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Analysis of Heat Transfer in Pipe with Twisted Tape Inserts

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Abstract- Twisted tape is widely used heat transfer enhancement technique. Present paper argues the underlying physical phenomenon of heat transfer and fluid flow through a pipe with twisted tape inserts. Effect of twist ratio for $2.0 \leq p/d \leq 5.0$ and Reynolds number for $800 \leq Re \leq 2000$ (where p/d is the pitch ratio and Re is Reynolds number) on Nusselt number and friction factor have been numerically obtained. It is noticed that decrease in twist ratio promotes radial convection while increase in Reynolds number promotes axial convection. This is evident through understanding the variation of velocities and temperature across a particular section; especially near the wall.

***Keywords*:** Reynolds number, twisted tape, Nusselt number, Conduction and convection heat transfer, Fluid flow, Heat transfer augmentation, Axial Convection, Radial Convection

Nomenclature

Cp	Specific heat capacity, J/Kg.K
d	Internal diameter of the tube, m
E	Energy per unit mass, J/kg
p	Pitch of twisted tape for 180°rotation
h	Heat transfer coefficient, W/m2°C
k	Thermal conductivity, W/m K
L	Length of the tube, m
Nu	Nusselt number, Dimensionless
Re	Reynolds number, Dimensionless
v	Velocity, m/s
y	Distance measured from tube wall towards twisted tape, m
Δp	Pressure difference, Pa
μ	Dynamic Viscosity, kg/ m. s
t	Thickness of twisted tape, m
σ_T	Turbulence stress, Pa
ρ	Density, kg/m3
u	Component of velocity along the direction of fluid flow, m/s
v	Component of velocity along the direction parallel to surface of twisted tape, m/s
w	Component of velocity along the direction perpendicular to surface of twisted tape, m/s
T	Temperature of fluid, K
Ti	Temperature of fluid at inlet, K
Ui	Velocity of fluid at inlet, m/s
p/d	Twist Ratio

1. Introduction

Twisted tape is one of the passive heat transfer enhancing technology which has been extensively studied due to steady performance results and easy installation. Optimization of thermohydraulic performance of tubes fitted with twisted tapes has gained increasing attention.

Manglik and Bergles (1993) experimentally studied effect of twisted tape in laminar flow. They concluded that the main reasons for heat transfer augmentation by twisted tape inserts are partitioning of tube flow resulting in higher flow velocities, reduction in hydraulic diameter increasing the heat transfer coefficient, helically twisting fluid motion increasing flow path and secondary fluid motion improving convective heat transfer.

Saha et al. (1989) analyzed the heat transfer and pressure drop characteristics of laminar flow in a circular tube fitted with regularly spaced twisted tape inserts connected with rod. They experimentally found out that the pressure drop of the tube fitted with twisted tape inserts connected with rod is 40%

smaller than that of tube fitted with a continuous twisted tape, and the former has better thermo hydraulic performance. They further studied the effects of width of tape inserts and diameter of connecting rod on heat transfer and pressure drop characteristics. The conclusion was that narrow width of tape elements led to a worse thermo-hydraulic performance, while a thinner connecting rod resulted in better one.

Eiamsa-ard et al. (2006) extended the study to the heat transfer behaviors in a circular tube fitted with regularly spaced twisted tape inserts in laminar and turbulent flows. They argued that the heat transfer coefficient and friction factor were both significantly reduced as compared with those of tube fitted with continuous twisted tape.

Similar type of efforts were carried out by Jaisankar et al. (2009), Ayub and Al Fayed (1993), Rahimia et al. (2009), Ray and Date (2001), Zimparov (2001) and others wherein the amount of augmentation in heat transfer and friction factor are studied for different types of twisted tape inserts with their various arrangements and different types of fluid ducts.

Although tremendous efforts have been taken to enhance heat transfer by using twisted tapes, underlying physics for the enhancement of heat transfer has been rarely discussed in the literature.

The present paper proposes a physical reason for enhancement of heat transfer as a function of twist ratio and Reynolds number.

2. Physical Model

The geometry of standard twisted tape is depicted in Fig. 1. Twisted tape with thickness (t) of 1 mm is fitted in full length of tube. The diameter (d) and length (L) of tube are 22 mm and 2200 mm, respectively. The 180^0 twist pitch (p) considered in present study are 44 mm, 66 mm, 88 mm and 110 mm and thus the relative twist ratios (p/d) are 2, 3, 4 and 5. The Reynolds number, Nusselt number and friction factor are defined as follows:

$$Re = \frac{\rho v d}{\mu} \tag{1}$$

$$Nu = \frac{hd}{k} \tag{2}$$

$$\Delta p = \frac{f L \rho v^2}{2d} \tag{3}$$

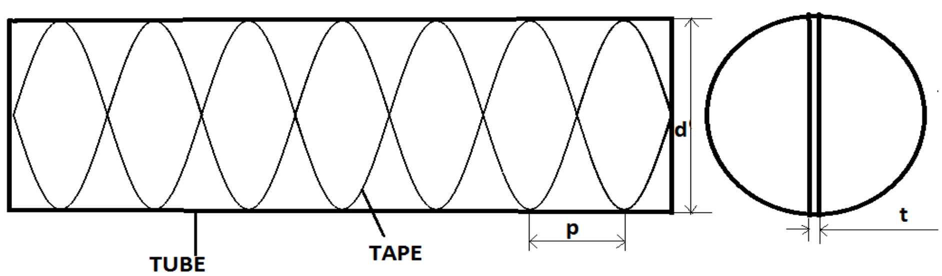

Fig. 1. Twisted Tape fitted inside tube.

Water is selected as working fluid which is assumed to be incompressible. The natural convection has been neglected and the thermo-physical properties of fluid are assumed to be temperature independent. The dynamic viscosity (μ), thermal conductivity (k), density (ρ) and specific heat at constant pressure (C_p) of water are given as $\mu = 0.001003$ kgm^{-1} s^{-1}, k = 0.6Wm^{-1} K^{-1}, ρ = 998.2 kg.m^{-3} and C_p = 4182J.kg^{1}k^{-1}. The Reynolds numbers considered in present study are 800, 1200, 1600 and 2000.

3. Mathematical Model and Numerical Method

The problem under consideration is three dimensional, laminar with steady state conditions. The heat conduction in twisted tape is neglected.

3. 1. Governing Equations and Boundary Conditions

Equations of continuity, energy and momentum are:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho \vec{u}) = 0 \tag{4}$$

$$\rho(\frac{\partial \vec{u}}{\partial t} + \vec{u}.\nabla \vec{u}) = -\nabla p + \mu \nabla^2 \vec{u} + \nabla \sigma_T \tag{5}$$

$$\frac{\partial T}{\partial t} + \nabla.\vec{u}T = \frac{k}{\rho C_P} \nabla^2 T \tag{6}$$

A constant wall temperature is imposed at tube wall. At the inlet, velocity and temperature are specified. At outlet, a pressure-outlet condition is used. On the surfaces of the tube wall and twisted tape, no slip conditions are applied.

3. 2. Numerical Method

The commercial software, Fluent 13.0 is used as the CFD tool for this work. The time independent incompressible Navier Stokes equations and K-ω turbulence model is discretized using finite volume technique. A second order upwind scheme is applied for convective and diffusive terms. To evaluate pressure field, the pressure-velocity coupling algorithm SIMPLE (Semi Implicit Method for Pressure Linked Equations) is selected. The turbulence intensity is kept at 10% at inlet. Convergence criteria of 10^{-5} for continuity and 10^{-6} for energy are set.

4. Results and discussions

To make sure that the results obtained do not vary with the cell size, grid independence test has been carried out for physical model. Three grid systems with 800000, 1000000, 1200000 and 1400000 cells are used to calculate Nusselt number and friction factor for baseline case with twist ratio 5 and Reynolds number 800. As shown in Fig. 2, the difference between calculated values of 1200000 and 1400000 grid system is very small. Hence, grid system with 1200000 cells is used for further calculations. To validate accuracy of present results, they are compared with those obtained by Manglik and Bergeles (1993) as shown in Fig 2.

In plane circular tube, the motion of fluid particles in laminar regime is in the longitudinal direction only and heat transfer in transverse direction occurs due to conduction only. In case of tube inserted with twisted tape, along with longitudinal motion there is motion of fluid particles in transverse direction due to the helically rotating fluid flow.

Table. 1. Grid Independence Study.

Grid Size	Nu	% difference
800000	20.01	
1000000	22.50	12.44
1200000	24.76	10.04

1400000	25.01	1.030

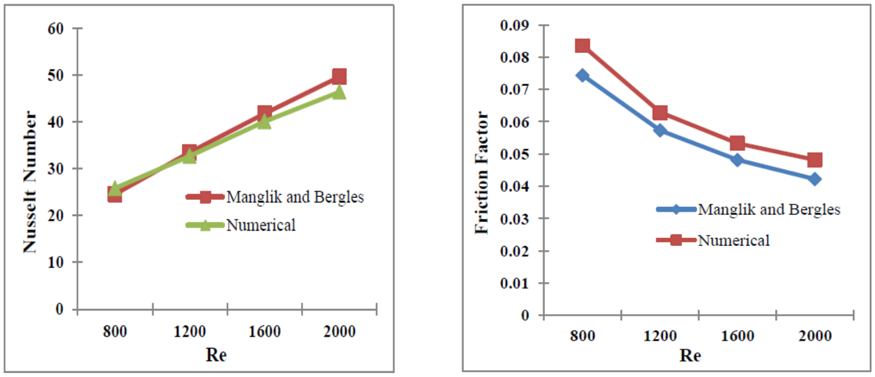

Fig. 2. Validation of numerically predicted Nusselt number and friction factor with those obtained by Manglik and Bergles for p/d=5.

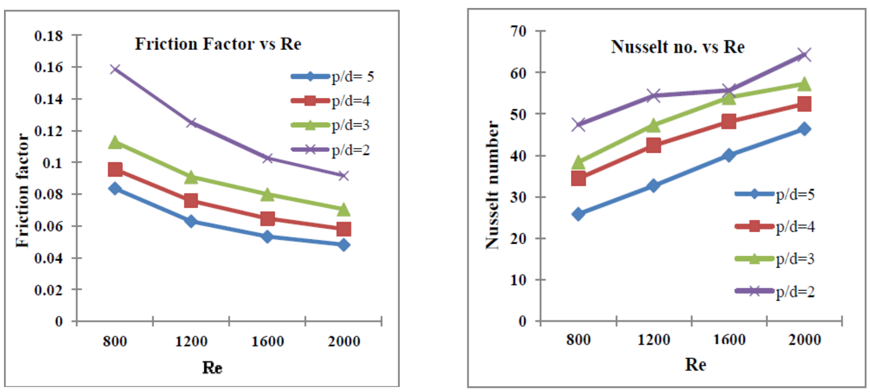

Fig. 3. a) Variation of Friction Factor and b) Nusselt number with Reynolds number.

This transverse motion is called the secondary motion of fluid. The secondary fluid motion provides heat transfer by convection mode along with conduction mode in transverse direction. This boosts heat transfer in tube fitted with twisted tape. The quantification of intensity of the secondary flow occurring due to varying flow conditions (Re and twist ratio) will be important to understand physics behind heat transfer enhancement.

The overall heat transfer occurring from tube to fluid is due to combined effect of conduction and convection. Tube flow can be divided into two parts- boundary flow and core flow. The boundary flow is a fluid region near the wall, beyond which, in the tube, the core flow is defined. If the secondary flow created due to flow conditions (Twist ratio and Reynolds number) disturbs only the core flow, it will promote uniform temperature in core region and thus will enhance conduction heat transfer from wall to fluid. On the contrary if secondary flow disturbs boundary flow, convection heat transfer will dominate. Thus it is important to analyze velocity profile and temperature profile near wall and at core to understand effect of secondary flow on heat transfer at various flow conditions.

4. 1. Effect of Twist Ratio on Flow and Heat Transfer

In order to understand the effect of change in pitch of twisted tape on the flow physics, results of Reynolds number 800 and twist ratio 2, 3, 4 and 5 are considered. A cross section of tube is considered at half of the tube length. Contours of u-velocity, v-velocity, w-velocity and temperature at this plane are plotted. Also normalized u-velocity profile, v-velocity profile, w-velocity profile and temperature profile are plotted near to the wall in order to capture boundary phenomenon. Normalization is done with reference to inlet velocity and temperature.

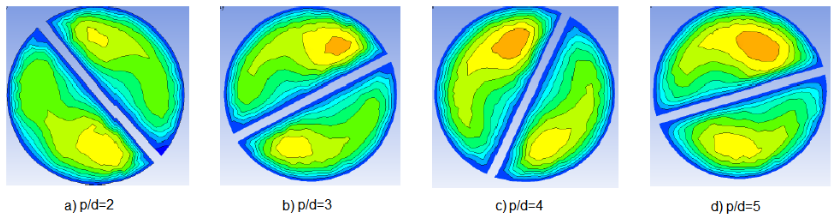

Fig. 4. U-velocity contours for varying p/d at Re=800.

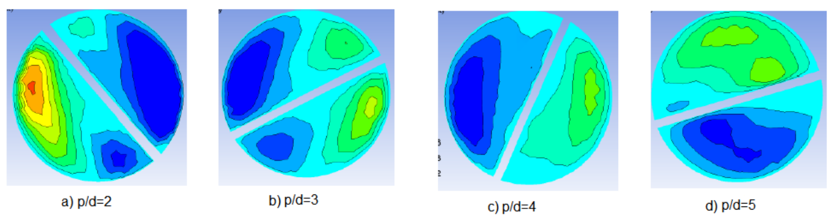

Fig. 5. V-velocity contours for varying p/d ratio at Re=800.

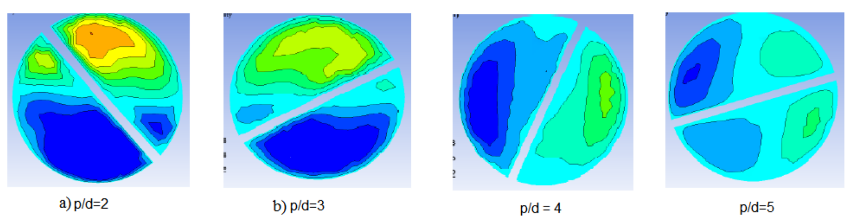

Fig. 6. W-velocity contours for varying p/d ratio at Re=800.

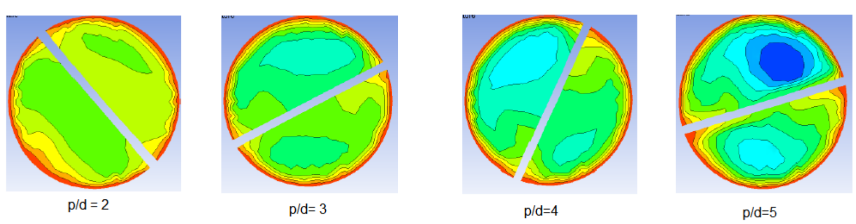

Fig. 7. Temperature contours for varying p/d at Re=800.

From the u-velocity plots near the wall in Fig 8, it is evident that there is no significant variation in magnitude of u-velocity in boundary region with variation in twist ratio, which denotes constant Reynolds number. But if we look at the contours for u-velocity in Fig 4, it is seen that magnitude of u-velocity in core shows decreasing trend along with decrease in twist ratio. This difference can be related to bulk temperature of fluid in core flow.

In case of twist ratio 2, because of more disturbed flow at the core, the temperature is uniformly distributed. For twist ratio 5, the temperature at the core is minimum, as evident from the temperature contours in Fig 7. Hence we conclude that uniformity of temperature in core flow in case of twist ratio 2 gives better heat transfer results.

From w-velocity plots near the wall (Fig 8), it can be observed that increase in w-velocity is maximum in case of twist ratio 2. As the Reynolds number is the same in all cases, the only reason of increase in transverse directional velocity i.e. the w-velocity, is the centrifugal force imposed on fluid particles due to helically rotating motion of the fluid.

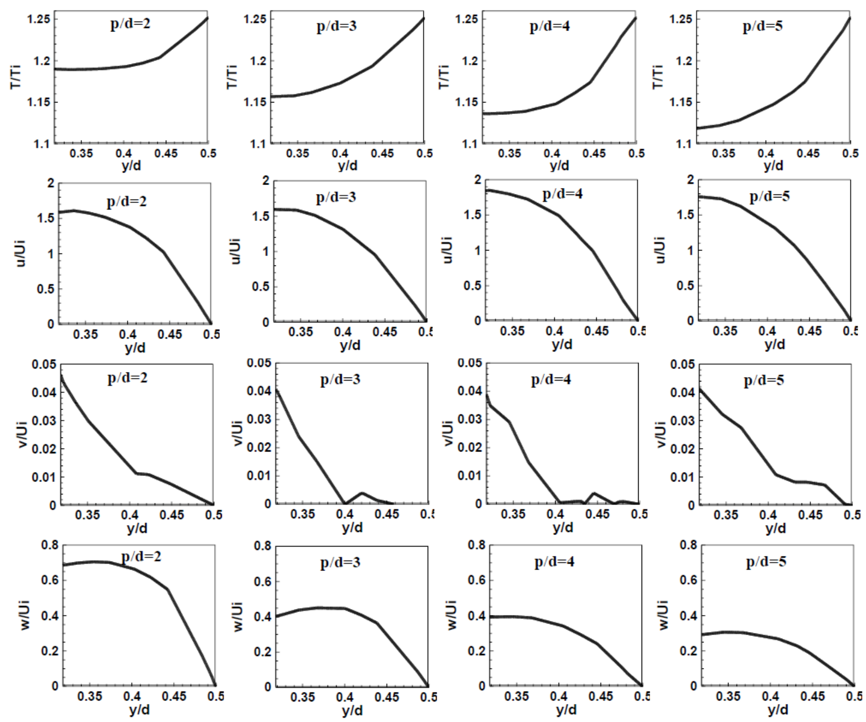

Fig. 8. Effect of variation of twist ratio on u-velocity, v-velocity, w-velocity and temperature near the wall.

It is obvious that the twist ratio 2 will provide more rotation to the flow of fluid and cause high amount of centrifugal force on the fluid particles. There is subsequent decrease in the transverse velocity with increasing twist ratio, which is logical because there is less amount of secondary rotating flow. The variation in w-velocity also provides the explanation for the variation in temperature. With increase in transverse velocity, more and more particles move towards the wall, which disturbs the thermal boundary layer. The thickness of the thermal boundary layer decides the amount of the diffusion heat transfer from wall to fluid. At lower twist ratio, the thickness of thermal boundary layer is small, causing a very small amount of diffusion heat transfer. Therefore, the overall heat transfer is mainly because of the convective currents generated by the transverse velocity.

In case of v-velocity, we do not find any significant variation near the wall, as observed from normalized v-velocity plots. The major changes in v-velocity are occurring near the tape that contribute to the flow variation at the core and promote heat transfer enhancement.

4. 2. Effect of Reynolds Number on Flow and Heat Transfer

In order to understand the effect of change in Reynolds number on the flow physics, results of twist ratio 4 and Reynolds number 800, 1200, 1600 and 2000 are considered. As Reynolds number is being varied in this case, the u-velocity will have a major effect on heat transfer. Similar contours as discussed in Section 4.1 are also developed as a function of Reynolds number. These are not shared here due to space limitations.

It is evident from these contours that u-velocity increases with increase in Reynolds number along the centerline. The decrease in the thickness of the u-velocity boundary layer indicates that there is decrease

in thermal boundary layer at the tube wall. This clearly shows that there is decrease in diffusion heat transfer and the increase in overall heat transfer is mainly because of the convective currents generated as an effect of increase in longitudinal velocity. It is further noticed that the core of flow i.e. away from tube wall and tape wall, the fluid motion in longitudinal direction is fastest in case of Reynolds number 2000 and least for Reynolds number 800. This difference can be related to bulk temperature of fluid in core flow. In case of Re = 2000, the motion of fluid is more turbulent resulting in generation of secondary flow. This increases the heat transfer. But owing to the lesser time it gets for developing uniformity, the temperature at the core is less which is evident from temperature plots (Fig. 9).
The transverse velocity contours indicate that motion of fluid in transverse direction is not too pronounced to create any significant effect on heat transfer. In case of v-velocity, we do not find any significant variation near the wall, as observed from normalized v-velocity plots shown in Fig. 9. However, the high v-velocities at the core impart transverse fluid motion promoting better mixing and convection.

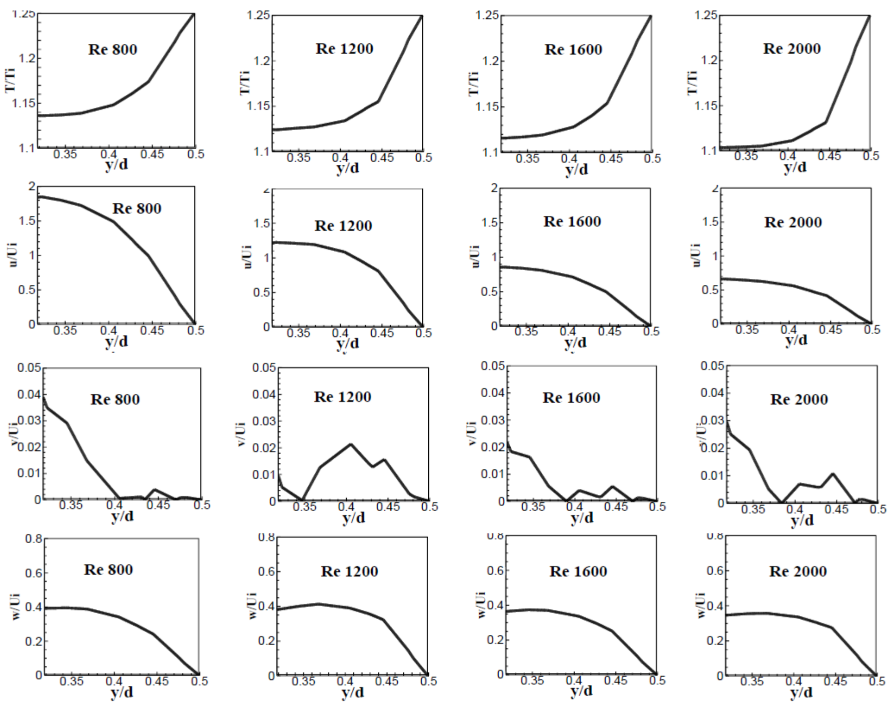

Fig. 9. Effect of varying Reynolds number on u-velocity, v-velocity, w-velocity and temperature near the wall.

5. Conclusions

Effect of variation of twist ratio and Reynolds number on heat transfer and flow characteristics using twisted tape inserts is studied. The heat transfer increases with decrease in twist ratio and increase in Reynolds number.

The heat transfer increases with decrease in twist ratio due to increase in radial convection. The lower twist ratio enhances the heat transfer due to increase in transverse fluid motion in core flow caused by increase in degree of helically rotating flow path.

The heat transfer increases with increase in Reynolds number due to increase in axial convection. Increase in Reynolds number increases the heat transfer due to disturbance in boundary layer causing increased convection heat transfer from wall to fluid.

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