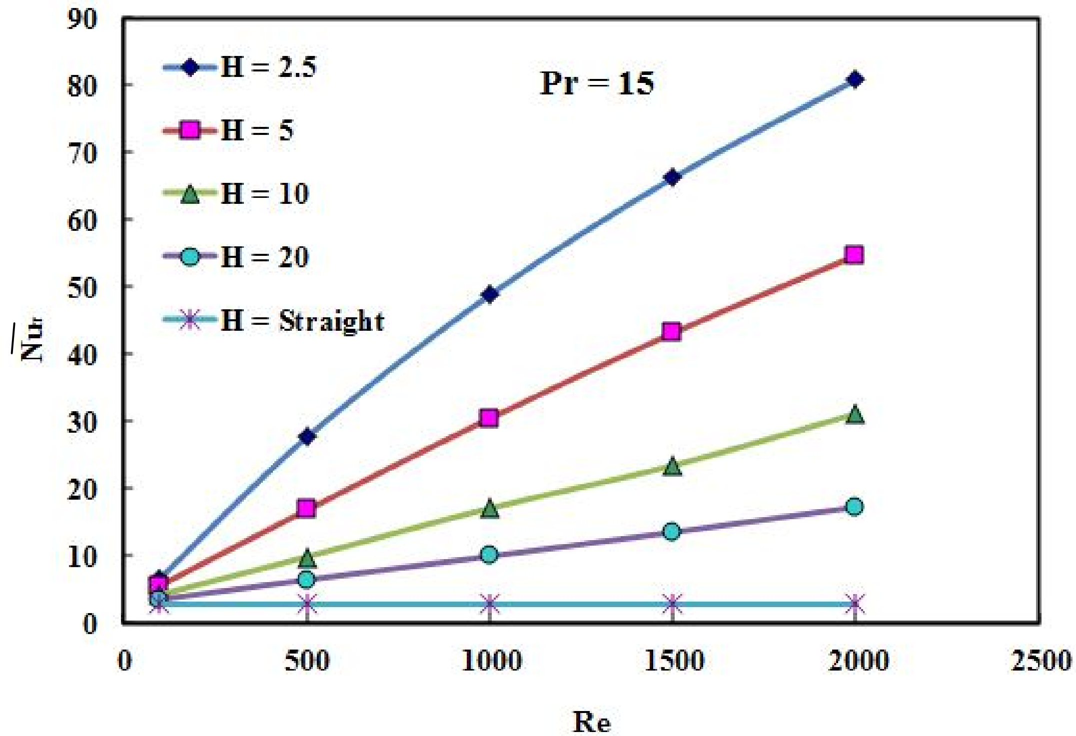

Figure 9 Heat transfer solution for Pr = 15

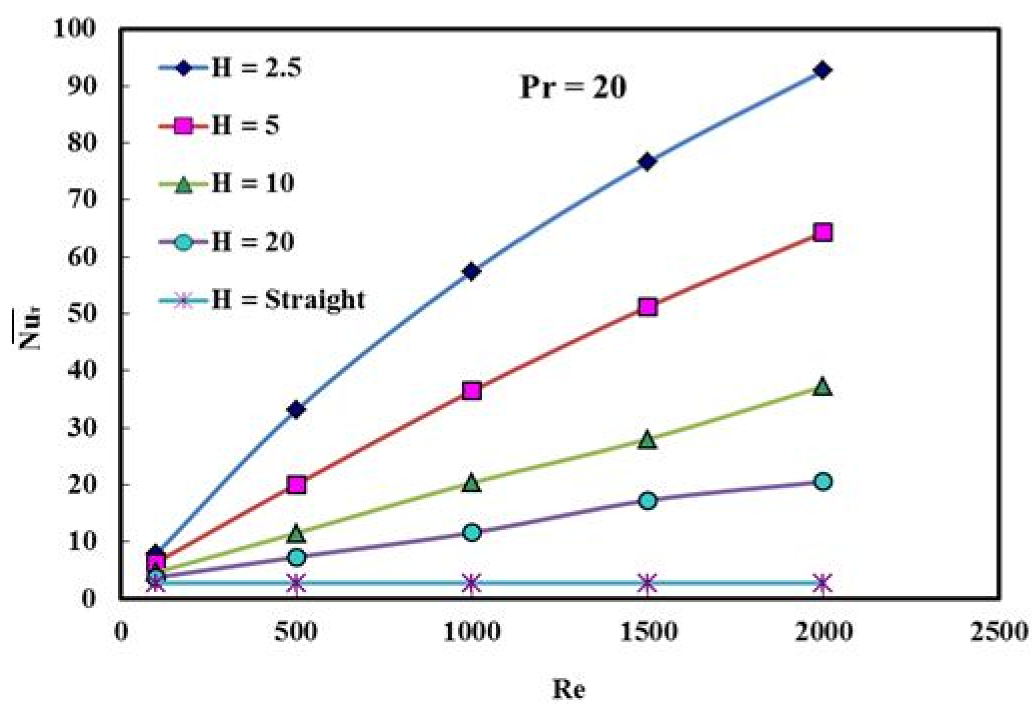

Figure 10 Heat transfer solution for Pr = 20

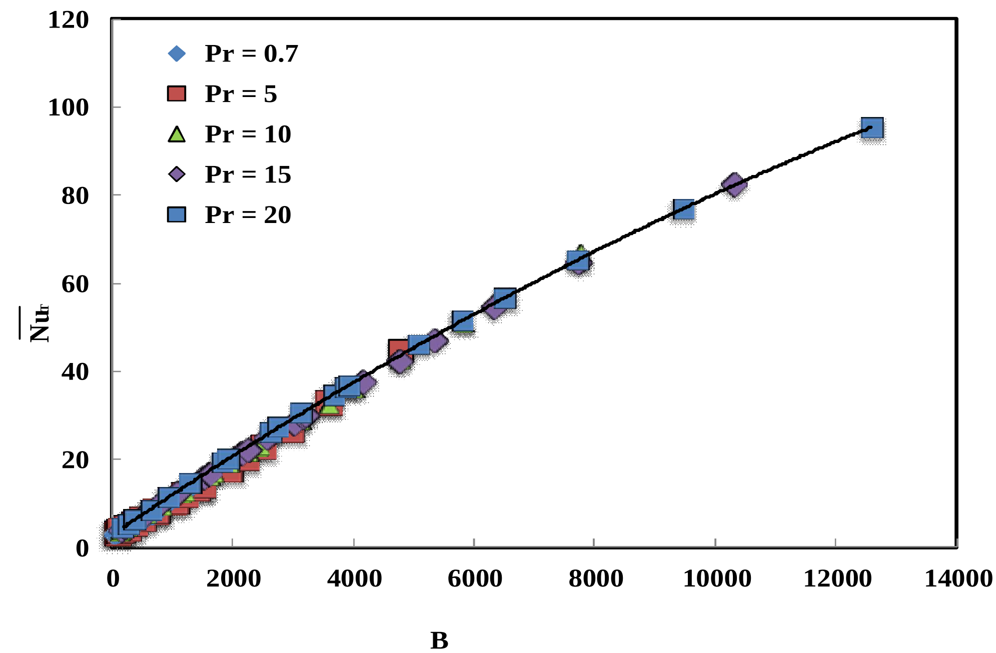

Figure 11 Correlation fit for Nu_T and Swirl Parameter

V. RESULTS AND DISCUSSION

In the previous section, f.Re ~~and~~ Nu_T ~~values~~ calculated for twisted square duct were dependent on Re, Pr and H values, which is contrary to straight square duct for which f.Re and Nu_T remains constant in laminar flow regime. In this section, local variation of f.Re and Nu_T is studied. Figure 12 shows cross section of square tube. Z axis is the flow direction and X and Y axes are in plane of cross section. Line cb represents half edge of square side. Line oc and line ob are half side and half diagonal of cross section with o as center of cross section.

A. Local Friction factor variation

Figure 13 shows local f.Re variation along half edge cb for twist ratio 10. In Straight tube maximum f.Re occurs at centre of side of duct and zero at corner. However for twisted duct, location of maxima is function of Re and its location shift away from centre of side with increase in Re. This is attributed to the presence of secondary flow in twisted duct. Figure 13 shows contours of dimensionless x velocity (V_x / V_{mi}) for different twisted ratio at Re = 1000. It shows strong presence of secondary flow near the wall.

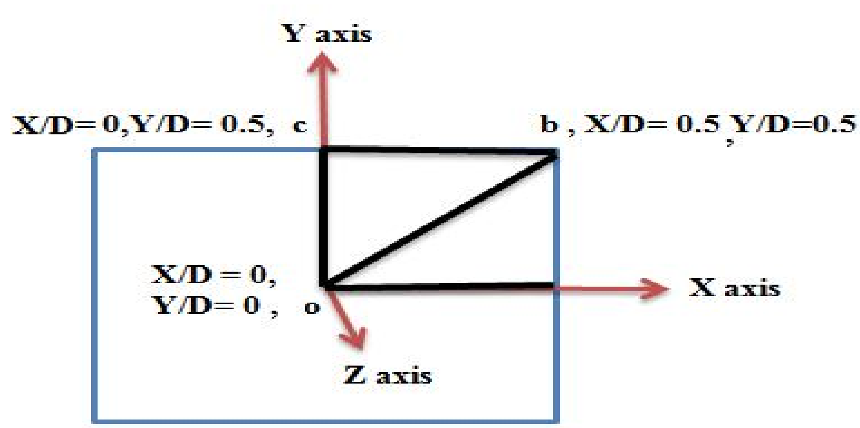

Figure 12 Cross section of Square duct

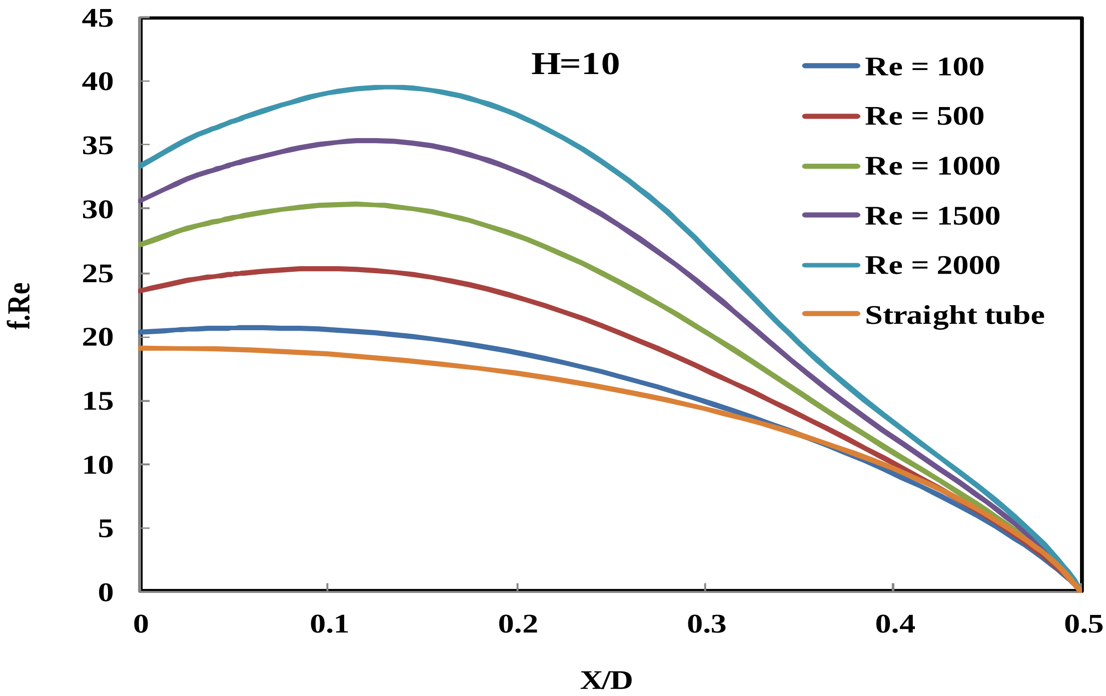

Figure 13 Local f.Re variations along wall (cb) for H = 10

Region. Also it is seen that magnitude of secondary flow is higher for small twist ratio. To analyse it further V_x and V_z component of velocities are plotted for different twist ratios. Figures 15 and 16 shows V_x and V_z component of velocities respectively along line ob. Velocities are non-dimensionalized with respect to mean inlet velocity (V_{mi}). It is seen from Fig. 15 that secondary flow reduces with increase in twist ratio. From Fig.16 it is observed that maximum V_z occurs at center of duct and it is 2.1 times that mean inlet velocity or mean velocity at given cross section for H= 2.5 & Re = 2000. It can be concluded that V_x (same as V_y) component of velocity is mainly responsible for creating swirl in flow and it is more effective at smaller twist ratios. Contrary to straight square duct where V_x and V_y component of velocities are nearly zero compared to V_z component, twisted duct shows comparable values of this components. V_x and V_y components of velocities observed are maximum for H = 2.5 and is about 2.56 % of inlet mean velocity. It is also seen that location of maximum value of V_x and V_y velocity component is in region of X/D = 0.3-0.5, which is near wall region. For a given twist ratio, as Re is increased, location of maximum value of x and y component shifts towards wall. For a given Reynolds number as twist ratio is decreased location of maximum of x and y component shifts towards wall. X and y components of velocity generated (also referred as secondary flow) are result of swirling motion occurring inside the twisted tube. As magnitude and location of secondary flow are function of Re and twist ratio, Shear stress at wall and hence f.Re values will also be function of the same. Observation leads to conclusion that swirling motion (secondary flow) is responsible for higher f.Re values and it distribution at given cross section in case of twisted duct.

B. Local heat Transfer Variation

In the earlier section, it was found that Nu_T values ~~for t~~wisted square ducts were dependent on Re and Pr values. In this section local Nusselt number variation along the wall has been analysed. Figure 17 shows local Nu_T variation along cb at Re = 2000. It is seen that for same twist ratio as Nu increases with Pr. It is observed that at given Re and Pr as twist ratio decrease Nu_T increases. Also at given Re and H as Pr increase Nu_T values increases. Figure 18, 19, 20 and 21 shows Nu_T solutions for twist ratio 5, 10, 15 and 20 respectively. It is seen that location of maximum Nu_T is not at center of edge (point c) as expected for straight duct .Location of maximum Nu_T point depend on H and Re. Location of maximum Nu_T point shifts away from point c with increase in the magnitude of secondary flow. This is important observation as it will determine location of maximum heat flux point on duct wall.

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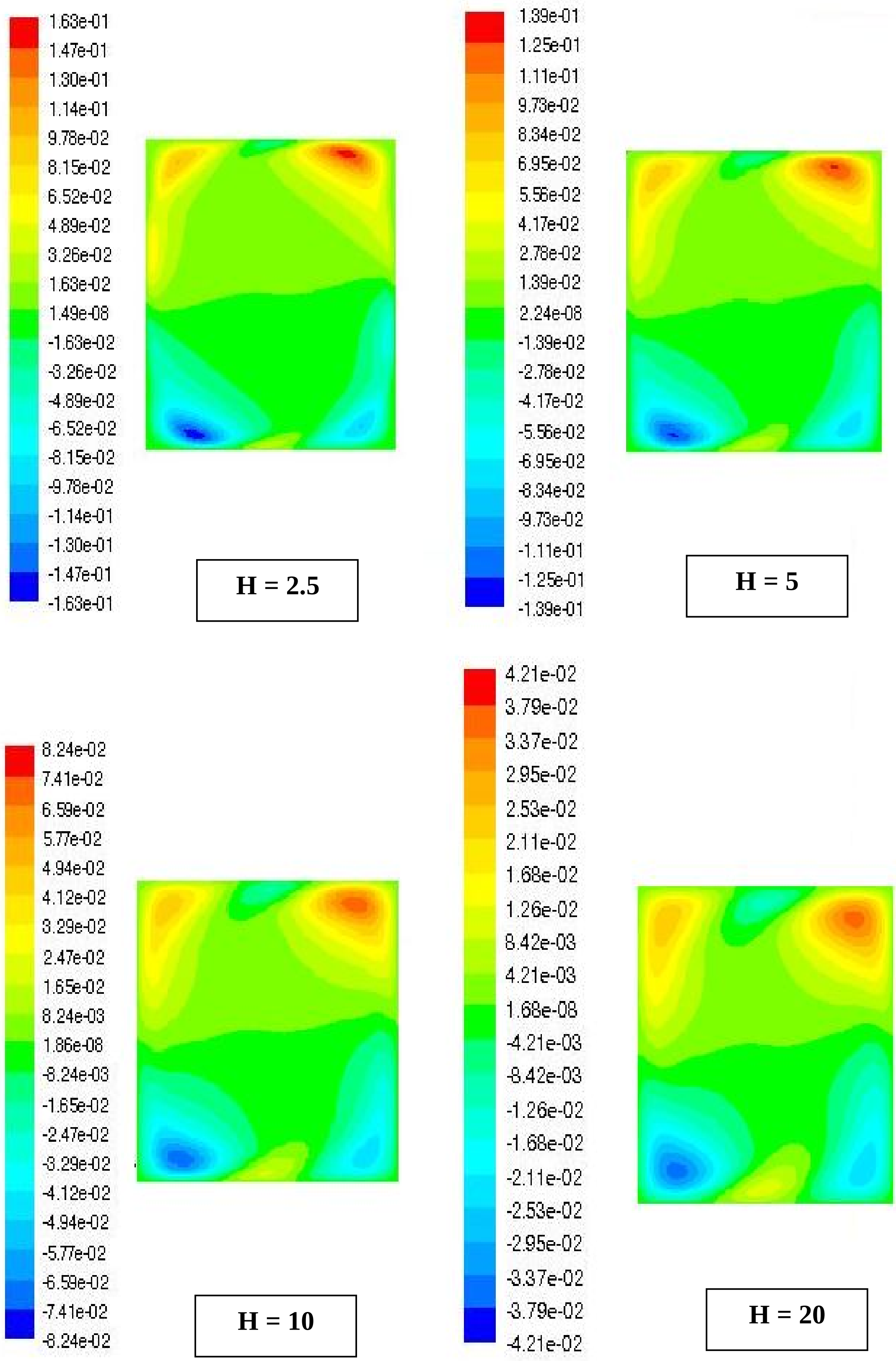


Figure 14 Contours of Non dimensional x velocity (V_x / V_{mi}) for different twist ratio at Re = 1000

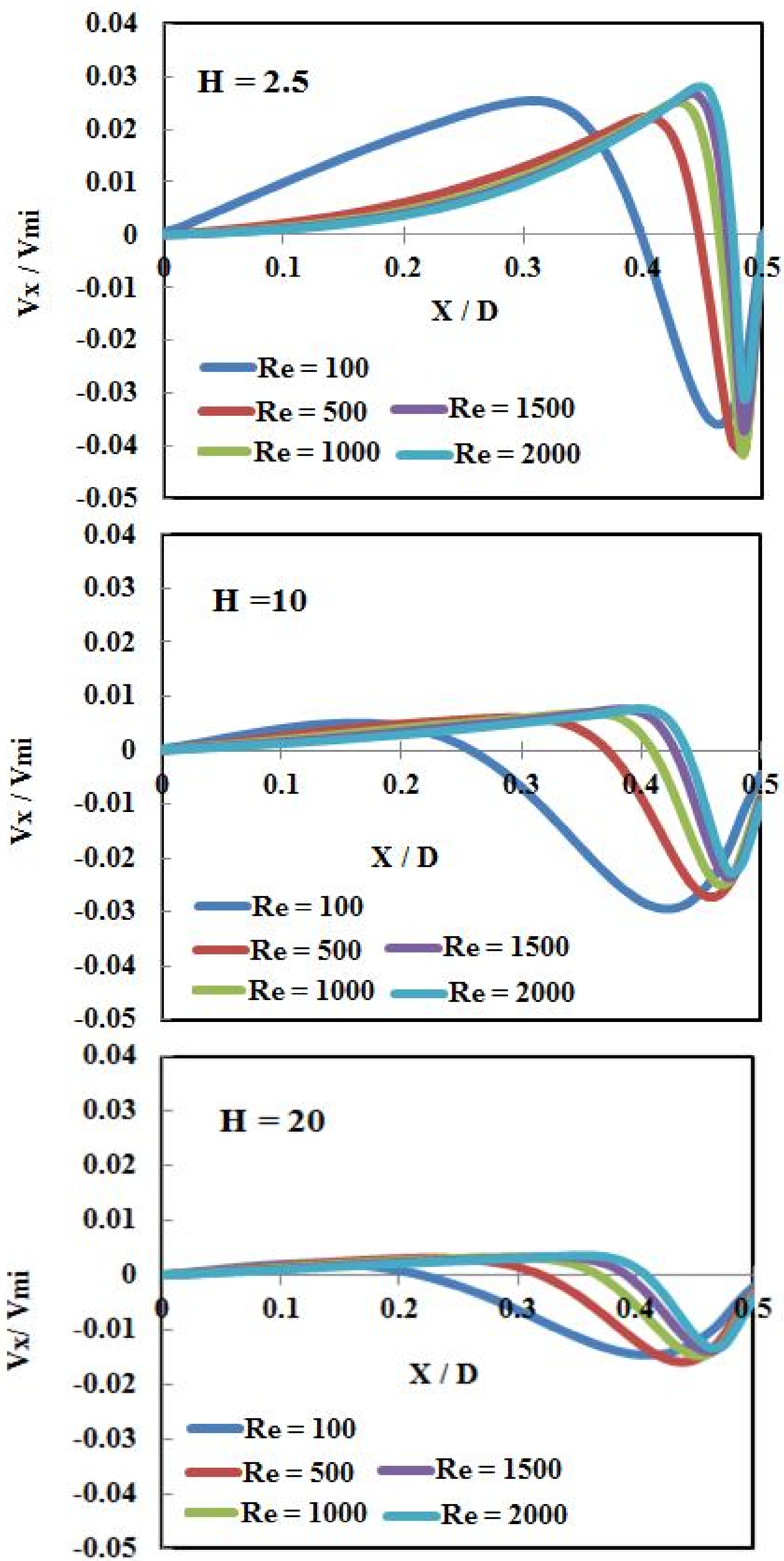


Figure 15 Effect of twist ratio on secondary flow (line ob)

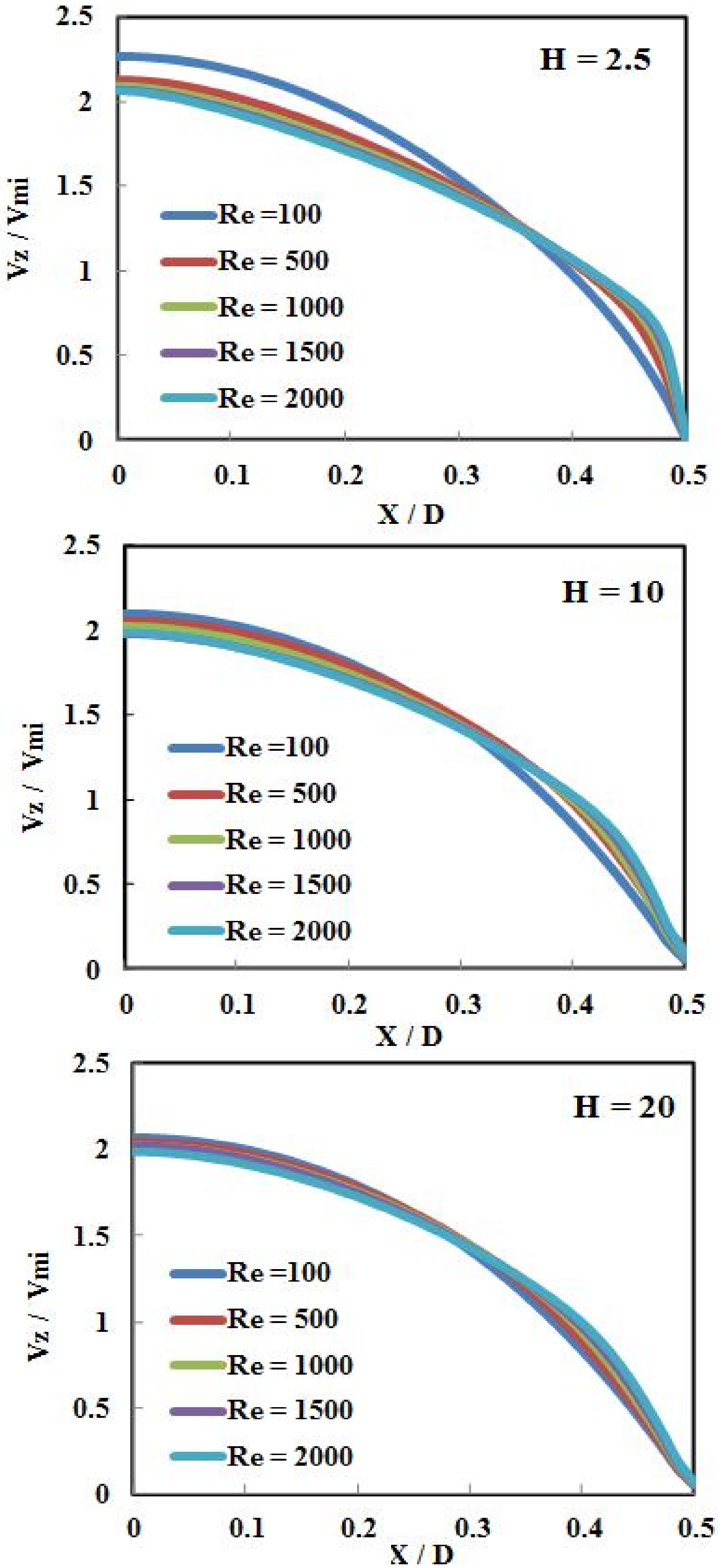


Figure 16 Effect of twist ratio on velocity in direction of flow.

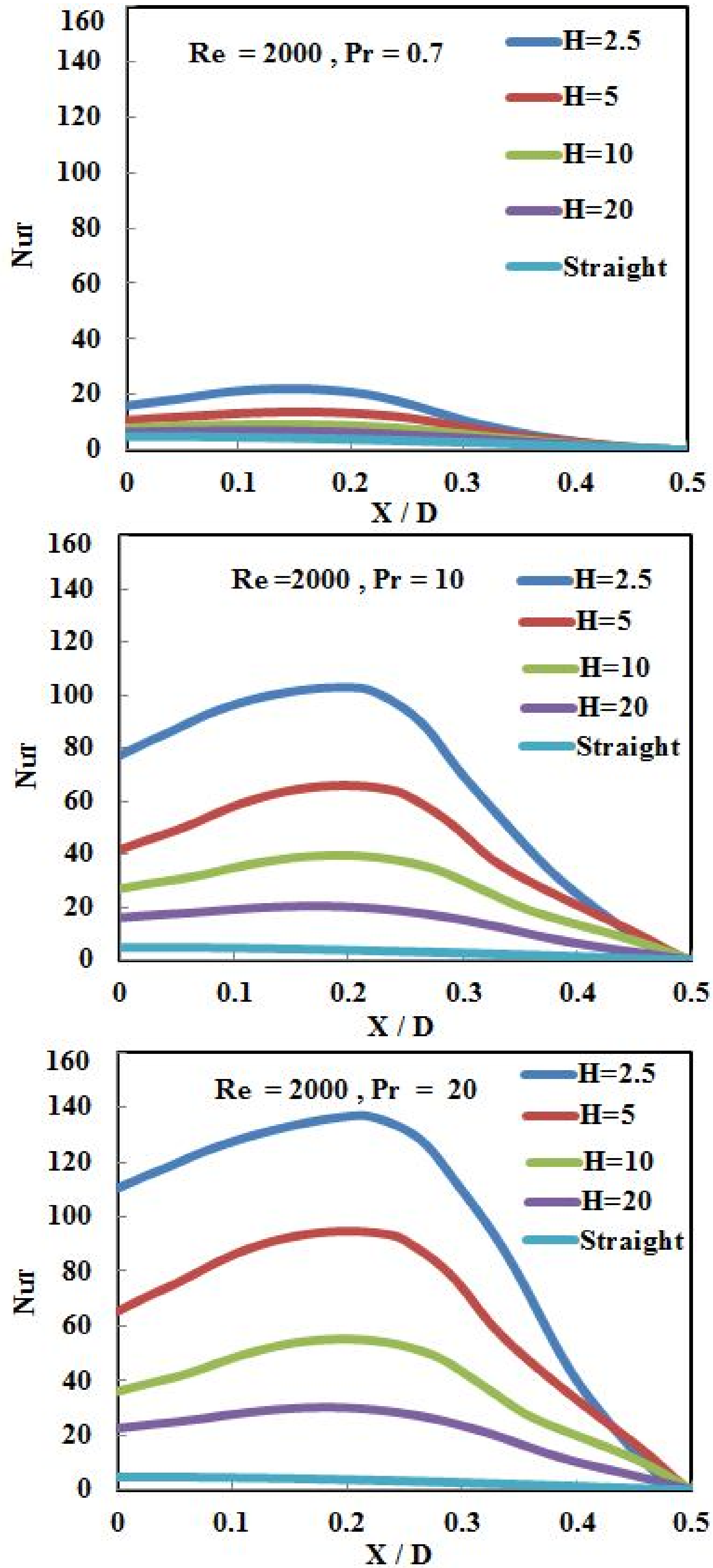


Figure 17 Effect of Prandtl Number on heat transfer at Re = 2000

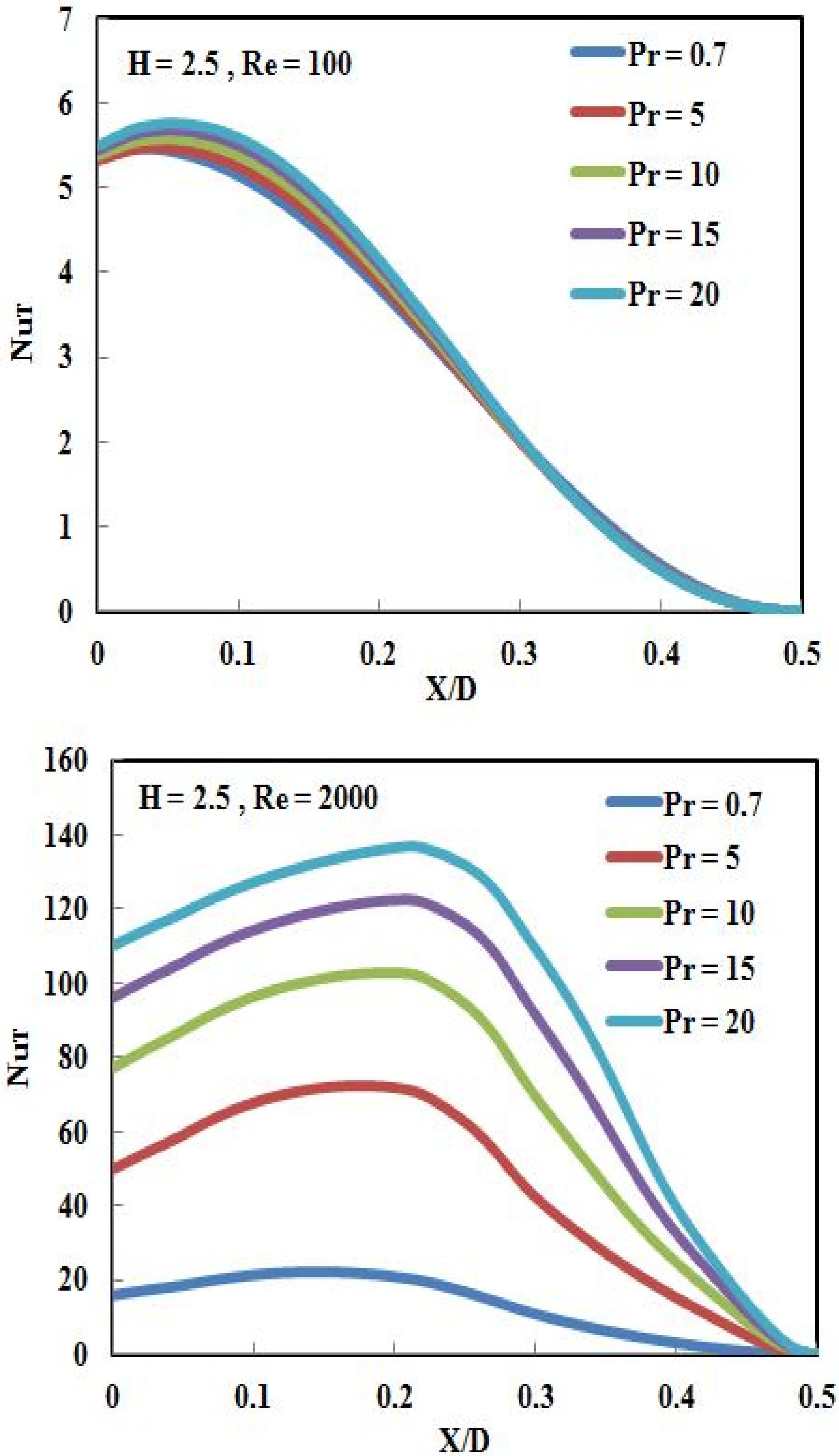


Figure 18 Local heat transfer solution at wall (cb) for H = 2.5

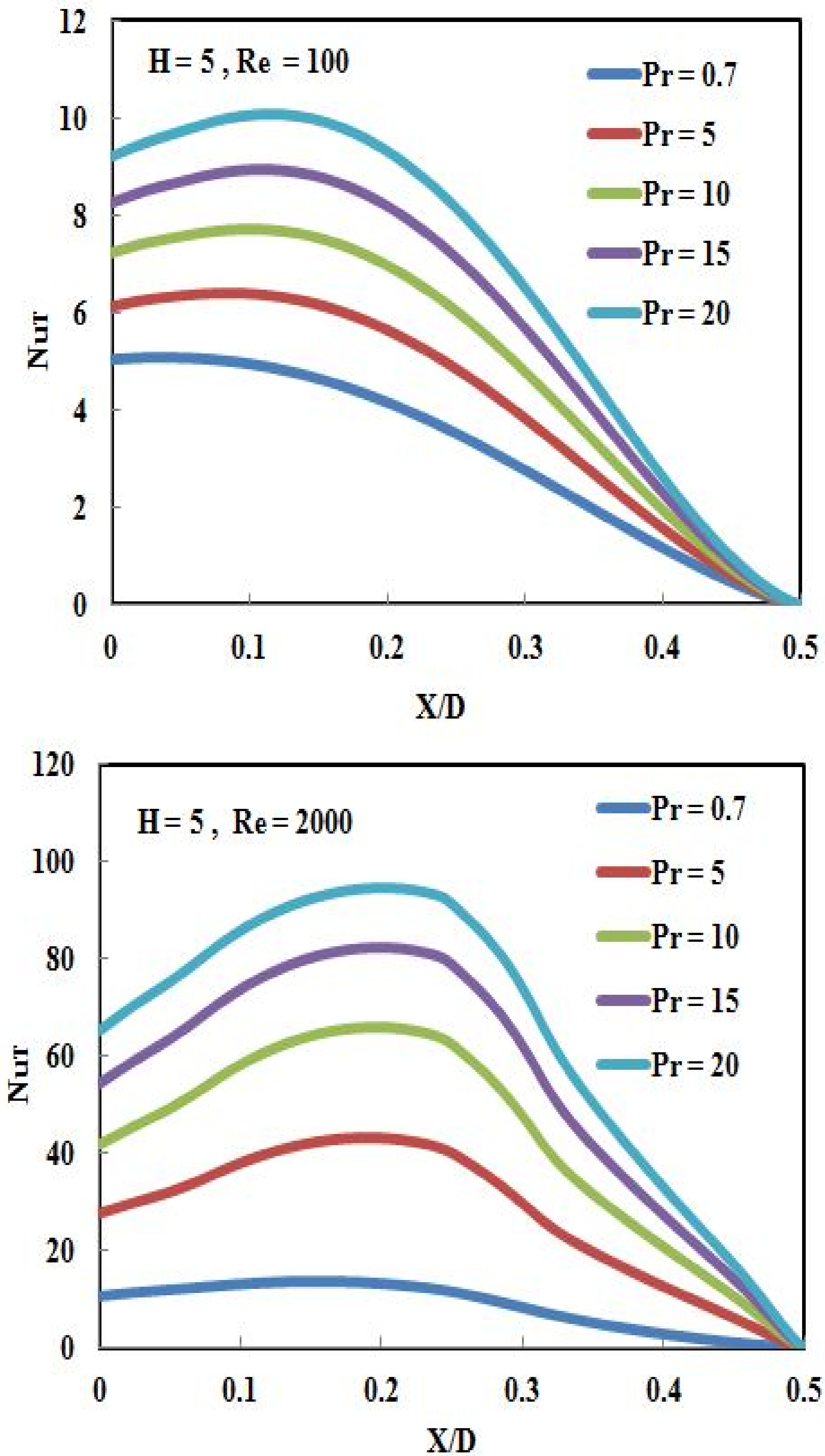
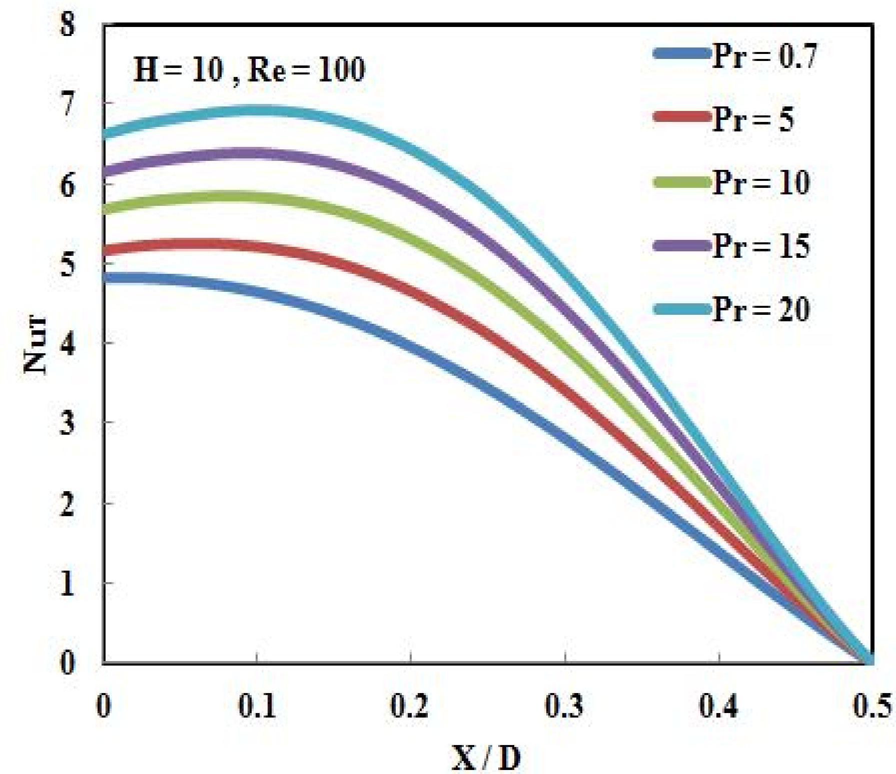


Figure 19 Local heat transfer solution at wall (cb) for H = 5



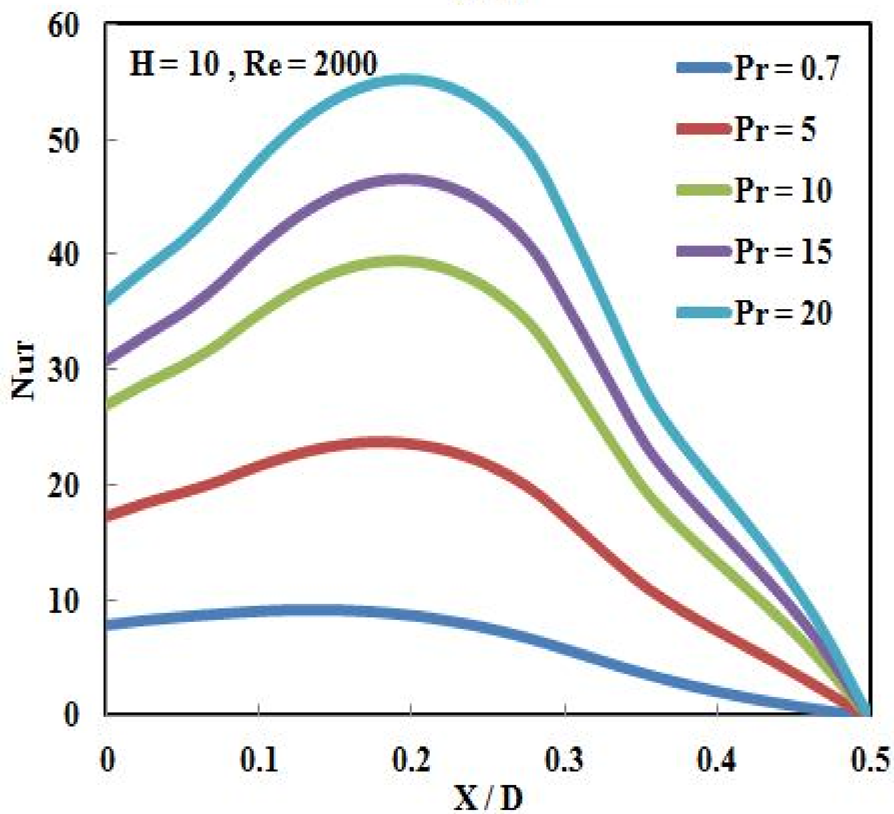


Figure 20 Local heat transfer solution at wall (cb) for H = 10

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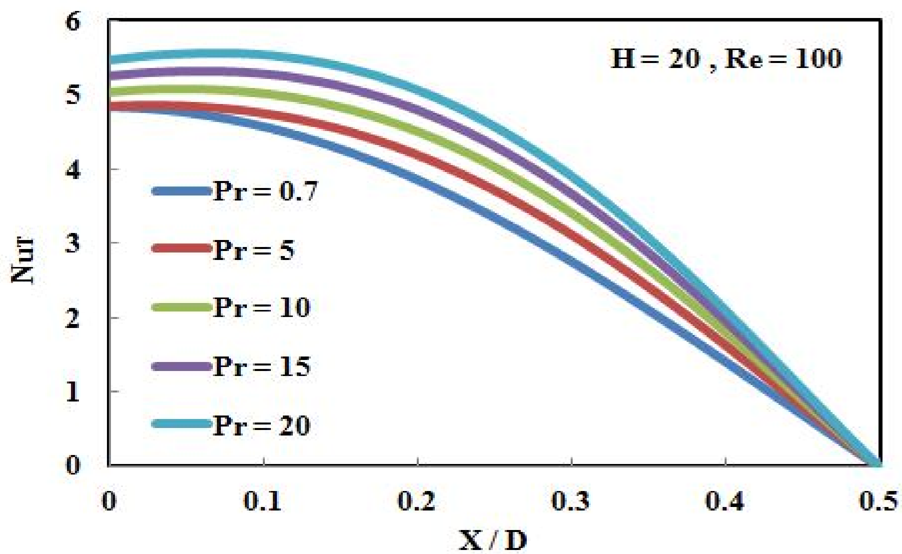


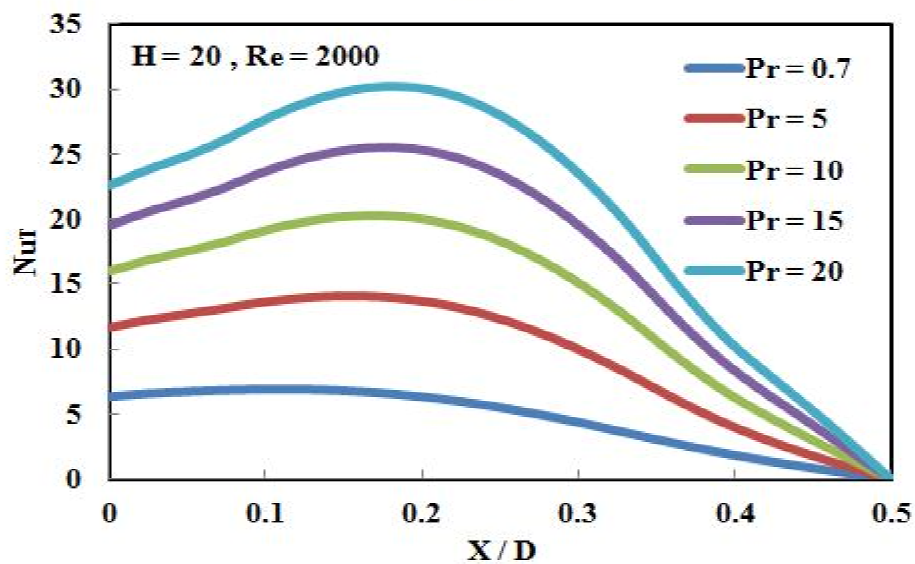


Figure 21 Local heat transfer solution at wall (cb) for H = 20

C. Heat Transfer Enhancement

In the earlier sections it is realised that average f.Re and Nu_T-values in ~~twis~~ted duct are higher than straight duct. Hence suitability of this heat transfer enhancement can be better judged by comparing heat transfer enhancement with respect to rise in f.Re values. A suitability factor E can be defined, as shown by Eq. 11. Table 4 shows heat transfer enhancement factor (E) for twisted tube with respect to increases in f.Re values for Pr = 0.7. It is seen that twist duct provides considerable increases in heat transfer up to twist ratio 20.

$$E = \frac{\dfrac{(\overline{Nu}_T)tw}{(\overline{Nu}_T)st}}{\dfrac{(\bar{f}.Re)tw}{(\bar{f}.Re)st}} \qquad (11)$$

Table 4 Enhancement factor for twisted tube (Pr = 0.7)

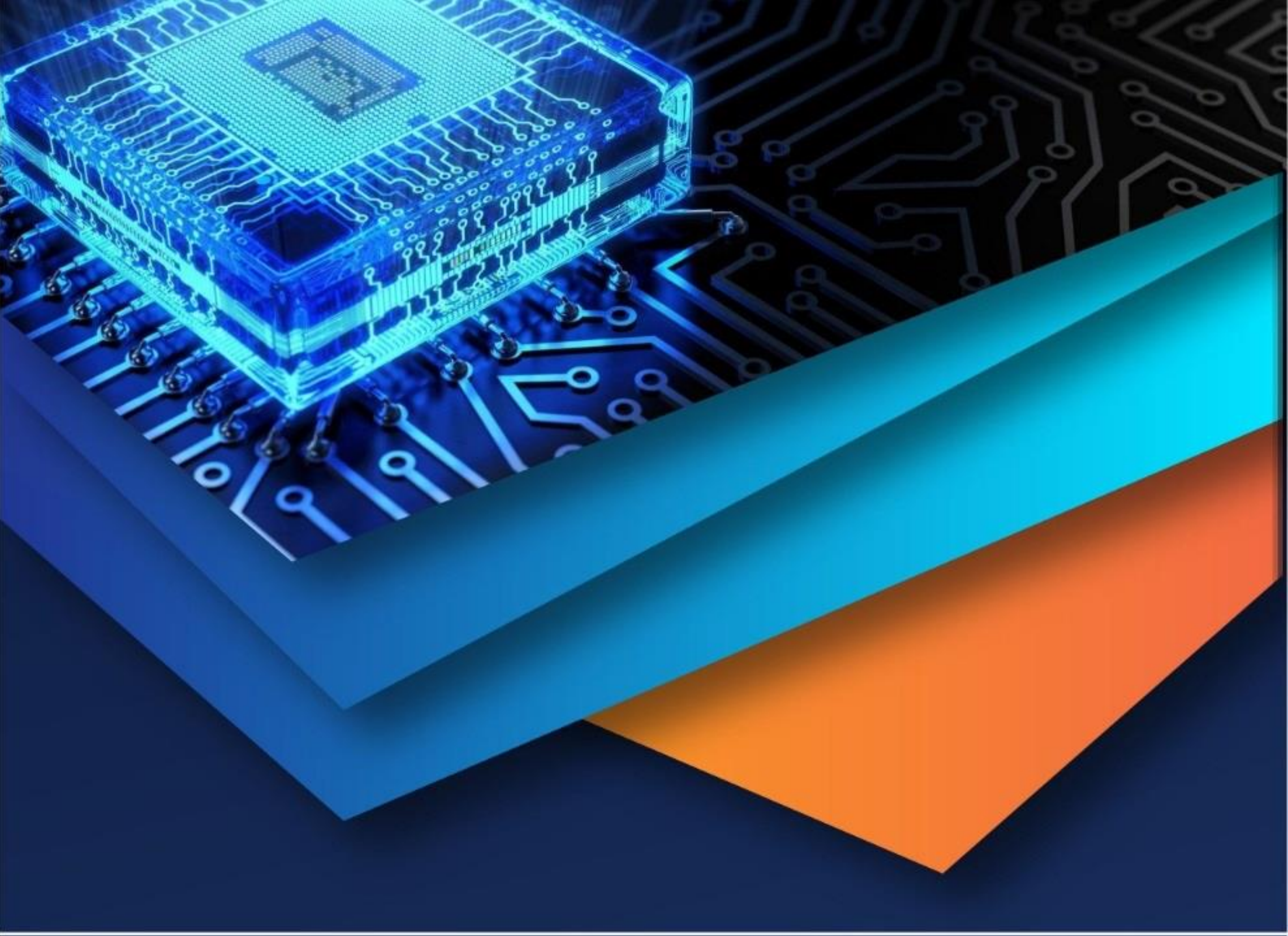
E					
Re \ H	2.5	5	10	20	Straight
100	1.07	1.05	1.02	1.01	1.00
500	1.13	1.06	1.04	1.03	1.00
1000	1.22	1.11	1.07	1.05	1.00
1500	1.29	1.15	1.10	1.07	1.00
2000	1.33	1.18	1.10	1.07	1.00

VI. CONCLUSIONS

In the present work, fluid flow and heat transfer solution had been obtained for fully developed laminar flow with uniform peripheral wall temperature boundary condition. Reynolds number range considered was 100 – 2000 and Prandtl number was 0.7 – 20. Twist ratio used was 2.5, 5, 10 and 20. Average friction factor and Nusselt number correlation are generated in terms of swirl parameter. Maximum average Nusselt number was reported was for H = 2.5 at Re = 2000. Nusselt number was found to be direct function of Re and Pr and inverse function of H. Local distribution of friction factor and Nusselt number are provided and it is found that location of maxima changes with Reynolds number, Prandtl number and twist ratio. Comparison of heat transfer enhancement with respect to friction factor had been provided in terms of enhancement factor. Maximum value of enhancement factor for Pr = 0.7 is 1.33 and it is for H = 2.5 at Re = 2000. It is found that twisted tube enhancement heat transfer in the studied range of parameters hence it is good substitute for straight tube in heat exchange equipment's operating under similar parameters. These will contribute in one way for development of compact heat exchanger.

REFERENCES

[1] Bishara F., 2010. Numerical simulation of fully developed Laminar flow and Heat transfer in Isothermal Helically Twisted tubes with Elliptical cross-section, M.S. Thesis, University of Cincinnati, Cleveland state, USA.
[2] Chang K.S., Choi J.S. and Kim J.S., 1988. Laminar fluid flow in a Twisted Elliptic tube, KSME Journal, Vol.2, No.1, pp. 44-51.
[3] Manglik R.M. and Bergles A.E., 1993. Heat transfer and Pressure drop Correlations for Twisted –Tape inserts in isothermal flows, Journal of Heat transfer, Vol.115, pp. 881-889.
[4] Masliyah J.H. and Nandakumar K., 1981a. Steady Laminar flow through Twisted pipes - Fluid flow in Square tubes, ASME journal of Heat Transfer, Vol. 103, n4, pp.785-790.
[5] Masliyah J.H., and Nandakumar K., 1981b. Steady Laminar flow through Twisted pipes - Heat transfer in Square tubes, ASME journal of Heat Transfer, Vol. 103, n4, pp. 791-795.
[6] Todd L., 1977. Some comments on steady Laminar flow through Twisted pipes, Journal of Engineering Mathematics, Vol.11, pp. 29-48.
[7] Xu C.G. and Fan D.N., 1986. Some Remarks on the Helical Cartesian coordinates system and its application, ASME J. Heat transfer, 108, pp. 483-486.
[8] Yang S., Zhang L. and Xu H., 2011. Experimental study on Convective heat transfer and flow resistance characteristics of water flow in Twisted Elliptical tubes, Applied Thermal Engineering, 2011, Article in press, pp. 1-11.



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