CFD analysis of fully decaying, partially decaying and partly swirl flow in round tubes with short length twisted tapes

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Abstract

CFD investigation was carried out to study the heat transfer characteristics of air flow inside a circular tube with a fully decaying, partially decaying and partly swirl flow. Four combinations of tube with twisted-tape inserts, half-length upstream twisted tape condition (HLUTT), half-length downstream twisted tape condition (HLDTT), full-length twisted tape (FLTT), inlet twisted tape (ILTT) are considered along with plain tube (PT) for comparison.. Three different twist parameter, $\lambda = 0.14$, 0.27, and 0.38, for twisted tape configuration have been studied for the above four configurations. 3D numerical simulation was performed for an analysis of heat transfer and fluid flow for turbulent regime. The results of CFD investigations of heat transfer, and friction characteristics are presented for the FLTT, HLUTT, HLDTT and the ILTT along with a velocity and temperature profiles analysis in comparison with the PT case.

Keywords: HLUTT, HLDTT and FLTT, enhancement, Tape inserts, partially decaying swirl flow.

Nomenclature

$C\mu$	turbulence model constant
D	inner tube diameter, m
E	total energy, J
f	friction factor
h	enthalpy, J or convective heat transfer coefficient, $Wm^{-2} K^{-1}$
k	thermal conductivity, $W\ m^{-1} K^{-1}$
keff	effective thermal conductivity, $W\ m^{-1} K^{-1}$
L	test section length, m
Nu	Nusselt number
p	static pressure, Pa
Δp	pressure drop, Pa
Re axial velocity	Reynolds number, Re_a , Axial Reynolds number $= \rho U\ \ \ D/ \mu$, where U will be the mean
u	mean velocity, $m\ s^{-1}$
ui′	fluctuation velocity components, $m\ s^{-1}$
w	tape width, m
y	twist ratio (numbers of diameter per 180° twist, H / D)
H	pitch for 180° rotation of the twisted tape (mm)
Twm	mean inside wall temperature,
Tbm	mean bulk fluid temperature,

Greek symbols

μ	dynamic viscosity, $kg\ s^{-1}\ m^{-1}$

μ_t	eddy viscosity, kg s^{-1} m^{-1}
η	thermal performance factor, $(Nu/Nu_0)/(f/f_0)^{1/3}$
δ	thickness of the twisted tape (mm)
λ	twist parameter, reciprocal of tape twist ratio, Y
ε	turbulent dissipation rate, m^2 s^{-3}
ρ	density, kg m^{-3}
δ_{ij}	Kronecker delta

Subscript

| ave | average |
| 0 | plain tubes |

1. Introduction

In last decade heat transfer enhancement technologies (HTET) have been widely applied in refrigeration, automotives, process industry, nuclear reactors, solar water heaters, etc. applications. Until now, many investigations are carried out to reduce sizes, cost and energy consumption of heat exchangers. The most influential factors are heat transfer coefficients and pressure drops. HTET can offer significant economic benefits in various industrial processes.

Bergles, 1985 and Webb, 1994 have reported comprehensive reviews on techniques for heat transfer enhancement. For a single-phase heat transfer, the enhancement had been brought using roughened surfaces and other augmentation techniques, such as swirl/ vortex flow devices and modifications to duct cross-sections and surfaces. These are the passive augmentation techniques, which can increase the convective heat transfer coefficient on the tube side. Many techniques for the enhancement of heat transfer in tubes have been proposed over the years. Inside the round tubes, a wide range of inserts, such as tapered spiral inserts, wire coil, twisted tape with different geometries, rings, disks, streamlined shapes, mesh inserts, spiral brush inserts, conical-nozzles, and V-nozzles have been used   Promvonge et al. (2006, 2007).

Smithberg and Landis (1964) had estimated the tape fin effect assuming a uniform heat transfer coefficient on the tape wall, equal to that on the tube wall. Authors reported that the fin-effect increases the heat transfer but in practice, the tape-fin effect will not attain such a high value due to the poor contact between the tape and the tube.   Lopina and Bergles (1969) conducted experiments using insulated tapes to estimate the tape fin effect. Assuming zero contact resistance between tube and tape with equal and uniform heat transfer coefficients on tube and tape walls, authors predicted 8% to 17% of the heat was transferred through the tape. Date (1974) reported heat transfer rate for fully developed laminar and turbulent flow in a tube containing twisted tape. Manglik and Bergles (1993a) and (1993b) presented   pressure drop and heat transfer coefficient correlations for laminar, transition, and turbulent flow in isothennal-wall tubes with twisted- tape inserts.   Authors included the parameter tape thickness to consider   helical twisting of the streamlines.Al-Fahed et al. (1999) reported   the heat transfer and friction   in a microfin tube fitted with twisted tape inserts for three different twist/width ratios under laminar flow region. Saha et al. (2000, 2001) presented   swirl flow characteristics due to twisted tape in laminar region. The experiments were carried for large Pr number   range 205 to 518. Authors concluded that for short length twisted tapes heat transfer increased along with less pressure drop as compared to full length twisted tape.

Whereas, the concluding remarks from earlier studies on numerical and experimental work, are as follows

Rahimia et al. (2009) carried out experimental and CFD studies on heat transfer and friction factor characteristics of a tube equipped with modified twisted tape inserts. The investigations are with the classic and three modified twisted tape inserts. The authors observed that the Nusselt number and performance of the jagged insert were higher than other ones. Eiamsa-ard et al. (2009a, 2009b) carried out the numerical analysis of heat and fluid-flows through a round tube fitted with twisted tape. The author investigated the effect of tape clearance ratio on the flow, heat transfer and friction factor. Wei and Jang (2009) studied

numerically and experimentally three-dimensional gas-fluid flow and heat transfer inside tubes with longitudinal strip inserts (both with /without holes) and twisted-tape inserts twisted at three different angles. (α = 34.3º, 24.4º, 15.3 º)

Few attempts have been made to characterize and understand the structure of twisted-tape induced swirl flows. In perhaps the first and only such work for laminar flows, Manglik and Ranganathan (1997) have visualized the secondary flow patterns using smoke injection techniques. From their observations of a limited number of cases, they suggest a three-zone flow behavior: viscous-flow, no swirl regime; swirl-transition regime; and fully developed swirl-flow regime. The fully established swirl was found to consist of two dissimilar, counter-rotating, helical vortices. This latter structure had earlier been suggested by Seymour (1966) as well in another flow visualization study, but for turbulent flows.

From above studies it could be concluded that the tube tape inserts in a full length of tube provides one of the most attractive heat transfer augmentative techniques for flow inside the tube on account of its simplicity and the effectiveness. The short-length twisted tapes have been considered by many researchers due to reduction of pressure drop while augmenting the heat transfer simultaneously.

Computational simulations are also very useful for determining the local flow and thermal characteristics. However, the reported numerical investigations (Date and Singham, 1972; Date, 1974; Duplessis and Kroger, 1984, 1987; Date, 2000); P. Sivashanmugam et.al.(2008,2009,2010); Wei and Jang, 2009; S. Eiamsa-ard et.al., 2009, Rahimia et al.(2009)   have primarily focused on evaluating global thermal-hydraulic behavior (friction factor, Nusselt number and the relative heat transfer enhancement) rather than the structure of velocity and temperature fields. This is rather unfortunate, as computational techniques are ideally suited for extracting such information.

The data presented by the previous researchers indicate that, in spite of the large body of information accumulated to date, the physics of swirling flows in tubes/channels with swirlers still calls for further study. In particular, the questions are still open both of the very structure of internal swirling flows and of the interaction between organized vortex structures of different scales arising under conditions of turbulent flow in channels with different swirlers. A CFD prediction of the heat transfer and friction characteristics of the fully and partially decaying swirl flows in the turbulent flow regime has been taken up to study the structure of velocity and temperature fields. The effects of the twisted tape location on pressure drop and heat transfer characteristics due to creation of swirl in the turbulent flow within the tubes were also studied.

2. Numerical simulation

2.1 Problem Definition

The numerical simulations were carried out using the CFD code FLUENT-6.3.26 that uses the finite volume method to solve the governing equations.

Geometry was created for air flowing in an electrically-heated stainless steel tube of 22 mm diameter(D) and length(L) 90 times the diameter as in the experimental setup as shown in Figure [1]. A computational model has been created in GAMBIT-2.3.16 as shown in Figure 3.

Twisted tape inserts under the following locations of the twisted tape configurations were used:

A] Upstream condition (HLUTT) – tube with twisted tapes located in the first half of 50 diameters of the heated section.

B] Downstream condition (HLDTT) – tube with twisted tapes located in the second half of 50 diameters of the heated section.

B] Full-length condition (FLTT) – tube with twisted tapes located in the full length of the heated section.

C] The inlet condition (ILTT) – tube with twisted tapes located in the inlet before the full length of the heated section.

D] Plain Tube condition (PT) – tube without twisted tapes in the full length of the heated section.

 The parameters considered in this study are described in table 1.

Table 1.Parameters considered in the present study

Geometrical configiration	FLTT, HLUTT,HLDTT,ILTT and PT
TWIST PARAMETER λ (d/H)	0.14, 0.27 and 0.38
Re	25000 to 110000
Stainless Steel tube of Inner Dia. (D) of Pipe	22 mm (Constant)
Length of test section	1900 mm (Constant)
Heat Flux input (Electrical)	2300 and 6200 W/m2 (Constant)
Twisted tape material : Brass	22 mm (Width) x 0.8 mm (Thk)

2.2 Numerical methods

The available finite volume procedures for swirling flows and boundary layer are employed to solve the governing partial differential equations. Some simplifying assumptions are required for applying of the conventional flow equations and energy equations to model the heat transfer process in tube with twisted tape.

For turbulent, steady, and incompressible air flow with constant properties, while neglecting the natural convection and radiation, the three-dimensional equations of continuity, momentum, and energy, in the fluid region are solved. These equations are as below

Continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla .(\overline{\rho v})= 0 \qquad\qquad -----(1)$$

Momentum equation

$$\frac{\partial(\overline{\rho v})}{\partial t} + \nabla .(\overline{\rho v v}) = -\nabla p + \nabla .(\bar{\tau}) + \rho \bar{g} \qquad\qquad -----(2)$$

Energy equation

$$\frac{\partial(\rho E)}{\partial t} + \nabla .(v(\rho E + p)) = \nabla .(k_{eff}\nabla T + (\overline{\tau eff}.\bar{v})) \qquad\qquad -----(3)$$

In the Reynolds-averaged approach for turbulence modeling, the Reynolds stresses, in Equation (2) are appropriately modeled by a method that employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradients as shown below:

$$-\rho \overline{ui'}\,\overline{uj'} = \mu t(\frac{\partial ui}{\partial xj} + \frac{\partial uj}{\partial xi}) - \frac{2}{3}(\rho \kappa + \mu t \frac{\partial u\kappa}{\partial x\kappa})\delta ij \qquad\qquad -----(4)$$

An appropriate turbulence model is used to compute the turbulent viscosity term μt. The turbulent viscosity is given as:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \qquad\qquad -----(5)$$

The Second Order Upwind Scheme was used to discretize the convective term. The linkage between the velocity and pressure was computed using the SIMPLE algorithm. The standard Wall Treatment model was chosen for the near-wall modeling method.
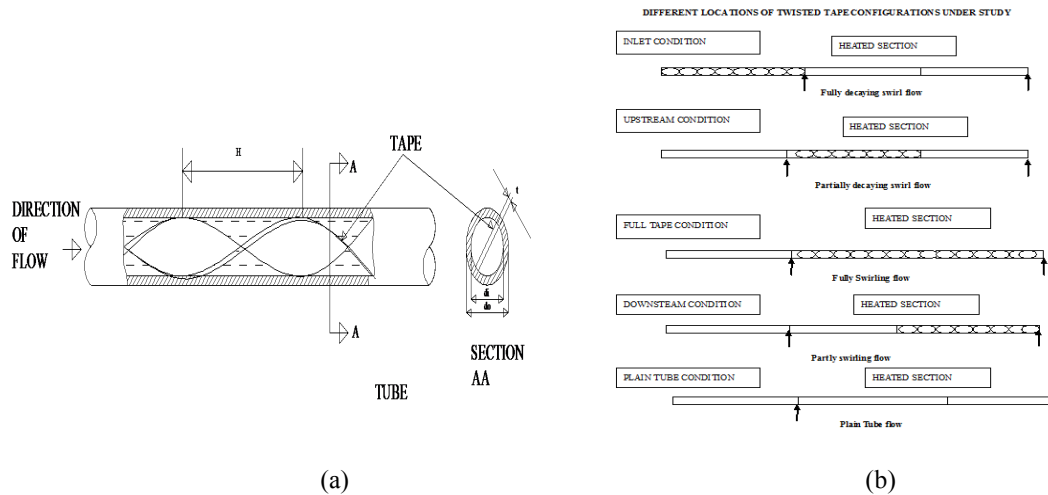
(a)   (b)

Fig. 1 Problem definition-geometrical configuration a) Physical Model of the problem defined [b] .
Different configurations under study

The Second Order Upwind Scheme was used to discretize the convective term. The linkage between the velocity and pressure was computed using the SIMPLE algorithm. The standard Wall Treatment model was chosen for the near-wall modeling method.

For validating the accuracy of numerical solutions, the grid independent test has been performed for the physical model. The grid is highly concentrated near the wall and in the vicinity of the twisted tape. Four grid systems with about 130000, 300000,660000 and 1200000 cells are adopted to calculate grid independence. We compared the friction factors for these four mesh configurations as shown in fig.2. After checking the grid independence test, the simulation grid in this study was meshed using about 6, 60,000 cells that consisted of tetrahedral grid.
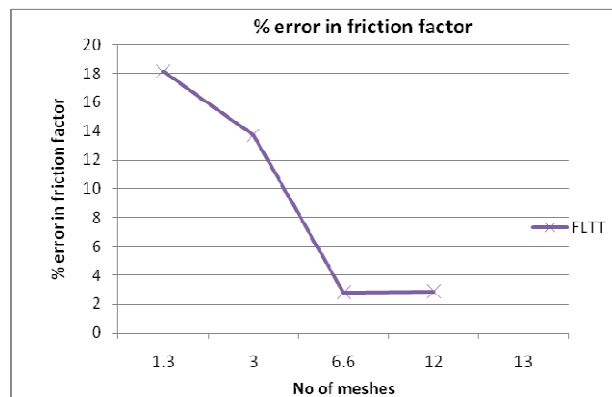

Fig.2 Grid Independence Test

Fig. 3 shows an example of the partial-meshed configuration of the round tube equipped with a twisted tape. It consists of a tube of diameter   22 mm containing twisted tape insert, test section 2000 mm and calming section of 1200 mm dimensions just like those in experimental set up with twist angle 0.14 To capture wall gradient effects, mesh has been finer toward the walls. There are a total of 6, 60,000 nodes in the domain simulation.

In addition, a convergence criterion of 10^{-6} was used for energy and 10^{-3} for the mass conservation of the

calculated parameters.

The air inlet temperature was specified as 300 K, and three assumptions were made in the model: (1) the uniform heat flux along the length of test section (2) the wall of the inlet calming section was adiabatic and (3) the physical properties of air were constant and were evaluated at the bulk mean temperature. The Velocity inlet boundary condition was adopted at the inlet and Outflow at the outlet of the domain shown in Fig. 1
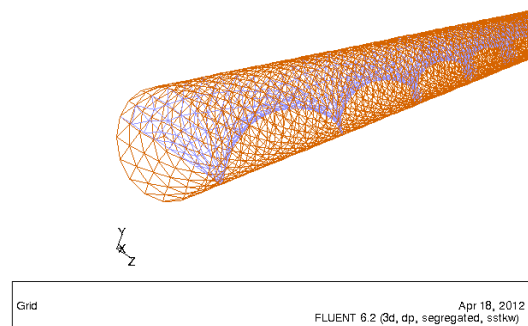

Fig. 3 Partial view of tube with twisted tape inside

3. Results and discussion

Evaluation of global thermal-hydraulic behavior (friction factor, Nusselt number and the relative heat transfer enhancement) was carried out by Yadav and Padalkar (2012) whereas the structure of swirl number, Nu number, velocity and temperature fields were discussed in this paper.

The present work is one of the very few investigations, which provide detailed data on local wall temperature profiles and local heat transfer coefficients along with swirl number profiles due to the effect of the location of a partially extending twisted tape on pressure drop and heat transfer. A careful study of the previous literature on tape generated, swirl flow revealed that only two other investigations those of Seymour (1963) and Klepper (1971) - were similar to the present work, in the use of partially extending tapes. Seymour, however, investigated the effect of shortening the tape in a very short test section, 28 diameters in length. It is expected therefore that his results would contain a significant effect of entrance conditions.

Although Klepper provided data on a combination of non-decaying and decaying swirl using partially extending tapes, his use of a 20 diameter smooth section upstream of the tape alters the thermal, entry conditions in the tape section significantly, when viewed over the wide range of Reynolds numbers he employed.

Thus, inspite of well defined and repeatable boundary conditions used in the present work, there are no other results with which the results of the present work can be compared. Results will therefore be presented, discussed and where possible compared with data obtained on full length tapes or with smooth tubes.

3.1. Validation of set up

The CFD simulation result of the plain tube (PT) without a twisted tape insert has been validated with the experimental data as shown in fig.4 (a) and 4(b). The Dittus-Boelter equation for the heat transfer and the Blasius equation for the friction factor are the correlations used for the comparison. These results are within ±15% deviation for the heat transfer (Nu) and ±6% for the friction factor (f). Similarly, the CFD results for the plain tube are compared with analytical correlations. The CFD results are within ±9% deviation for the

heat transfer (Nu) and ±6% for the friction factor (f) with slightly higher deviation of ±17% for Re higher than 75000.
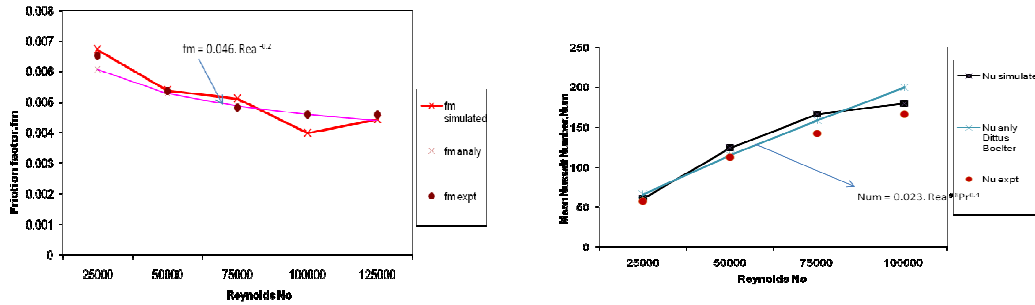

(a) Friction factor verses Reynolds Number   (b) Nu verses Remolds Number
Fig. 4 Comparison of CFD results with experimental data and correlations

3.2 Velocity and temperature profiles analysis

The dimensionless velocity and temperature profiles of turbulent flow in a round tube are depicted in Fig. 5 (a-d) under the Reynolds number of 25000 with twist ratio of 0.14. Where, the dimensionless temperature is defined as $T^* = (T_w - T)/(T_w - T_m)$. As shown in the Figs. 5, the longitudinal vortices generated along the different part of the length by the insertion of twisted tapes lead to the separation of the velocity boundary layer and the temperature boundary layer. Compared with the plain tube, the velocity profile changed fundamentally while due to these changes the temperature profile has changed extremely. Thus the thermal diffusivity is enhanced greatly due to the momentum diffusivity change. And this is the use of core-flow heat transfer enhancement. So, the insertion of twisted tapes in the parts of test section enhances the heat transfer greatly, but does not increase the friction resistance very much due to absence of twisted tape partially.

(a)   (b)
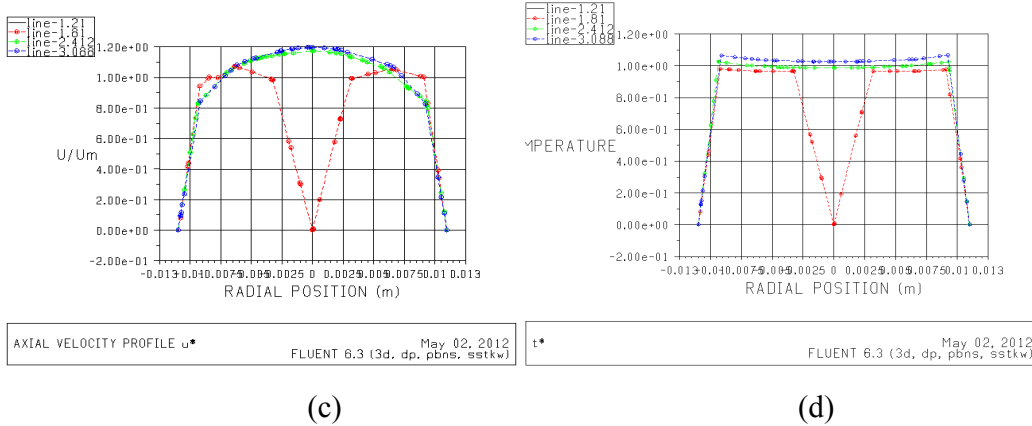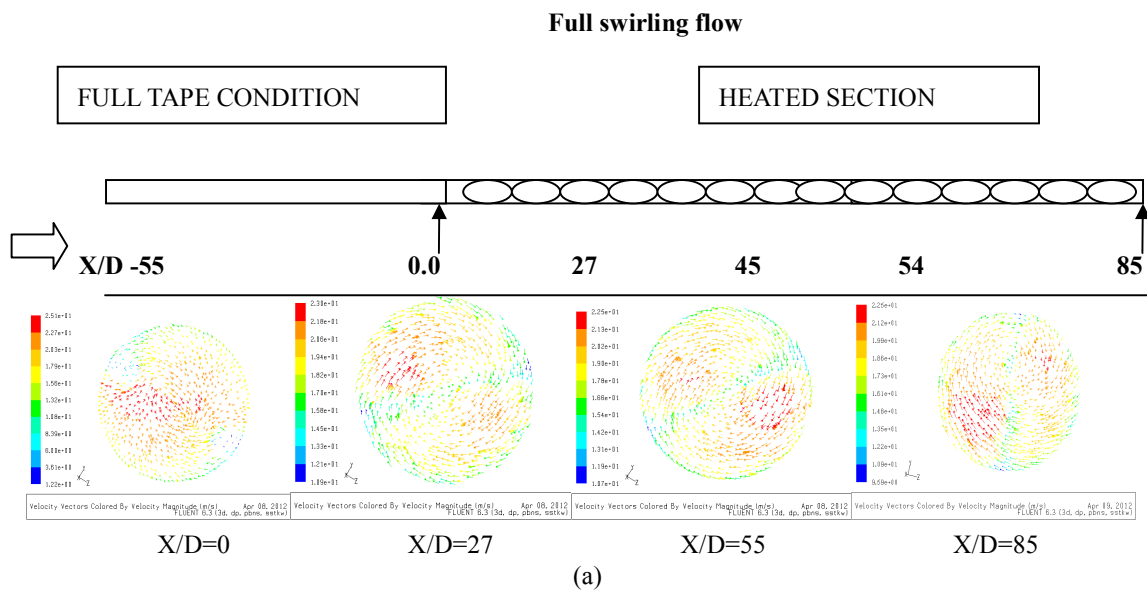
(c)

(d)
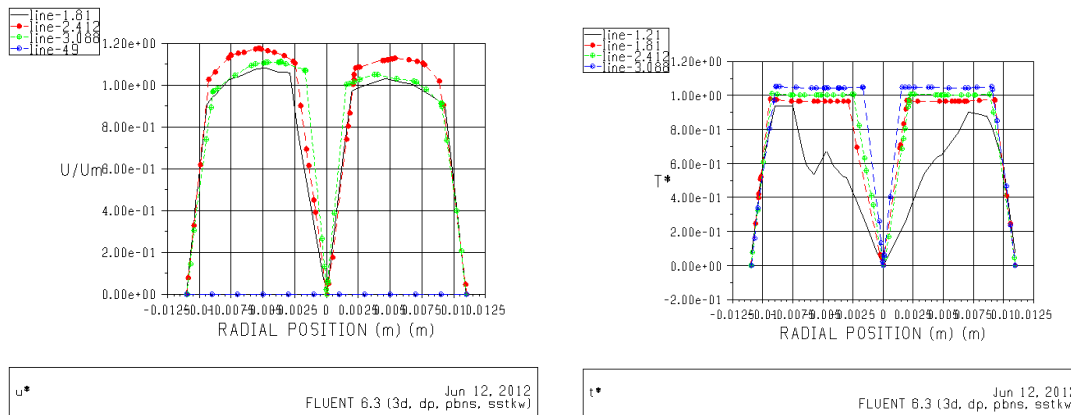
Fig. 5 (a) Dimensionless Axial Velovity vs Radial Position for FLTT (b) Dimensionless Temperature vs Radial Position for FLTT (c) Dimensionless Axial Velovity vs Radial Position for HLUTT (d) Dimensionless Temperature vs Radial Position for HLUTT

3.3. Velocity vector plots

Fig. 6 shows the vector plots for the FLTT and the PT conditions at four different cross-sections of the tube along the length respectively. The effect of the twisted tape on the velocity magnitude in the creation of tangential and radial velocity components is clearly seen at different locations as shown in Fig. 8.
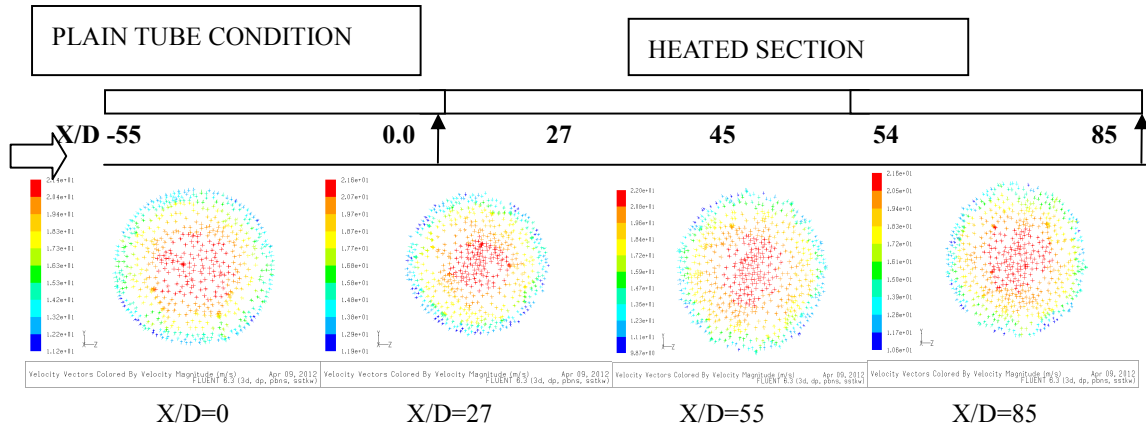
The velocity vectors in a vertical slice that go through a tube equipped with the FLTT twisted tape insert have been compared with those of the HLUTT and the PT case. The figure 8 shows that due to the insertion of the insert in the tube, swirl flow results, which slides the fluid upon the inner tube wall.

The tangential velocity is almost zero for the plain tube at all the Reynolds numbers. However, it is seen that this velocity component increases when any of the above mentioned inserts are placed inside the tube. From the values, it is seen that the inserts are able to disrupt the boundary layer and provide more contact between the fluid and the tube wall. The predicted values of velocity for the HLUTT, HLDTT and ILTT inserts are lower than those obtained for the FLTT insert.
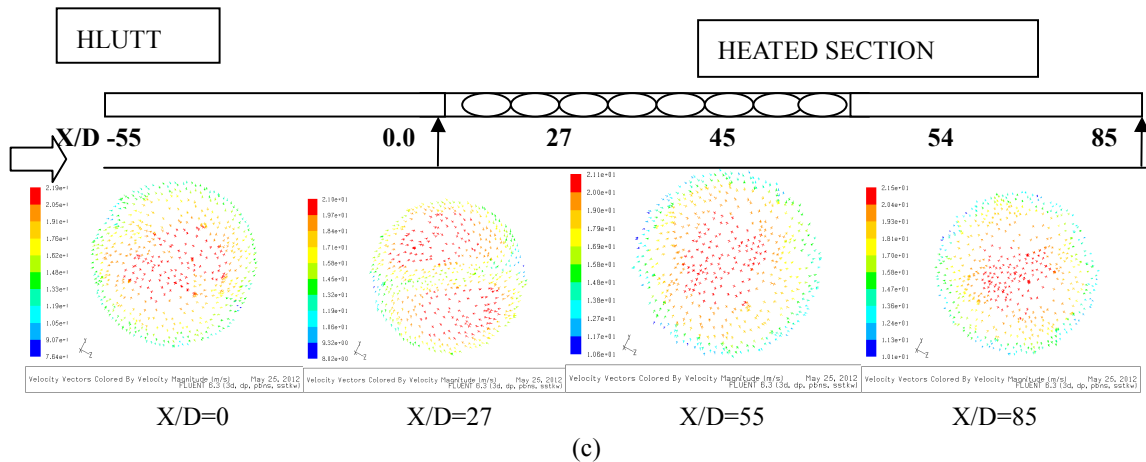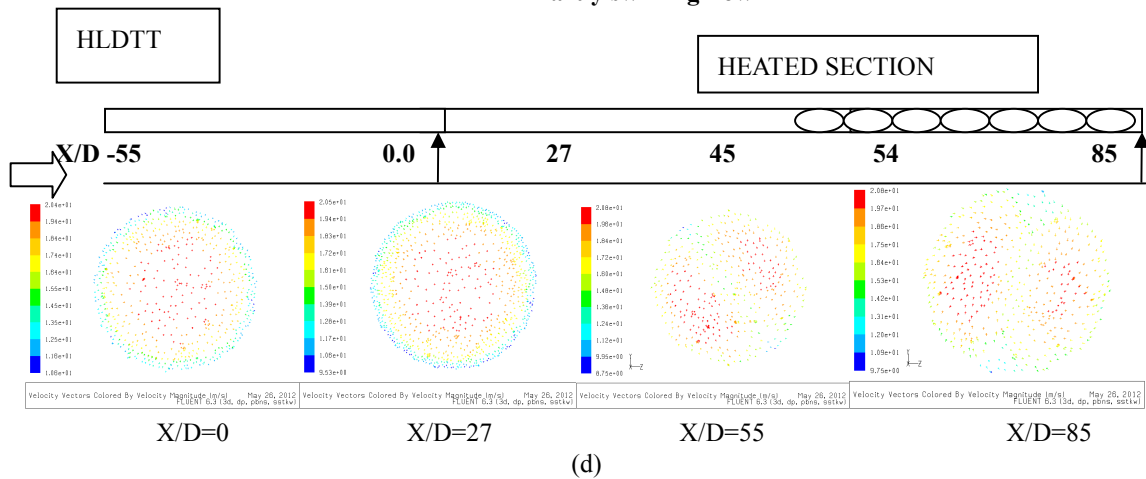
Full swirling flow

FULL TAPE CONDITION   HEATED SECTION

X/D=0   X/D=27   X/D=55   X/D=85

(a)

Smooth flow

X/D=0   X/D=27   X/D=55   X/D=85

(b)

Partially decaying swirling flow

X/D=0   X/D=27   X/D=55   X/D=85

(c)

Partly swirling flow

X/D=0   X/D=27   X/D=55   X/D=85

(d)

Fully decaying swirl flow

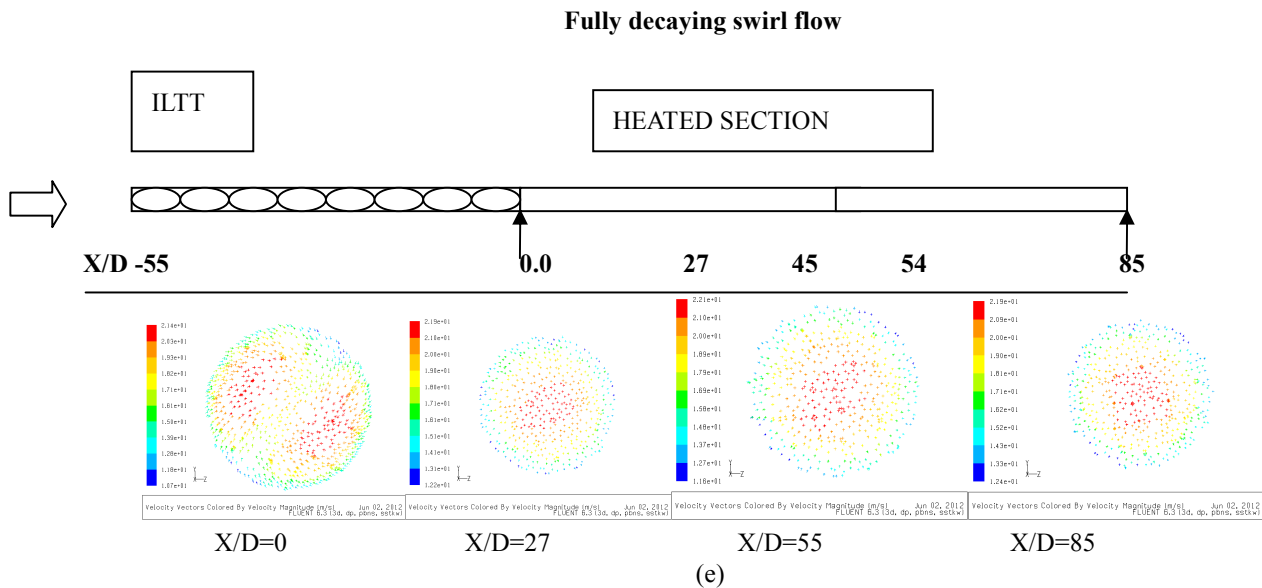X/D=0   X/D=27   X/D=55   X/D=85

(e)

Fig. 6   Vector Plots across tube cross-section along the axial length for (a)FLTT (b)PT (c) HLUTT(d)HLDTT and (e)ILTT with twist ratio 0.14 and Re = 25000, q = 2300 W/ m2

3.5 Temperature profiles analysis

(a)   Plain tube Data

It is observed from the smooth tube temperature profile that, the maximum wall temperature in the test section, Twx,max and the maximum temperature difference between, the tube wall and the fluid, (Twx – Tbx)max   both occur at X/D = 86. Twx,max   varies from 48°C to 215°C, (Twx -Tbx)$_{max}$ varies from 10°C to 56°C, the wall temperature profile shows, fully developed characteristics at roughly 17 diameters from entrance, At any location wall temperature decreases with Reynolds number and increases with heat flux.

(b)   Twisted Tape Data

Figs. 7 shows wall temperature profiles, for all the cases investigated. The data presented reveal the following trend.

(i) Effect of Reynolds number on Twx : At any axial location the local wall temperature decreases with increasing Reynolds number. This is quite expected since heat transfer coefficients increase with Reynolds number bringing down both the wall to fluid temperature difference and the absolute wall temperature.

(ii)   Effect of Heat Flux on Twx:

In all the cases, an increase in heat flux results in an increase in the local wall temperature.   For the upstream condition at λ = 0.14, an increase in heat flux, caused, in addition to an increase in the local wall temperature, a shift in the location of (Twx-Tbx)min from X/D = 36 to X/D = 45.

(iii)   Effect of Twist Parameter on Twx:

It is observed that the twist parameter λ has a significant effect on both the magnitude of Twx and its variation along the test section.   Effect of λ on Twx will be discussed separately for the upstream and downstream conditions.

Upstream condition

A dip in wall temperature is observed at the end of tape section for all values of λ. In the swirl decay section following the tape, the local wall temperature increases with an increase in λ.

Downstream Condition

The maximum wall temperature, Twx, max is located at X/D == 86, for all values of λ. A steep temperature drop is noticed rear the entrance to tape section, for λ = 0.38 and 0.27, except for λ = 0.14.

For all values of λ, the tape section records the lowest wall temperature and the smooth section, the highest.

Inlet condition

The local wall temperature distribution differs slightly from the smooth tube values due to the presence of the decaying swirl. As a result of the decaying swirl, the temperature difference (Twx - Tbx) increases throughout the test section. An increase in λ results in a decrease in the local wall temperature at all locations.

(iv) Effect of Tape Location on Twx:

It is observed that the local wall temperatures are least for downstream condition. The effect of tape location on the maximum test section temperature, Twx,max is given below. Values are given as a percentage decrease from corresponding smooth tube values,

Tape location	Min	Max
HLDTT	5.2 %	19 %
HLUTT	0.0 %	6 %
ILTT	0.0%	1%

It is seen that downstream location of tape is most effective in bringing down the maximum test section wall temperature.
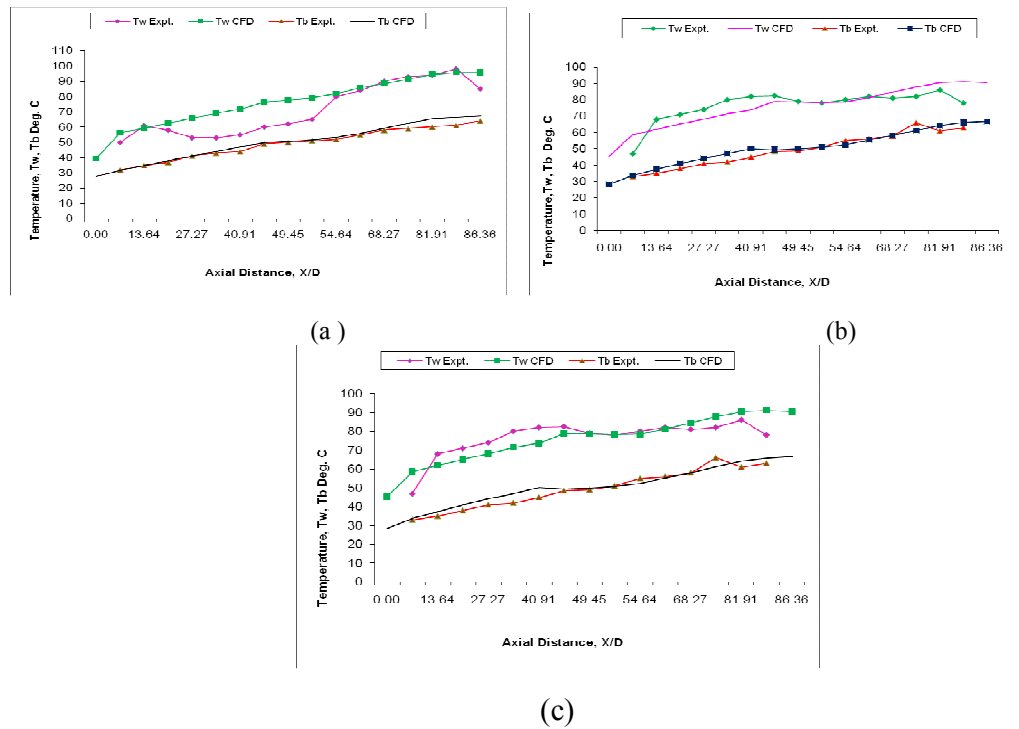

(a)

(b)

(c)

Fig. 7 Experimental and CFD Wall Temperature & Surface Temperature for q = 2300W/m2, Re = 25000, and λ =0.14for (a) upstream location (b)   Downstream location   and (c)   inlet before test secrion location for q = 2300W/m2, Re = 25000, and λ =0.14

3.6 Local Nusselt Number analysis

An examination of the local Nusselt number profiles for the upstream and downstream conditions as shown in Fig 8 shows that the Nusselt number attains local peaks, a characteristic which was noticed by Klepper (1971) also in his experiments on partially extending tapes. This unusual behavior of the local Nusselt number has not been reported by any other investigator. That the occurrence of these peaks is real appears beyond doubt when it is observed that they occur at all Re, λ and q and for both the

downstream and upstream tape locations.

While, the characteristic of local peaks observed in the investigation can be used in avoiding the local hot spots in heat exchanger application in such diverse areas as the cooling of an overheated Rocket nozzle throat, prevention of burnout in space and earth power plants, reduction of wall temperature in circulating fuel reactors and in the heat exchange equipment used in process industries. In most of the above applications temperatures critical to material life are likely to be reached, and as such any reduction in wall temperature would imply an improvement in performance.
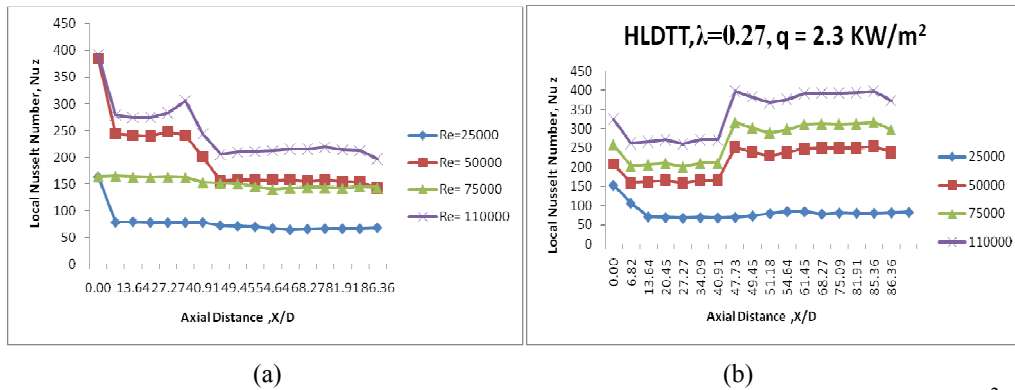

(a)   (b)

Fig. 8 Local Nusselt Number along axial distance for for λ =0.14, q= 2300W/m^2 for (a) HLUTT (b)HLDTT

4. Conclusion

The important issue in the present work can be expressed as the understanding of heat transfer and temperature analysis for fully, partially decaying and partly swirl flow using the FLTT, HLUTT, HLDTT and ILTT twisted tape insert.

It was found that thermal performance and local peak in heat transfer could be increased by using a combination of inserts with different geometries in the plain tubes while reducing the pressure drop. While, the characteristic of local peaks observed in the investigation can be used in avoiding the local hot spots in heat exchanger application.

Since the Nusselt number peaks were observed for both the downstream and upstream tape locations, the choice of tape location would be governed by the actual location of hot spots.

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