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CFD Analysis of Fluid Flowing Through a Heat Exchanger Tube Having a Twisted Tape with a Centrally Placed Semi-Circular Groove

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Abstract: ***Heat exchangers are very much essential industrial as well as domestic equipment used every day. Overall performance of any machines and many systems as a whole depends on the performance of a heat exchanger. Number of researches has been done to develop devices or means to enhance the performance of a heat exchanger. The most successful development of a component for the most efficient enhancement of heat exchanger performance is development of twisted tape inserts for heat exchanger tubes. It increases flow path of the fluid inside the tube so that more and more and more heat can be transferred from the tube. At the same time due to this twisted tape insert turbulence of the flow inside the tube can be increased. This increase in turbulence will in other way decrease the overall efficiency of the heat exchanger. So optimum design of tube insert is very essential for having maximum efficiency. In the present work few design modifications have been adopted over a previously designed tube insert and their thermal performance have been analysed using a CFD software named ANSYS Fluent.***

Keywords: Tube insert, twisted tape, CFD, ANSYS Fluent, Reynolds's number, Nusselt Number

1. Introduction

Heat exchanger is a device that facilitates the exchange of heat between two fluids that are at different temperatures. Heat exchangers are used for both the purpose to remove heat from a fluid and to provide heat to a fluid. Common examples of heat exchanger are boiler, condenser, super heater, economizer, automobile radiators. Heat exchangers are commonly used in practice in a wide range of applications, from heating and air conditioning systems in a household, to chemical processing and power production in large plants.

To predict the performance of a heat exchanger, it has to be relate to the governing parameters such as heat transfer area, overall heat transfer coefficient, mean temperature difference. Assume there in no heat transfer to the surrounding and negligible kinetic and potential energy then by energy balance we can find a relation. Block diagram of heat exchanger is shown in fig 1 and parameters are indicated

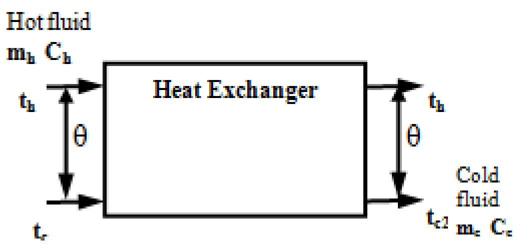


Figure 1: Heat Exchanger

m_h - mass of hot fluid entering(kg)
m_c - mass of cold fluid entering(kg)
C_h - Sp. heat of hot fluid entering (KJ/kg K)
C_c – Sp. heat of cold fluid entering (KJ/kg K)
t_{h1}- temperature of hot fluid entering(K)
t_{c1} - temperature of cold fluid entering(K)
t_{h2} - temperature of hot fluid exits(K)
t_{c2} - temperature of cold fluid exits (K)

Heat rejected by hot fluid Qh = mh Ch (th1 - th2) (1)
Heat absorbed by cold fluid Qh = mc Cc (tc2 - tc1) (2)
Heat exchange by two fluids Q = UA θm (3)

Where
U- Overall heat transfer coefficient
Effective heat transfer area
θm- appropriate mean temperature difference across heat exchanger

As there is variation in temperature difference of hot and cold fluids point to point, so that by the concept of mean temperature difference the term ϴm has introduced which is appropriate mean temperature difference across heat exchanger or known as log mean temperature difference.
For parallel flow log mean temperature difference is given by

$$\theta m = \frac{\theta 2 - \theta 1}{In(\theta 2/\theta 1)} \quad (4)$$

For counter flow log mean temperature difference is given by

$$\theta m = \frac{\theta 1 - \theta 2}{In(\theta 1/\theta 2)} \quad (5)$$

Where
θ1 - th1- tc1
θ2 - th2- tc2

Heat transfer coefficient has important role in heat exchanger so that it is essential to derive an expression for it. Heat transfer through the plane wall is shown in fig 2 and overall heat transfer coefficient is given by

$$U = \frac{1}{\frac{1}{h_i} + \frac{L}{k} + \frac{1}{h_o}} \quad (6)$$

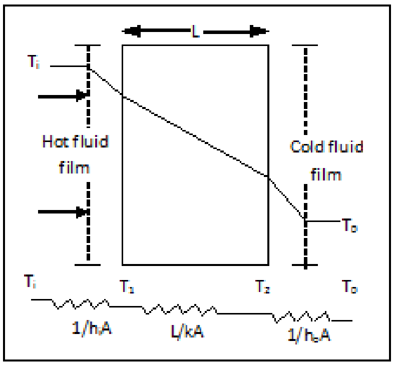


Figure 2: Overall heat transfer Coefficient of two fluids separated by a plain wall

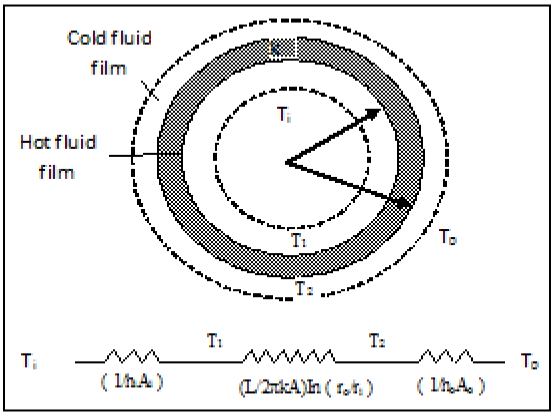


Figure 3: Overall heat transfer Coefficient of two fluids separated by a plain wall

Where, h_i and h_o are the heat transfer coefficient at inlet and outlet respectively, L is the thickness of wall and k is thermal conductivity of wall material [22], [23].

If two fluids flowing are separated by a tube wall then overall heat transfer coefficient is given by

For inner surface of tube

$$U_i = \frac{1}{1/h_i + r_i/k \, In\left(\frac{r_o}{r_i}\right) + \left(r_i/r_o\right) 1/h_o} \qquad (7)$$

For outer surface of tube

$$U_0 = \frac{1}{1/h_0 + r_0/k \, In\left(\frac{r_o}{r_i}\right) + \left(r_0/r_i\right) 1/h_i} \qquad (8)$$

heat exchanger performance can be enhanced by different ways such as considering the initially heat exchanger is operating correctly, consider increase in pressure drop if available in heat exchanger with single phase heat transfer, heat transfer coefficient can be improved with increased velocity, consider fouling factor, for certain condition consider enhanced heat transfer through the use of finned tubes, inserts, twisted tubes or modified baffles [15].

A variety of different techniques are employed for the heat transfer process, which is generally referred to as heat transfer enhancement, and extensive reviews of these methods and their applications. These techniques are broadly classified as active or passive techniques. Passive techniques require no external energy input, except for pump or blower power to move the fluid, and involve the use of roughened surfaces, extended surfaces, displaced promoters, and swirl flow devices, among some others. Active enhancement techniques, on the other hand, need extra power to affect the process to improve heat transfer. Consequently, passive techniques are often the preferred choice and they have seen wider applications.

Of the many enhancement techniques that can be employed, swirl flow generation by means of full-length twisted-tape inserts is found to be extremely effective [1], [3], [4]. Significant heat transfer improvement can be obtained, particularly in laminar flows. Other examples of techniques that promote swirl flows include curved ducts, tangential fluid injection, and twisted or convoluted ducts. Their thermal-hydraulic characteristics, heat transfer improvement potential, and typical applications have been outlined.

It is well known that heat transfer is considerably improved if the flow is stirred and mixed well. This has been the underlying principle in the development of enhancement techniques that generate swirl flows. Twisted tape insert mixes the bulk flow well therefore heat transfer increases [12]. Among the techniques that promote secondary flows, twisted-tape inserts are perhaps the most convenient and effective. They are relatively easy to fabricate and fit in the tubes of shell-and-tube or tube-fin type heat exchangers.

Heat transfer enhancement may occur for three reasons:
1) The tape reduces the hydraulic diameter, which affects an increased heat transfer coefficient, even from zero tape twist.
2) The twist of the tape causes a tangential velocity component. Hence, the speed of the flow is increased near the wall. The heat transfer enhancement is a result of increased shear stress at the wall. Also heat transfer is enhanced by mixing fluid from the core region (cold) with fluid in the wall region (hot).
3) There may be heat transfer from the tape.

The geometrical features of a twisted tape, as depicted in Fig. 4, are described by its 180º twist pitch H, the thickness δ, and the width w. In most usage, where snug-to-tight-fitting tapes are used, w ≈ d, and the severity of the tape twist is characterized by the dimensionless ratio y=(H/d). The helical twisting nature of the tape, besides providing the fluid a longer flow path or a greater residence time, imposes a helical force on the bulk flow that promotes the generation of secondary circulation. The consequent well-mixed helical swirl flow significantly enhances the convective heat transfer [5].

The fully developed swirl flow performance, in particular, has been found to be influenced by the severity of the tape's helical twist y, and its tube blockage factor (δ / d), and the functional relationship can be expressed as[22]

$$f = \emptyset(Re, y, \delta/d) \qquad (9)$$

$$Nu = \emptyset(Re, y, \delta/d, Pr) \qquad (10)$$

Based on a fundamental balance between inertia, viscous, and tape-geometry helical curvature induced forces, it is proposed that tape-induced swirl flows can be scaled by a swirl parameter defined as[13]

$$Sw = Re/\sqrt{y} \qquad (11)$$

where, the swirl Reynolds number is based on the swirl velocity, and

$$Re_s = \frac{\rho V_s d}{\mu} \tag{12}$$

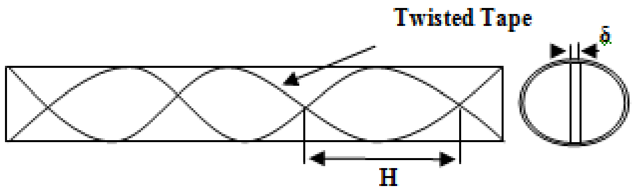


Figure 4: Twisted tape insert

The consequent well-mixed helical swirl flow significantly enhances the convective heat transfer [13]. In most cases, depending on how tightly the tape fits at the tube wall and what material it is made of, there may be some tape-fin effects as well. The enhanced heat transfer due to twisted-tape inserts, is also accompanied by an increase in pressure drop and suitable trade-offs must be considered by designers to optimize their thermal-hydraulic performance.

2. Problem Definition

In the present work a validation has been done for different following mentioned parameters of fluid flow inside a heat exchanger tube with a simple twisted tape insert. The validation has been done with work of Padalker et al [17]. The parameters of the flow which have been considered in this work are determination of friction factors for different Reynolds Number, Temperature distribution of the fluid inside the tube with twisted tape insert along its length, evaluation of velocity distribution at different cross-section and its representation in the form of vector, determination of local velocity and mean velocity at different position along the length. The validation work has been done using Fluent 6.3.26 and the Pre-processing work has been done with GAMBIT 2.4.6.

After validation work a design modification has been done on the twisted tape insert of the tube. As a modification, a groove has been crated at the center position of the tape so that flow can be made prolonged for the better heat transfer without sacrificing much the amount of pressure drop. After generating grooved twisted tape in GAMBIT above mentioned parameters have been determined by simulating it in Fluent and the results have been compared.

3. Review of Previous Work

Many works have already been done on the enhancement of heat transfer capability of the fluid flowing through a heat exchanger tube across the tube wall. Most of the research in this area been on the development of efficient flow swirl generator by modifying the topology of twisted tape insert incorporating different types of discontinuity like cuts or projections.

A. E. Bergles and S. W. Hong[1] did a work in 1964 regarding the investigation of effect of the geometrical parameters on the Friction and Forced Convection Heat-Transfer Characteristics in lutes fill Twisted Tape Swirl Generators. In their work they generated a relation mentioned below and validated the results from the relation with some previously published experimental results.

$$Nu_s = 5.172\,[1 + 5.484 * 10^{-3} Pr^{0.7} (Re_s/y)^{1.25}]^{0.5}$$

In the year of 1967 J. P. Du Plessis, D. G. Krogers [2] did a very important research on nature of flow over twisted tape. They considered the flow as laminar and developed a correlation between friction factor and flow parameters like Reynolds number. The relation they have derived is

$$f_D = f_D(\varepsilon)[1 + \left(\frac{Re_D}{y^{1.3}}\right)^{1.5}]^{1/3}$$

J. P. Du Plessis along with D. G. Kroger[3] created another relation to calculate the hydro-dynamical entrance length of a twisted laminar flow over a twisted tape insert in a heat exchanger tube. The relation they developed is

$$Nu_{m,D} = 1.58\Psi_s\,[1 + 0.153(x^{*}_{\infty})^{-1.05}]^{1/3} \times [1 + 0.000064(\Omega_s Pr)^3]^{0.117} \times [1 + 0.002\Omega_s^{1.4}]^{1/7}$$

After the work of J. P. Du Plessis many works were done [4]-[6] on the flow of different fluid over twisted tape in heat exchanger tube with different geometrical parametric considerations of twisted tape topology for the investigation of thermal properties and flow properties.

Sivashanmugam and Suresh [7] and Selvakumar and Suresh [8] did experimental work on the turbulence nature of fluid flow over a twisted tape with different geometrical configuration of a twisted tape. They also investigated the investigated heat transfer phenomenon and friction factor of the swirl flow. P. Murugesan, K. Mayilsamy, S. Suresh [10] studied the heat transfer and flow characteristic of a turbulent flow over a twisted tube with U cuts. The authors incorporated the ‘U’ cuts to enhance swirl phenomenon for the betterment of the heat transfer across the flow boundary of two fluids. Many researchers work on the enhancement of swirling in the flow over twisted tap by incorporating different types of cuts in the twisted tape. C. Nithiyesh Kumar, P. Murugesan [13] have done a detailed review on the works of many researchers on the investigation of different thermal aspects like temperature distribution, heat flux etc and different flow parameters like pressure distribution, velocity distribution etc for few recent years.

A very recent work has been done by Rupesh J Yadav, Atul S. Padalkar [17] in the year of 2013 on the characteristics of flow through a heat exchanger tube with twisted tape insert. The authors have considered four combinations of twisted tape insert, half-length upstream twisted tape condition, half-length downstream twisted tape insert, full-length twisted tape, inlet twisted tape and plain tube. It has been observed 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.

From the review of many works on the enhancement of heat transfer phenomenon by using different form of twisted tape insert, it is quite clear that the discontinuity which have been incorporated by many researchers are intermittent one. No work has been done on the flow and thermal analysis with any continuous discontinuity incorporated in the twisted tube. In the present work a study of flow and thermal characteristics of fluid flowing over a twisted tape insert having a semi-circular continuous groove as a continuous discontinuity has been done numerically using a CFD software named Fluent. In the present work, work of Rupesh J Yadav, Atul S. Padalkar [17] has been conserved a base paper.

4. Numerical Model of flow analysis and validation

First a topology of a plain tube and a twisted tube has been generated in GAMBIT 2.3.16 with following dimensions as per Padalkar et al [17]. Here in the present work study has been done on a Plain tube and Full length Twisted Tape inserted tube.

Table 1: Geometrical configuration for validation

Geometrical Configuration	Plain Tube (PT) and Full Length Twisted Tube (FLTT)
Twist parameter	0.14
Reynolds Number (Re)	25000, 50000, 75000, 100000
Inner diameter of the tube (d)	22 mm
Length of the heated or test section	2000mm
Length of the clamming section	1200 mm
Heat flux input	2300 W/m^2
Air temperature at the inlet	300 K
Twisted tape configuration	22 mm (Width) × 0.8 mm (Thick)

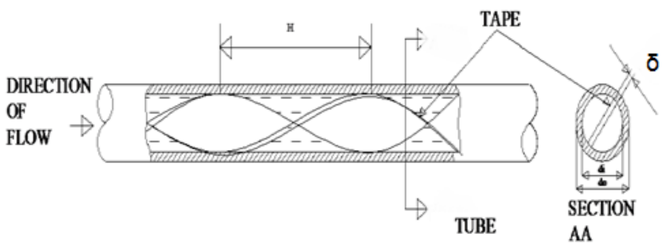


Figure 5A: Representation of partial tube with a twisted-tape insert

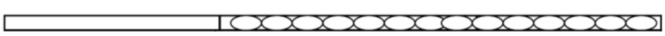


Figure 5B: Representation of position of twisted tape in the tube

In the work of validation first a plain tube topology has been created in GAMBIT and then it has been meshed and Boundary types have been identified as following.

Table 2: Detail of surfaces and boundary type

Surface	Boundary Type
Inlet cross-section of the tube	Velocity Inlet
Outlet cross-section of the tube	Outflow
Cylindrical surface of the clamming section	Wall
Cylindrical surface of the heated section	Wall

Where meshing is concern it is worthy to be mentioned over here that, mashing of this topology has been done by 'Boundary Layer' Technique with first row length 0.00022m, Growth rate 1.2 and nomber of rows 10 imposed on cylindrical surface areas of the tube. Following are the figures of meshed view of the tube.

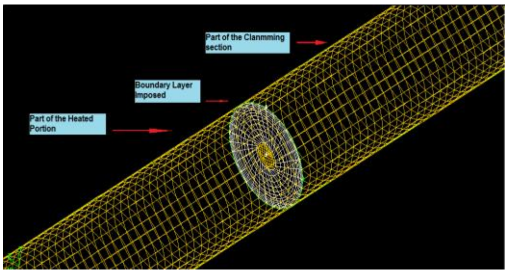


Figure 6: Partial and enlarged view of meshed geometry

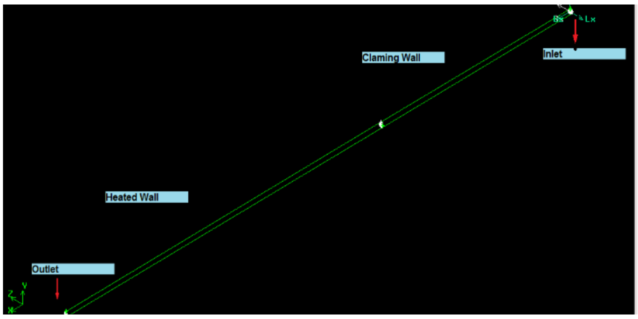


Figure 7: Geometry showing boundary conditions

4.1 Simulation result of plain tube

Table 3: CFD result of friction factor for plain tube at different Reynolds number

Reynolds Number	Density (kg/m3)	Area Weighted Average Wall Shear Stress (Pascal)	Area Weighted Average Velocity (m/s)	Friction Factor from Present work
25000	1.225	1.0211	16.458	0.0062
50000	1.225	3.7225	32.889	0.0056
75000	1.225	7.6615	49.336	0.0051
100000	1.225	11.8236	65.787	0.0045

Table 4: Friction factor from CFD result, analytical result, experimental result and result for plain tube by reference [17] comparison at different Reynolds numbers

Reynolds Number	From present work	From experiment [17]	From Analytical [17]	By Padalkar et al [17]
25000	0.0062	0.00652	0.0062	0.0068
50000	0.0056	0.0055	0.0054	0.0055
75000	0.0051	0.005	0.0051	0.0054
100000	0.0045	0.0049	0.0049	0.0042

Graph below represent the comparisons of the above mentioned values and affirms this fact that simulation done on plain tube by the present work is fully validated.

In the graph below there are four series of curves and these curves or graphs corresponds to the following results

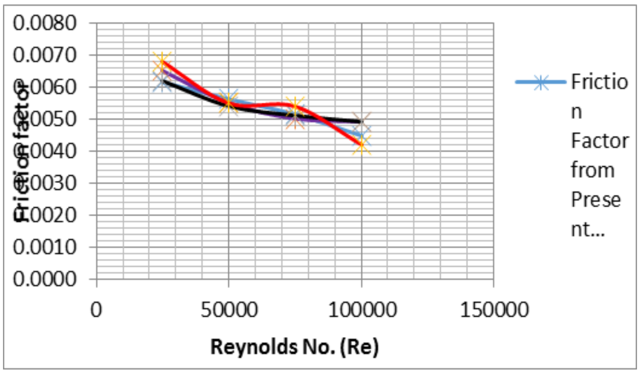


Figure 8: Graph depicting Friction Factor at different Reynolds Number

Now two parameters have been calculated from the simulation to depict the fact that how efficiently heat is being accepted by the flowing fluid i.e air and how efficiently it is being carried out by the fluid. These performance parameters are 'Area Weighted Average' of Outlet Temperature and Heat flux.

Before the above mentioned performance parameters have been mentioned, temperature distribution of the flowing air at Mid-Plane has been presented below for Reynolds no 25000.

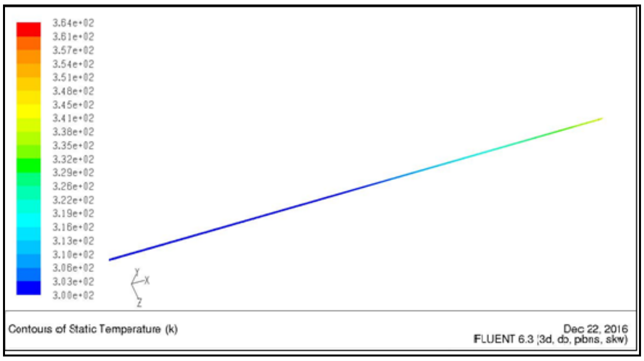


Figure 9: Temperature Distribution along the length

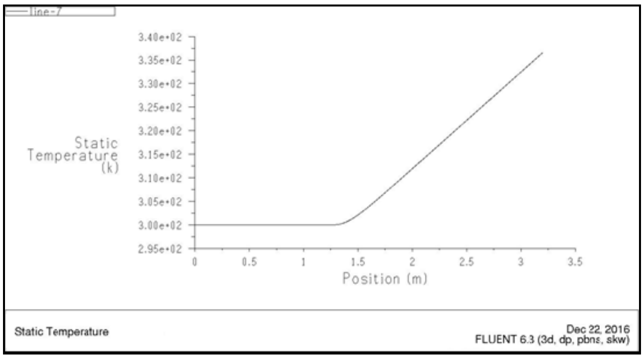


Figure 10: Temperature Distribution graph along the length

The maximum temperature reached is 337 K for the plain tube. Average heat flux is calculated as 89.75 W/m2.

Now a plain tube with an insert has been simulated using the data motioned at the beginning of this chapter. Following are the corresponding figures

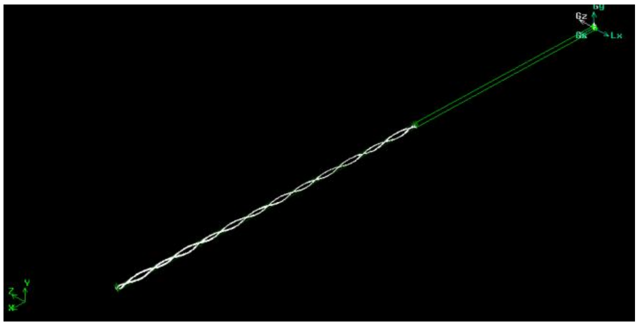


Figure 11: Tube topology with insert created in GAMBIT

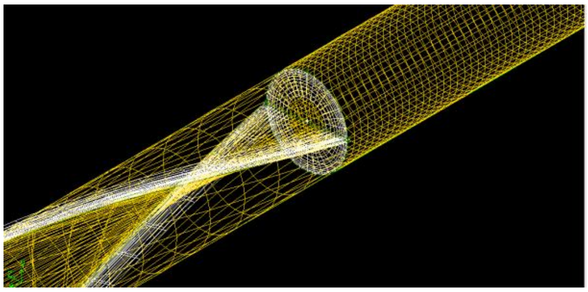


Figure 12: Enlarged and Partial view of tube with inserts showing the 'Boundary Layer'

Now this meshed model has been imported in Fluent and simulated for Reynolds's Number 25000 and under same other condition mentioned above. After simulation graphs for ratio of local velocity and mean velocity have been generated at cross-section at distances from inlet correspondingly 1.8m, 2.4m and 3.07m. Theses graphs are well agreed with the graphs generated by Padalkar et al [17] in their work. In this context it is worthy to be mentioned here that graphs generated in the present work have been presented separately due to the fact that number of cell at different cross-section are different.

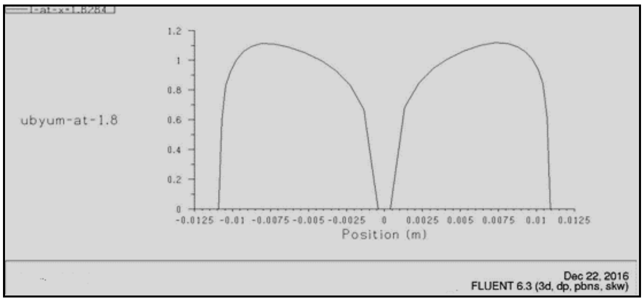


Figure 13: Graph between ratio of local velocity (U) and mean velocity (Um) and different radial position on the cross-section at distance of 1.8m from inlet.

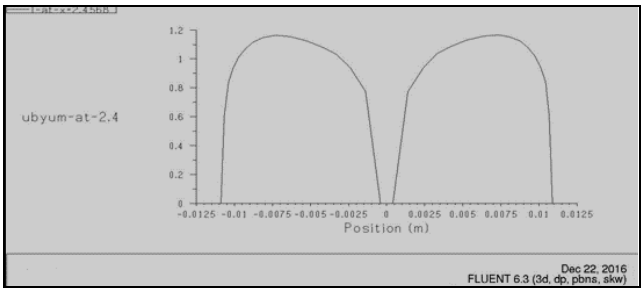


Figure 14: Graph between ratio of local velocity (U) and mean velocity (Um) and different radial position on the cross-section at distance of 2.4m from inlet

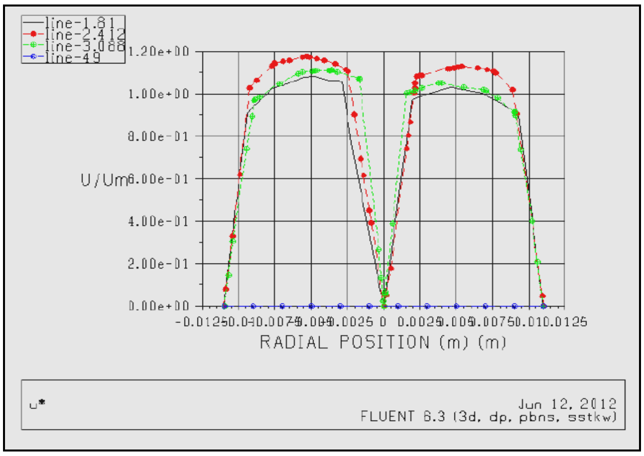


Figure 15: Graph between ratio of local velocity (U) and mean velocity (Um) and different radial position on the cross-section generated by Padalkar et al.

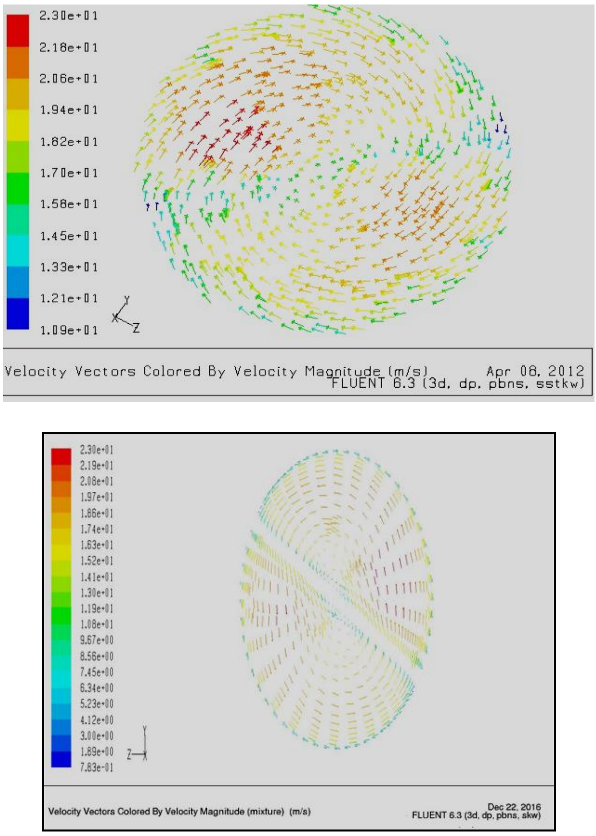
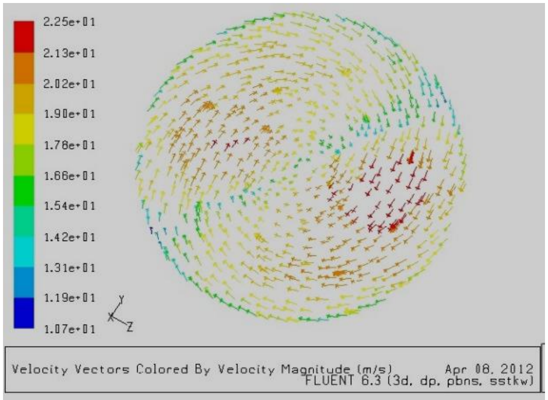


Figure 16: Velocity vector plot of full length twisted tape in reference [17] and present work at X/d =27



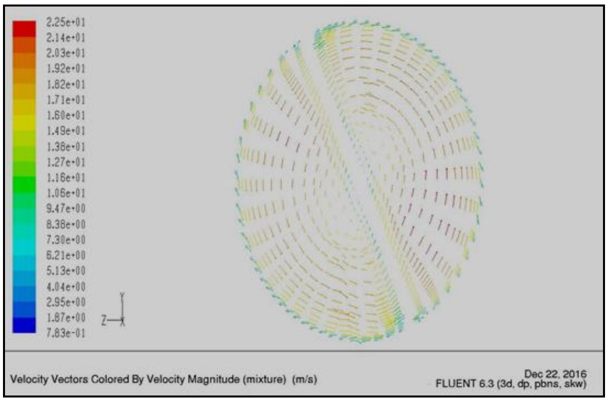


Figure 17: Velocity vector plot of full length twisted tape in reference [17] and present work at X/d =54

4.2 Design Modification Adopted

After Validation of the published work, investigation has been done for a tube with modified twisted tape insert. As modification on the twisted tape, a rib cut has been incorporated and simulated in Fluent. Below are the corresponding figures.

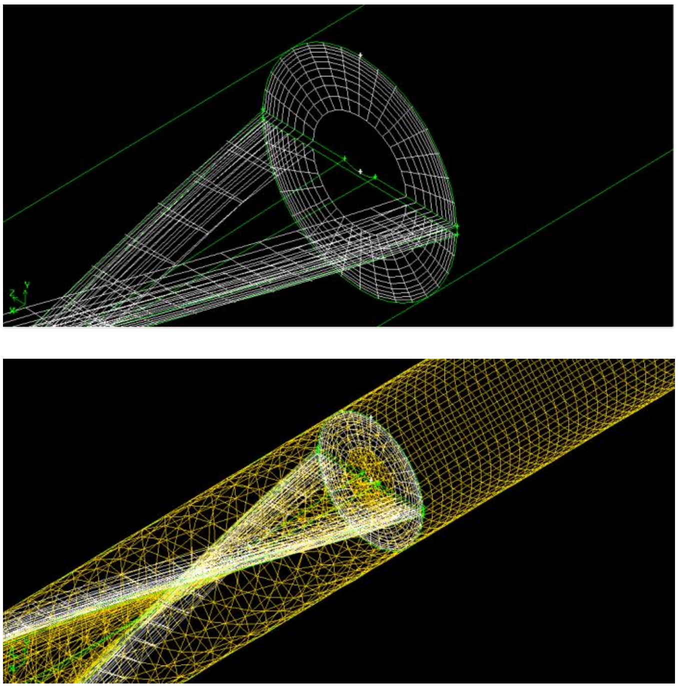


Figure 18: Enlarged and Partial meshed view of above topology

5. Result and Discussion

After generating mesh of the tube with modified tape, it has been imported by Fluent for further simulation. Simulation in Fluent has been done for Reynolds's Number 25000 and under the other conditions as mentioned above.

Now temperature contour has been generated at the mid pane and Temperature of the fluid at the outlet has been calculated and also the heat flux has been calculated at the outlet and then it has been compared with the plain tube and tube with twisted tape. Below is the figure for contour plotting of temperature at the mid plain.

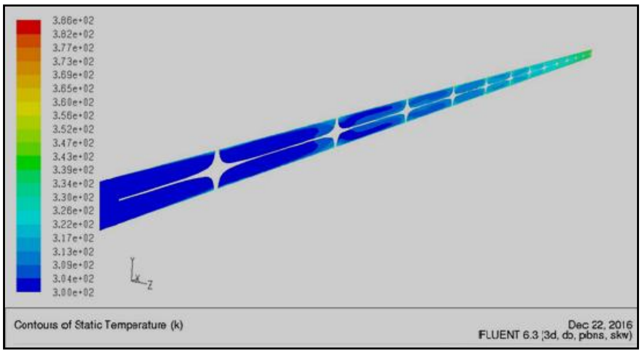


Figure 19: Temp distribution at mid plain for tube with twisted tape insert

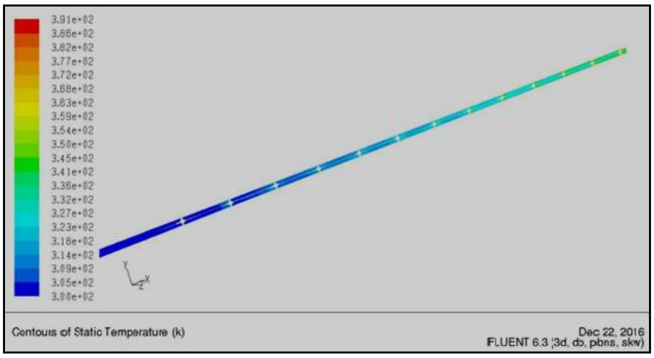


Figure 20: Temperature distribution at mid plane for tube with modified twisted tape inserts

Flowing table represents the comparisons of thermal values for plain tube without any insert, tube with twisted tape insert and tube with modified twisted tape insert.

Table 5: Comparison of maximum temperature heat flux of PT with twisted tape insert and Modified twisted tape insert

Tube Configuration	Maximum Temperature at Tube Outlet	Area Weighted Average Temperature at Outlet	Area weighted average heat flux at the outlet
Tube with Twisted tape insert	372.0807 K	340.82 K	91.929 W/m2
Tube with modified twisted tape insert	383.7201 K	341.5766 K	91.795 W/m2

On the basis of the above result a conclusion has been drawn in next section where insertion of the proposed modification has been justified and further scope on this research has been discussed.

6. Conclusions and Future Scope

From the previous chapter it is very much clear that insertion of a twisted tape in a plain tube increase the thermal performance of the tube and furthermore if a rib is cut on the twisted tape surface it increases the tube's thermal performance more. This fact has been depicted more clearly below.

Area Weighted Average of the fluid's temperature at the outlet of the tube has been increased due to the insertion of a twisted tape in the tube. The reason for the increment of these parameters is that, due to the insertion of a twisted tape a vortex flow is created in the pipe which helps the fluid to take more and more heat from the tube wall. The vortex has been shown below

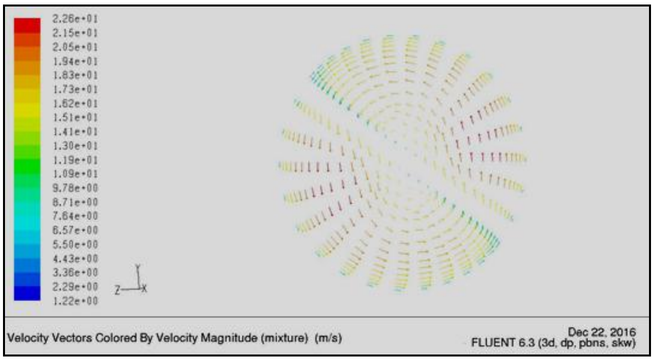


Figure 21: Vortex flow inside the tube with twisted tape

Now when a rib cut is given on the upper surface of the tape, an additional vortex is created and due to this Area Weighted Average of the outlet temperature has been increased little bit (temperature increased by 0.2%). But due to this extra vortex which has been shown below, stagnation effect comes into play and mass flow rate decreases. Due to this, heat flux at outlet also decreases.

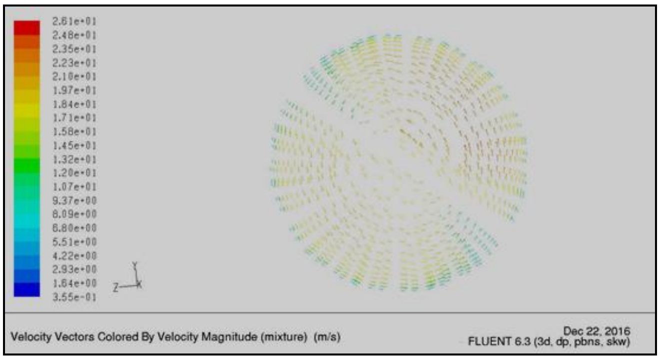


Figure 22: Vortex flow inside the tube with modified twisted tape

So it may be concluded that, modifications should be done in such a way so that average temperature as well as heat flux both increases. To do this, optimization procedure may be adopted to optimize different parameters to achieve the desired goal.

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