Numerical Simulation of Fluid Flow and Heat Transfer in Enhance Twisted Elliptical Tubes

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Abstract

This study is carried to find the thermal hydraulic efficiency of smooth twisted elliptical tubes and enhanced twisted elliptical tubes which has fins at both axes e.g. Major axis and the Minor Axis of elliptical cross section of tubes. And each tube has five numbers of sub types with different values of twist pitch (e.g.62.5, 83.33, 125 and 150 mm). In this way a total 10 numbers of tubes were generated. Water liquid is used as the working fluid. Reynolds number in this study ranges from 10000 to 50000. As flow is in turbulent regime, realizable K-ε turbulent model is used to simulate the flow in turbulent regime. Heat transfer mechanism is forced convection as the Test Section Wall has a constant outer temperature. Results shows that average Nusselt number increase with increasing Reynolds number also Pressure drop increases. Enhanced twisted tube with 62.5 mm twist pitch shows the best thermal results compared to smooth elliptical tube. For Reynolds number 10000 to 20000 ETETB with 62.5mm pitch, for Reynolds number 20000 to 40000 ETETB with 83.33 mm and for 50000 Reynolds number ETETB with 125mm pitch shows the best thermal hydraulic efficiency.

Key Words: *Heat transfer, Enhanced twisted elliptical tubes, numerical simulations*

Table of Contents

List of Figures

Chapter 1: Introduction

Heat transfer is widely applicable in all engineering processes and efficiency of energy utilization needs to improve. Different types of heat exchangers are being used in industry as well as in domestic appliances. In which heat exchangers with tubes are most common. This study is oriented to enhance the heat transfer in heat exchangers by varying the tube configuration using passive techniques. In passive techniques introducing twists in tube and introducing fins are most common. These twist and fins increase the turbulence and heating surface area in flow which increases the heat transfer and pressure drop at same time.

Bejan [1] illustrated the forced heat convection for four different flows and analyze the irreversibility due to finite temperature gradients and viscous effects analytically. Prakash and Patankar [2] studied the convective heat transfer in vertical tube with radial fins for free and forced convection. Also study to figure out the effect of fin heights and number of fins using finite difference technique. He found that a critical Rayliegh number at which reverse flow produced. Higher values for buoyancy increase the heat transfer and friction. GUOt, LI [3] suggested three methods for improvement convective heat transfer.(a)increasing the Reynolds number. (b) fullness of velocity and temperature profile. And (c) maximize the angle between velocity vector and temperature gradient. Copetti, Macagnan [4] carried out a single phase experimental study from laminar to turbulent regime for smooth and fined tube for Reynolds number varying from 5000 to 20000. And shows heat transfer co efficient ratio of micro fined to smooth tube is 2.9 in turbulent regime. Similarly pressure drop ratio is 1.7 and increase in heat transfer is about 80%. Han and Lee [5] investigated the heat transfer for four tube with different spiral angles and surface roughness experimentally for Reynolds number varying from 3000 to 40000. Tube with higher surface roughness and lower spiral angle shows better thermal efficiency comparatively to tubes with higher spiral angle and lower value of surface roughness. [6] studied the finned tube with Reynolds number ranges from 2500 to 90000 using oil and water as working fluid. He found that before critical Reynolds number smooth tube shows better result. After Reynolds number 30000 heat transfer twice. Zdaniuk, Chamra [7] studied the helix angles, number of fins start and the height to diameter ratio effect on the performance of tube for Reynolds number ranges from 12000 to 60000.Ağra, Demir [8] studied the enhanced tube with fluid flowing outside in annulus numerically. Reynolds number ranges from 12000 to 57000. Two types of enhanced tubes were considered in study. Firs case is two different finned tubes and the second case is two different corrugated tubes. Finned tube shows best results among all tubes. While corrugated tube shows better thermal performance than smooth tube but lower then finned tube. Dastmalchi, Arefmanesh [9] studied the laminar flow in smooth and finned tube for Reynolds number ranges from 100 to1000 by varying the helix angle and fin heights. Numerical results indicate 44% increase in heat transfer and 69% increase in friction factor comparatively to the smooth tube at same parameters.

Lei, Zheng [10] carried out stud to provide a better solution for heat transfer in tubes. He used insert with punched delta wings vortex generator. Numerical simulations were performed for different angles and pitch of wings. Results demonstrate that with increasing attack angle and reducing pitch of delta wing vortex generator increase Nusselt number. Naik and Tiwari [11] studied the position and angle of the vortex generating rectangular winglet pairs. Results shows 37.6% increase in thermal performance at optimal location and angle of attach. Chen, Chen [12] studied the condensation of R22 and R410A refrigerants outside the horizontal smooth tube, herring bone tube and enhanced tube (tube with dimples). Experimental results show that for low mass flux dimple tube has minimum thermal performance while for high values for mass flux smooth tub tube has minimum thermal performance. Finally, its concluded that herring bone tube shows the best performance for annulus side condensation. YongliangWan, RunhanWu [13] carried experiments on smooth tube and corrugated tube filled with copper foam. Also flow water containing Nano particles is studied. Combine effect of corrugated tube, copper foam and Nano fluids enhance the heat transfer 20%-600% compared to simple smooth tube. Nusselt number is proportional to the Reynolds number. The resistance for tube filled with copper foam increased 2-4 times.

Abdous, Holagh [14] numerically studied the sub cooled saturated flow boiling in curved and helical tube with phase change. Outer surface has higher values of heat transfer co efficient due to radial pressure gradient as more bubbles are accommodated at inner surfaces. Higher pitch vales show higher heat transfer co efficient as wet ae increases. Larger tube diameter increases the vapor generation while the smaller tube diameter has higher heat transfer coefficient. Jamshidi and Mosaffa [15] numerically studied the conical coil tube geothermal heat exchanger. They examine the coil diameter, coil pitch, conic angle, length to width ratio of fins at different Reynolds numbers. They also studied the fraction of AL_2O_3 to enhance the thermal efficiency. Nano particle increase heat flux on the surface of the tube by 18% at addition of volume fraction of 0.5%.

Meng, Liang [16] experimentally find he heat transfer performance in alternating axis elliptical tube for Reynolds number ranging from 500 to 50000. Results showed that better thermal performance with low flow resistance comparative to the other methods due to longitudinal vortices. Also result were validated by using numerical method for one case. Yang, Zhang [17] studied the twisted elliptical tubs with different aspect ratios and pitch. And concluded that tubes with higher aspect ratio and lower pitch values shows better heat transfer. experimentally studied the following five tube as shown in**Error! Reference source not found.** Table 1 Yang, Zhang [17] Test matrix. For Reynolds number 600 to 55000. TET enhance the results for low Reynolds number significantly. Nusselt number ratio increase and then decreased with increasing Reynolds number. Nusselte number ratio from 2.8 to 1.4 is observed for Reynolds number 2300 to 55000. Maximum pressure drop ratio is observed in tube three with maximum aspect ratio from 3.5 to 2.4. overall thermal hydraulic performance a higher Reynolds number is less then at low Reynolds numbers. Thermal hydraulic performance factor for tube 4 and 5 is lesser than tube 1.

Table 1 Yang, Zhang [17] Test matrix

Tube Number	Pitch [mm]	Aspect Ratio
01	104	1.6
02	105	1.9
03	192	2.15
04	192	1.76
05	192	1.49

Tan, Zhu [18] considered the tube and shell type heat exchanger with twisted elliptical tube. And compare the results with same sized rod and baffle type heat exchanger and found that TET and shell heat exchanger show better performance. Also studied the flow and heat transfer in twisted elliptical tube and shows that heat transfer and pressure drop is relatively higher than the smooth tube. Kim, Kim [19] studied aero thermal performance the different geometrical parameter for twisted elliptical tube via numerical simulation. Study was carried out for Reynolds number for 100, 1000 and 10000 using Realizable k-epsilon turbulence model and steady incompressible RANS equations. Result shows that variation in friction factor is greater with varying pitch and smaller for aspect ratio comparatively. Colburn j-factor [$\frac{NU}{RePr^{1/3}}$] increase with decreasing pitch. Also friction and Coburn-j factor both increased with decreasing pitch. Wu, Chen [20] numerically studied the effect of number of twists and the Reynolds number on the thermal hydraulic performance of TET. They studied the three different pitch values for Reynolds number ranging from 10000 to 15000. They use k-ω turbulence model. Results shows that thermal resistance decreases with increasing Reynolds number. While Nu No. and pressure drop increased with increasing Re No. Thermal resistance decrease as pitch decreases. Nusselt number increases as pitch values decreases. Also pressure drop increased with decreasing pitch. Also above study its concluded that tube with 128 mm shows best thermal hydraulic performance. Results show 58%-60% increase in pressure drop while 16%-19% increase in average Nusselt number. Due to twisted wall pressure drop increase significantly.

Numerical simulation of heat transfers and fluid flow is carried out in this study. Smooth twisted elliptical tube and finned twisted elliptical tubes with five different pitch values will be studied. Finned tubes has fins at major and minor axis of tube. Numerical study will be carried out to investigate the heat transfer efficiency and the pressure drop in each tubes with Reynolds number10000 to 50000. To introduce a new configuration of heat exchanging tube with better thermal efficiency.

Chapter 2: Physical Modeling and Simulations

2.1 Geometries

For said study two major categories for tubes are considered. First one is **Twisted Elliptical Tube** (TET), in which simple elliptical cross sectioned tube is provided with twists and the other one is **Enhanced Twisted Elliptical Tube** (ETETB). Enhanced tube has fins on both axes of elliptical cross sectioned e.g. major and minor axis.

Both tubes have five different configurations as per number of twists. Number of twist provided on tubes are 5, 6, 9 and 12. A simple smooth for both types was also prepared for comparison purpose. In this way total 10 number of tube were prepared for this study. Following Generic Twisted Elliptical Tube illustrate the general shape of tubes. While the Figure 2 Cross Section of Twisted Elliptical Tube and Figure 3 Cross Section of Enhanced Twisted Elliptical Tube with fins on Both AxesB) shows the cross section of TET and ETETB respectively. Pro E WF 05 is use for CAD modeling.

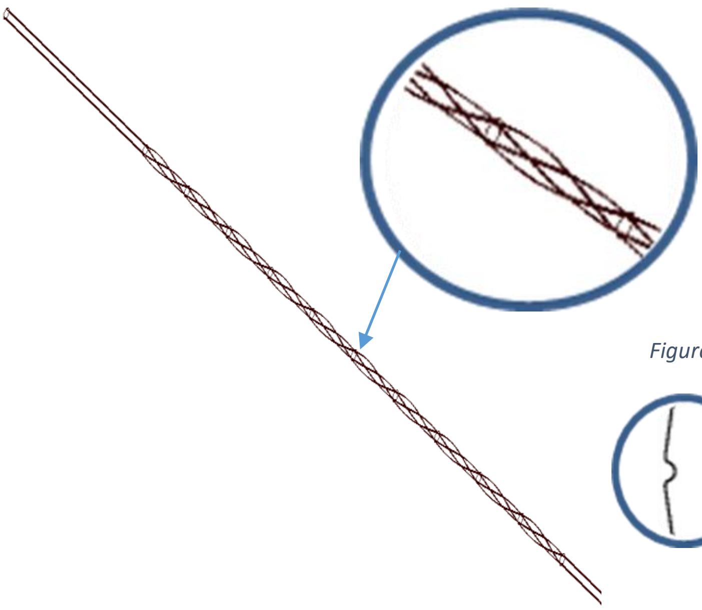


Figure 1 Generic Twisted Elliptical Tube

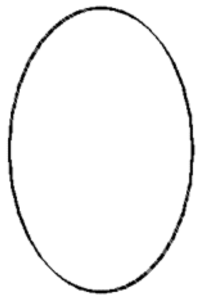


Figure 2 Cross Section of Twisted Elliptical Tube

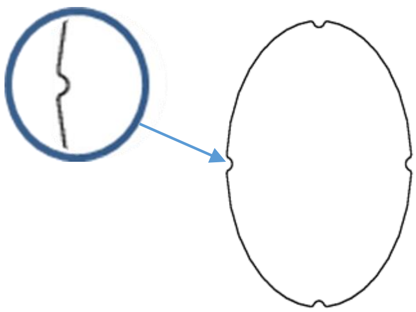


Figure 3 Cross Section of Enhanced Twisted Elliptical Tube with fins on Both Axes

2.1.1 Entrance and Exit length

For flow development, entrance and exit lengths are provided at entrance and exit of each tube. In this way each tube is consisted of three main sections inlet section, Test section (TS) and outlet section described in following Figure.

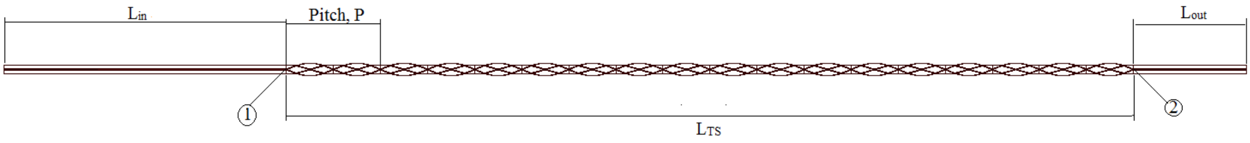


Figure 4 Descriptive Physical Model of Twisted Elliptical Tube.

Inlet length for fully developed flow is calculated by following formula by Çengel [21]

$$L_{in\,(Turbulent)} = 4.4Re^{1/6} *D_H$$

Exit length for turbulent flow is negligible as described by Perry [22] Anyway in his study an exit length of 100 mm is provided in each tube.

All inlet and outlet parameters were studied at point 01 and 02 which are test section's inlet and outlet faces respectively as shown in Figure 4 Descriptive Physical Model of Twisted Elliptical Tube.. All physical dimensions are mentioned below in Table 2 Physical Model Dimensions.

Table 2 Physical Model Dimensions

Sr. No.	Parameter	Value [mm]
01	Major axis Diameter	11
02	Minor axis Diameter	8
03	Aspect Ratio	1.375
04	Fin Diameter	0.5
05	Fillet Radius	0.15
06	L_{in}	250
07	L_{out}	100
08	L_{TS}	750
09	Pitch (P)	750/No. of Twist[1]
10	L	1100

[1] As Number of Twist are 05, 06, 09 and 12

2.2 Mesh

Due to certain limitation of analytical solution of governing equations for complex domains, we need to find out the approximate solution. For this purpose, all partial differential governing equations need to discretize to analyze the flow. Also domain needs to be discretized in further subdomains to solve discretized PDEs.

The process of dividing the main domain into sub domains is known as meshing. Mesh is consisted of elements or cells. Each element has a shape of triangle or quadrilateral for 2D geometries and each element has a shape of tetrahedron, quadrilateral pyramid, triangular Prism hexahedron. Also with advance shape such as polyhedron which has any number of vertices, edges and faces.

Solution accuracy, precision and rate of convergence are dependent on the quality and size of the mesh. As mesh plays a vibrant role on the accuracy of results so to reduce the discretization error grid independence study was carried out to ensure the results reliability.

2.2.1 Grid Independence

A grid independence study is carried out to find the accuracy of grid. For this purpose, the **Enhanced Twisted Elliptical Tube with Fins at both Axis (ETET),** having 12 numbers of twists is chosen due to its most complex geometry among all physical models of twisted tube considered in study.

Four number tetrahedron mesh (GI1, GI2, GI3, and GI4) were generated by using ANSYS 16 fluent's Mesh module with different numbers of cells and node. Then these grids were converted to polyhedral mesh by using fluent setup module as polyhedral mesh shows better results with low number of cells for 3D models. As IQBAL and CHAN [23] carried out simulation for tetrahedron, hexahedron and polyhedron meshes for high rise building. Hexahedron mesh are easy to generate for simple geometries whiles tetrahedron and polyhedron mesh are being used for complex geometries. They concluded that polyhedral show best performance with low usage of memory and increase convergence. They validate these result by experimental data acquired by the wind tunnel testing.

Polyhedral Mesh's Statistics and Quality for said tube are presented in Table 3 Polyhedral Mesh Statistics considered in Grid independence study.

Table 3 Polyhedral Mesh Statistics considered in Grid independence study.

Nos. of Twist	Type of Tube	Grid	No. of Cell	No of Node	Min. Orthogonal Quality	Max. Orthogonal Skew	Max. Aspect Ratio
12	ETETB	GI1	1078010	5219110	2.12E-01	7.88E-01	1.14E+01
		GI2	2810877	12790207	2.23E-01	7.77E-01	1.36E+01
		GI3	4530624	25240995	0.160742	0.839258	27.7048
		GI4	5133031	29219545	0.145898	0.854102	35.5808[i]

Same input and output boundary conditions were applied to study the accuracy of grid. With same input parameters output parameters were studied for all four cases. Results are shown in Table 4 Output parameters values of all 04 grids..

Table 4 Output parameters values of all 04 grids.

Flow Rate [kg/s]	Total Heat flux [w/m2]	Nu avg. [m^2]	Delta P [pa]	Delta T [K]
0.07671	108745.72	174.8946	3958.295	18.5063973
0.07683	113774.38	183.4207	3942.277	18.1702674
0.07689	116784.21	188.8476	3875.280	17.9576397
0.07691	117402.15	189.8704	3874.539	17.9151978

Table 5 Perecentage Error of coarse to fine grids

Nos. of Twist	Type of Tube	No. of Cells	% Error (Total Heat Flux)	% Error (Nu.Avg)	% Error (Delta P)	% Error (T out)
12	ETETB	1078010	4.624237165	4.875010435	-0.404654	-1.81629
		2810877	2.767768699	3.102931709	-1.692572	-1.14894
		4530624	0.568243054	0.584809365	-0.018738	-0.22934
		5133031	0	0	0	0

Above

Table 5 Perecentage Error of coarse to fine grids illustrate the percentage error reduction for each iteration. Figure 5 Percentage Error Reduction plots of Error in average Nussle number, Pressure Drop (Pa), Temperature Difference (K) and Total Heat Flux through wall (W/m^2) respectively.

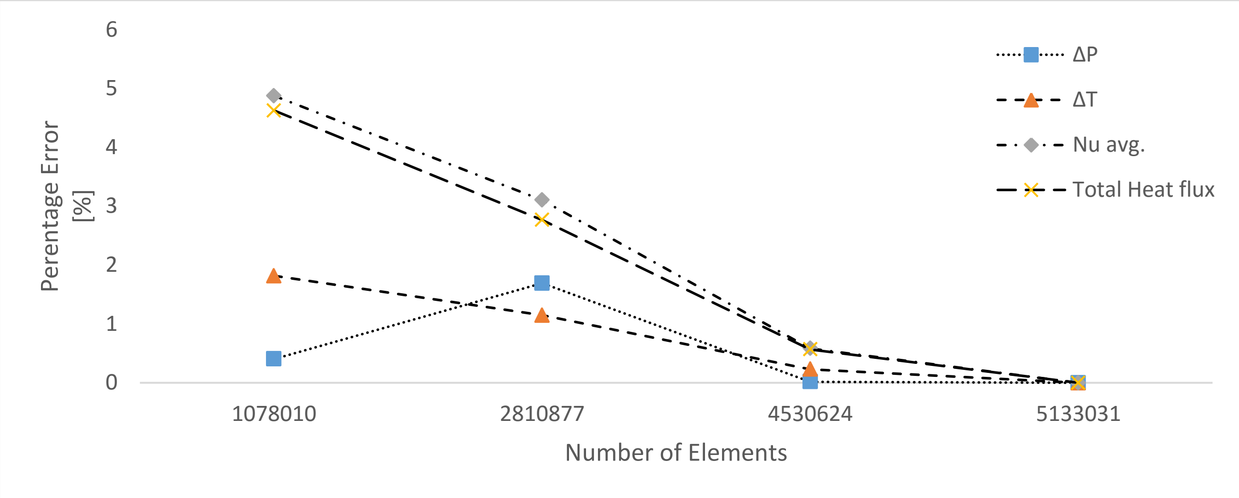


Figure 5 Percentage Error Reduction

After analyzing the results, it is concluded that by increasing the number of cell from 4530624 to 5133031, percentage error reductions is less than 0.6% for all parameters. It means that further refinement in grid just costs increase in calculation time rather than improving the results significantly. So, further refinement stops due to aforesaid reason. For simple geometries it has been decided to use same mesh with same minimum and maximum sizes with same methodology.

2.2.2 Mesh Statistics

On the basis of gird independence study tetrahedron mesh were generated for all geometries and all these meshes converted in polyhedron mesh. Mesh statistics such as number of elements, number of nodes, minimum orthogonal quality, maximum skewness and maximum aspect ratio for all types of meshes are given Table 6 Tetrahedron Mesh Statisticsand Table 7 Polyhedron Mesh statistics.

Following Figure 6 Tetrahedron Mesh and Figure 7 Polyhedron Mesh shows the tetrahedral and polyhedral mesh for all geometries.

Table 6 Tetrahedron Mesh Statistics

Type	No. of Twist	No. of Nodes	No. of Elements	Min Orthogonal Qality	Max Orthogonal Skew	Max Aspect Ratio
TET	0	5929039	32740481	0.23081	0.82505	10.35
	5	4863942	26855020	0.22554	0.80028	10.633
	6	4901241	27061339	0.22219	0.79983	10.913
	9	5009733	27657650	0.22336	0.81413	11.185
	12	5188879	28656770	0.23104	0.79987	10.443
ETETB	0	4912193	26958081	0.22789	0.83413	10.614
	5	4912193	26958081	0.22629	0.80948	10.537
	6	4986987	27374585	0.22398	0.83272	10.679
	9	5515432	30368285	0.22752	0.81185	10.933
	12	5203910	28533578	0.22156	0.82179	10.712

Table 7 Polyhedron Mesh statistics

Type	No. of Twist	No. of Nodes	No. of Elements	Min Orthogonal Qality	Max Orthogonal Skew	Max Aspect Ratio
TET	0	34538043	5929039	3.01E-01	6.99E-01	1.00E+01
	5	28325610	4863942	3.79E-01	6.21E-01	9.93E+00
	6	28543522	4901241	3.70E-01	6.30E-01	1.13E+01
	9	29173833	5009733	3.38E-01	6.62E-01	9.86E+00
	12	30220046	5188879	3.45E-01	6.55E-01	1.08E+01
ETETB	0	28614074	4938148	2.35E-01	7.65E-01	1.53E+01
	5	30197230	5739569	1.32E-01	8.68E-01	1.46E+01
	6	30594626	5789205	1.98E-01	8.02E-01	1.32E+01
	9	33638273	6304601	2.23E-01	7.77E-01	1.49E+01
	12	32033942	6120395	1.43E-01	8.57E-01	1.77E+01

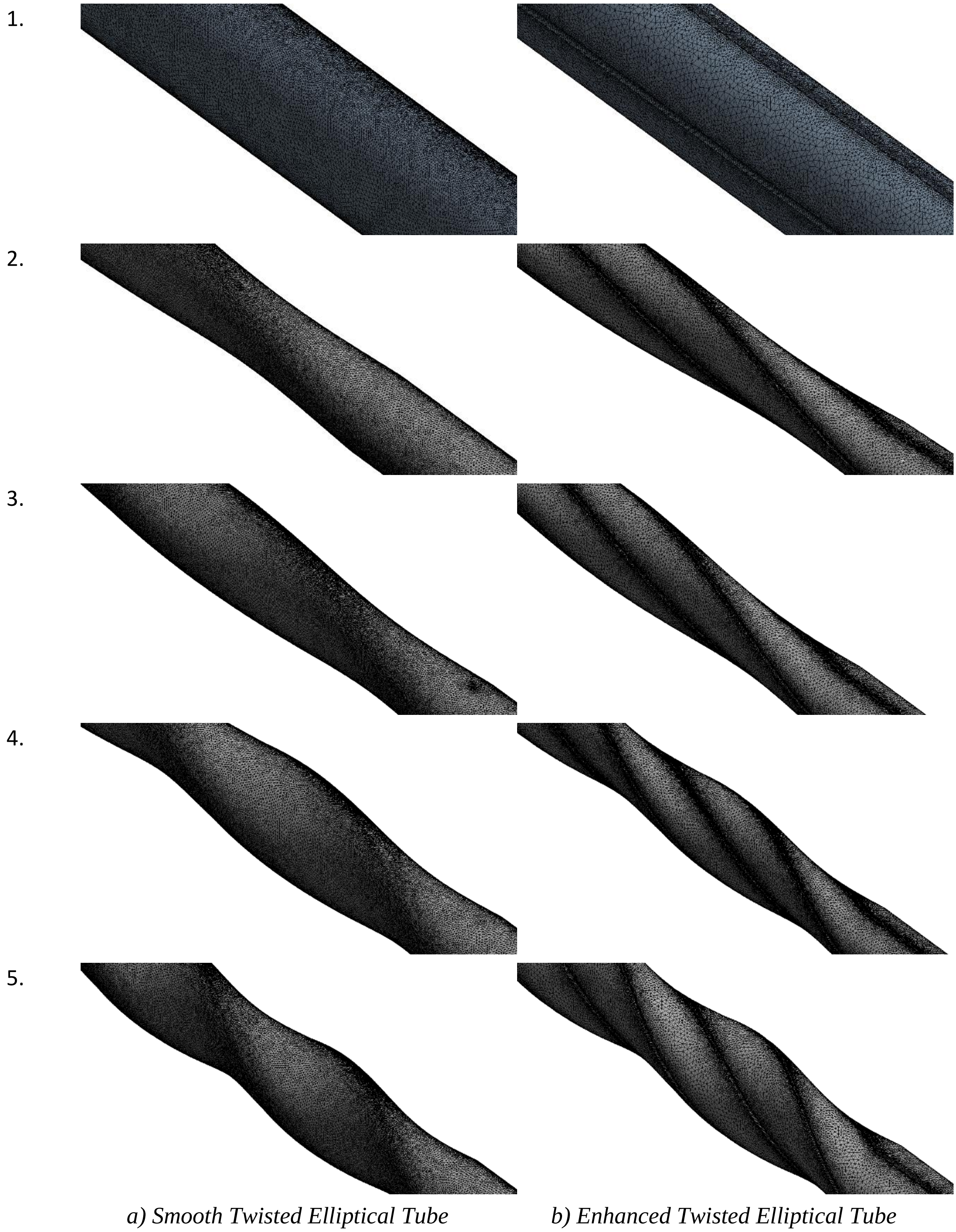
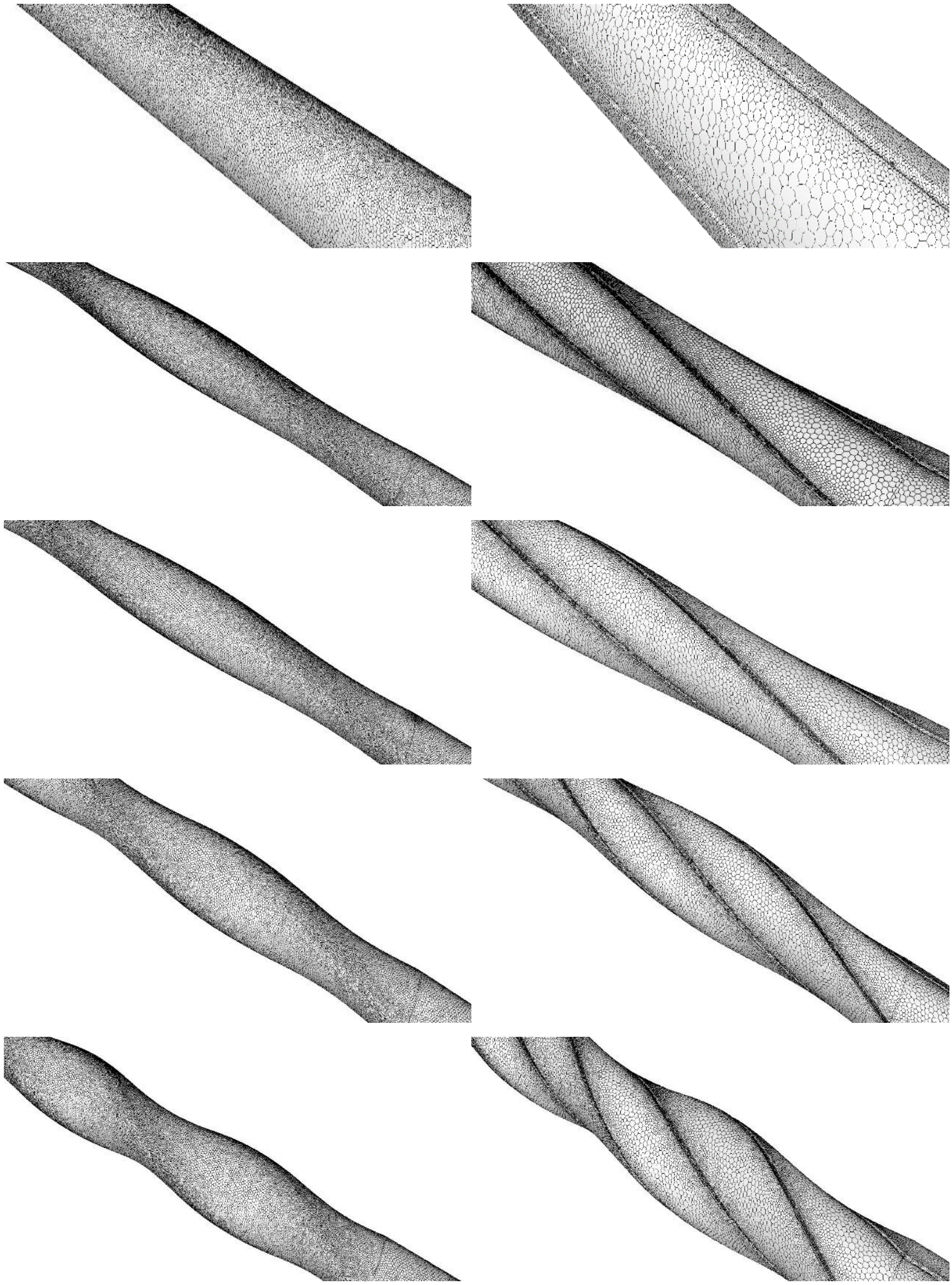


Figure 6 Tetrahedron Mesh

1) for smooth Tube 2) for twisted tube with Pitch 150 mm 3) for twisted tube with Pitch 125 mm 4) for twisted tube with Pitch 83.33 mm 4) for twisted tube with Pitch 62.5 mm



1.
2.
3.
4.
5.

a) Smooth Twisted Elliptical Tube *b) Enhanced Twisted Elliptical Tube*

Figure 7 Polyhedron Mesh

1) for smooth Tube 2) for twisted tube with Pitch 150 mm 3) for twisted tube with Pitch 125 mm 4) for twisted tube with Pitch 83.33 mm 4) for twisted tube with Pitch 62.5 mm

Finer mesh is generated near proximity and curvature as depicted in above figures. As per mesh statistics it is clear that polyhedron mesh for TET is higher in quality than tetrahedron. While for ETET 5, 6 and 12 have cells with skewness maximum value of 0.87. Number of cells in polyhedron mesh decreased 5 to 6 times.

2.3 Solver Setup

For said study an octa core machine with 08 MB cache and 22 GB of RAM is used. For solution purpose parallel processing option is chosen with 7 number of processors. Flow in this study is considered as single phase and turbulent. A 3D, steady state, pressure based solver is chosen of this study. Realizable K-Ɛ turbulent model with enhanced wall function for thermal effects is chosen for flow study in turbulent regime as is most suitable model for internal flows due to its versatile applications.

2.3.1 Governing Equations

As flow is steady state, uniform and turbulent. And fluid is viscous and incompressible with constant properties. Constant wall temperature with no slip condition with wall is considered. And radiation effects are negligible. So resulting governing equations are as follows.

i. Continuity Equation

$$\nabla \cdot \rho\, U = 0.$$

ii. Momentum Equation

$$\rho U . \nabla U = -\nabla\, p + \mu \nabla^2\, U$$

iii. Energy Equation

$$\nabla \cdot\ (\rho U C_P T) = \nabla . (k \nabla\, T)$$

Where ρ, U, T, p, C_P and μ are the density, velocity, temperature, pressure, specific heat and viscosity of water, respectively.

2.3.2 Turbulence Model

Governing equation for relizable K epsilon model is given below. First equation is used to calculate the rate of chnge in kienatic engery of fluid flow and second one for energy dessipation rate due to turbulance.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k + G_b - \rho\varepsilon - Y_M + S_k$$

$$\frac{\partial}{\partial t}(\rho\varepsilon) + \frac{\partial}{\partial x_j}(\rho\varepsilon u_j) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]\rho C_1 S_\varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\upsilon\varepsilon}} + C_{1\varepsilon}\frac{\varepsilon}{k}C_{3\varepsilon}G_b + S_\varepsilon$$

$$C_1 = max\left[0.43, \frac{\eta}{\eta+5}\right];\ \eta = S\frac{k}{\varepsilon};\ S = \sqrt{2S_{ij}S_{ij}}$$

In these equations, G_k represents the generation of turbulence kinetic energy due to the mean velocity gradients, G_b is the generation of turbulence kinetic energy due to buoyancy, Y_M presents the contribution of the fluctuating dilatation in compressible turbulence to the overall dissipation rate, C_1 and $C_{2\mathcal{E}}$ are constants. σ_k and $\sigma_{\mathcal{E}}$ are the turbulent Prandtl numbers for K and Ɛ, respectively. S_k and S_ε are user defined source function.

Following equation used to compute the turbulent viscosity

$$\mu_t = \frac{\rho C_\mu k^2}{\varepsilon}$$

The difference between the realizable model and the standard and RNG models is that C_μ is no longer constant. It is computed from

$$C_\mu = \frac{1}{A_0 + A_s \frac{kU^*}{\varepsilon}}$$

Where

$$U^* = \sqrt{S_{ij}\,S_{ij} + \widetilde{\Omega_{ij}}\widetilde{\Omega_{ij}}} \quad : \quad \widetilde{\Omega_{ij}} = \Omega_{ij} - 2\varepsilon_{ijk}\omega_k \ :\ \Omega_{ij} = \overline{\Omega_{ij}} - 2\varepsilon_{ijk}\omega_k$$

Where Ω_{ij} is the mean rate-of-rotation tensor viewed in a moving reference frame with the angular velocity ω_k. The model constants A_0 and A_s are given by following relation.

$$A_0 = 4.04;\ A_s = \sqrt{6Cos\emptyset}$$

Where

$$\emptyset = \frac{1}{3}\cos^{-1}\left(\sqrt{6W}\right);\ W = \frac{S_{ij}\,S_{jk}\,S_{ki}}{\tilde{S}^3};\ \tilde{S} = \sqrt{S_{ij}S_{ij}};\ S_{ij} = \frac{1}{2}\left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right)$$

It can be seen that C_μ is a function of the mean strain and rotation rates, the angular velocity of the system rotation, and the turbulence fields (K and Ɛ).

The constant for above model are given below

$$C_{1\varepsilon}=1.44,\ C_2=1.9,\ \sigma_k=1.0,\ \sigma_\varepsilon=1.2$$

2.4 Material

Water liquid is chosen as working fluid in this study. Table 8 Fluid (water) properties. These values were used for calculations. All domains are consisted of water liquid.

Table 8 Fluid (water) properties

Sr.No.	Property	Units	Method	Value
1.	Density	kg/m3	constant	998.20001
2.	Cp (Specific Heat)	j/kg-k	constant	4182
3.	Thermal Conductivity	w/m-k	constant	0.6
4.	Viscosity	kg/m-s	constant	0.001003
5.	Molecular Weight	kg/kgmol	constant	18.0152
6.	Thermal Expansion Coefficient	1/k	constant	0

2.5 Boundary Conditions

The flow assumes to be steady state, continuous and turbulent. Fluid is incompressible and has constant properties. Radiation effects considered negligible. Also no slip condition is set on wall.

2.5.1 Inlet Boundary Conditions

For flow in velocity inlet condition is considered. Turbulence intensity is provided 5 %, flow inlet, inlet wall, out wall temperatures kept constant at 298.15 [k]. while Test section wall has constant temperature of 323.15 [k]. Since the hydraulic diameter is 0.009203mm and 0.00893mm for TET and ETET respectively. So to attain the desired Reynolds number following velocities provided in Table 9 Inlet.were provided at inlet of all tubes.

Table 9 Inlet.

Reynolds No.	10000	20000	30000	40000	50000
TET	1.09	2.18	3.28	4.37	5.46
ETET	1.13	2.25	3.37	4.5	5.63

2.5.2 Outlet boundary Conditions

Outflow BC is applied at outlet. Conditions for out flow B.C. are as follows.

A zero diffusion flux for all variables at outflow cells mean that condition for boundary is extrapolated from within the domain and have no impact on upstream flow. Extrapolation carried out is consistent with fully developed flow state. And overall mass balance correction.

2.6 Solution Method

For Pressure-Velocity coupling SIMPLE scheme is chosen. Its segregated algorithm, in which a pressure relationship for velocity and pressure corrections is use to ensure the mass conservation and to obtain pressure field.

For spatial discretization for gradient green gauss cell based scheme, for pressure seconder order, for moment and energy equation second order upwind, for turbulent kinetic energy and turbulent dissipation rate first order upwind scheme is chosen.

2.7 Solution controls and Monitors

Default values for under relaxation factor are chosen which are the closest to optimal values. For pressure, density, body force, momentum, turbulent kinetic energy, turbulent dissipation rate, turbulent viscosity and energy have 0.3, 1, 1,0.7, 0.8 0.8, 1 and 1 respectively.

Residual monitor is set for continuity, velocity, energy. Kinetic energy and epsilon and absolute convergence stop criteria for all is 0.001 and for energy it is 0.000001.

Also some surface monitors for temperature, pressure at point 1 and 2 and average Nusselt number were created. Also stoppage criteria for all surface monitors is set to 0.001. First 10 iterations are set to be ignore. And pervious 10 iterations are set to consider. Hybrid initialization method is selected for initialization. 10 of iteration for scalars is set. 400 numbers of iteration are set and auto save is enable for every 50 iterations.

2.8 Data Reduction

Friction factor in tubes is calculated by following formula.

$$f = \frac{\Delta P D h}{\frac{1}{2} L \rho U 2} \; ; D_h = \frac{4A}{P}$$

Where ΔP, D_h, L, ρ *and* U is pressure drop, hydraulic diameter, length of test section and inlet velocity respectively. While hydraulic diameter is given below A and P area and perimeter respectively.

Flow inlet velocity is defined by using Reynolds Number formula given below where μ is dynamic viscosity. And Avg. Nussle number is calculated by following formula where k is the conductive heat co efficient of water.

$$Re = \frac{\rho U D_h}{\mu} \ ; \ \overline{Nu} = \frac{h D_h}{k}$$

Co efficient for convective heat transfer given below where $\dot{m}$, c_p, A_s, T_w, $T_{b,i}$ and $T_{b,o}$ are mass flow rate, specific heat co efficient, surface area, wall temperature, main stream inlet Temperature and main stream outlet Temperature respectively.

$$h = \left(\frac{\dot{m}c_p}{A_s}\right) ln\left(\frac{T_w - T_{m,i}}{T_w - T_{m,o}}\right)$$

Heat flux co efficient for convective heat transfer is can be calculated by following formula

$$q'' = \frac{\dot{m}Cp(T_{m,in} - T_{m,out})}{A_s} ; h_{avg} = \frac{q''}{\Delta T} \text{ Where } \Delta T = \frac{T_w - T_{m,in} - T_w - T_{m,out}}{ln\frac{T_w - T_{m,in}}{T_w - T_{m,out}}}$$

Entransy dissipation rate is found by

$$\dot{e} = \frac{1}{2}\dot{m}Cp(T_{m,in}^2 - T_{m,out}^2) - \dot{m}C_p(T_{m,in} - T_{m,out})\, T_w$$

Thermal Resistance can be found out by

$$R_h = \frac{\Delta T^2}{\dot{e}}$$

Thermal performance factor and thermal performance index is given below respectively

$$E_0 = \left(\frac{\overline{Nu}}{\overline{Nu}_0}\right)\left(\frac{\Delta P}{\Delta P_0}\right)^{-\frac{1}{3}} \ ; \ \eta = \frac{\left(\frac{h_{finned}}{h_{smooth}}\right)}{\left(\frac{f_{finned}}{f_{smooth}}\right)}$$

Where subscript “o” represent the smooth elliptical tube.

Chapter 3: Results and conclusions

3.1 Results and analyses

In this study two types of tubes were considered, fist one is simple smooth elliptical cross section tube and the second one with fins at major and minor axis. Each tubes have five different sub types depending on the twist pitch (details are given in Table 10 Test Matrix). Flow was studied numerically for Reynolds number 10000, 20000, 30000, 40000 and 50000. Performance evaluation is carried out relative to the simple smooth elliptical tube without any twist.

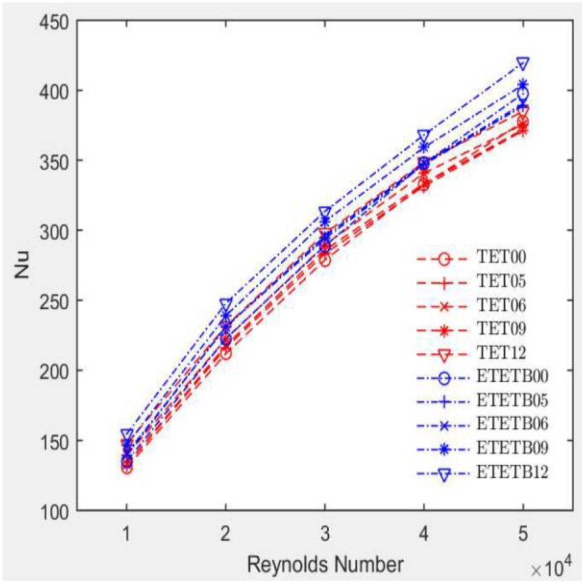
Table 10 Test Matrix

TET	Pitch [mm]	Aspect Ratio	ETETB	Pitch [mm]	Aspect Ratio
00	-	1.375	00	-	1.375
05	150	1.375	05	150	1.375
06	125	1.375	06	125	1.375
09	83.33	1.375	09	83.33	1.375
12	62.5	1.375	12	62.5	1.375

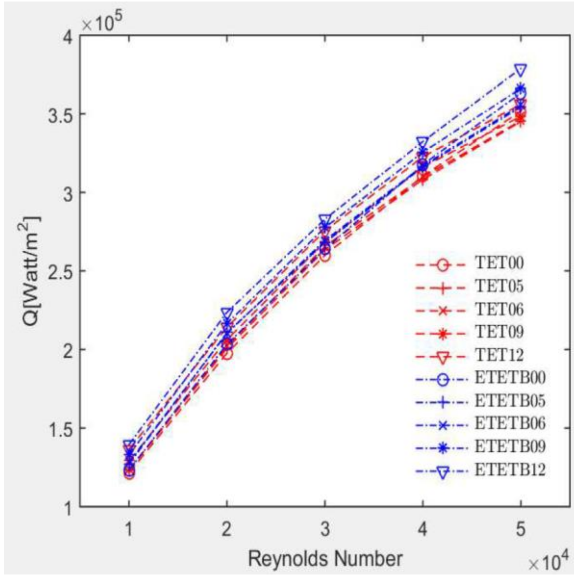
In present study results shows that average Nusselt number, hat flux, heat transfer co-efficient and logarithmic mean temperature difference increases with Reynolds number as shown in Figure 8 (a) Average Nusselte Number (b) Heat Flux (c) Heat Transfer Coefficient and (d) Temperature Difference VS Reynolds Number. ETETB with 62.5mm pitch has the highest vales for Average Nusselt number, heat flux and heat transfer coefficient.

Figure 9 (a) Temperature at point 02 (b) Thermal Resistance (c) hydraulic Friction and (d) Pressure drop against Reynolds number. shows that outlet temperature decreases with increasing Reynolds number. And thermal resistance also decreases with increasing Reynolds number. It is clear from the figure that ETETB 12 has the highest outlet temperature for each value of remolds number. ETETB 12, TET 09, ETETB 12, ETETB 12, ETETB 00 has the maximum friction value while TET 00, TET 05, ETETB 09, ETETB 09 and ETEB 06 has minimum friction value for Reynolds No. 10000, 20000, 30000, 40000 and 50000 respectively shown in Figure 9 (a) Temperature at point 02 (b) Thermal Resistance (c) hydraulic Friction and (d) Pressure drop against Reynolds number. ETETB 12 show maximum pressure drop for Reynolds number ranges from 10000 to 40000. While ETETB 00 has maximum pressure drop for

Remolds 50000 among all the tubes. TET 00, TET 05, TET 05, TET 12 and TET 09 shows minimum pressure drop for Reynolds number 10000, 20000, 30000, 40000 and 50000 respectively as shown in Figure 9 (a) Temperature at point 02 (b) Thermal Resistance (c) hydraulic Friction and (d) Pressure drop against Reynolds number.



(a)



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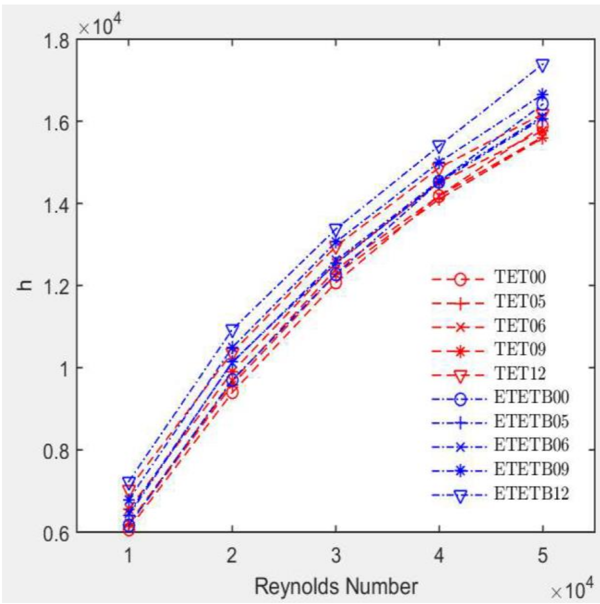
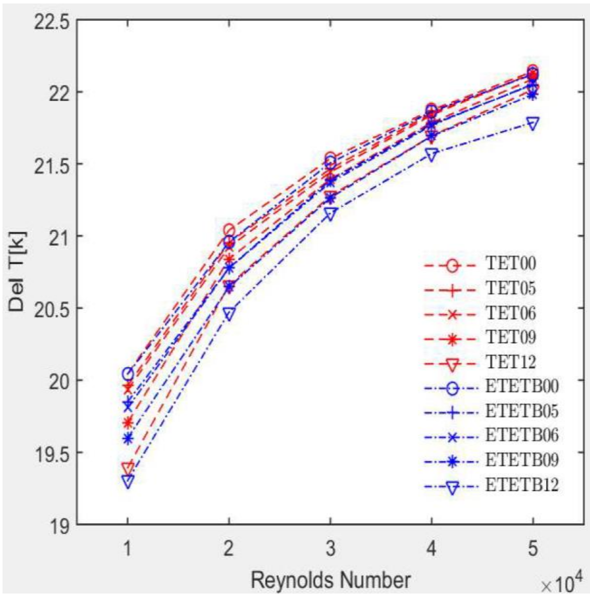
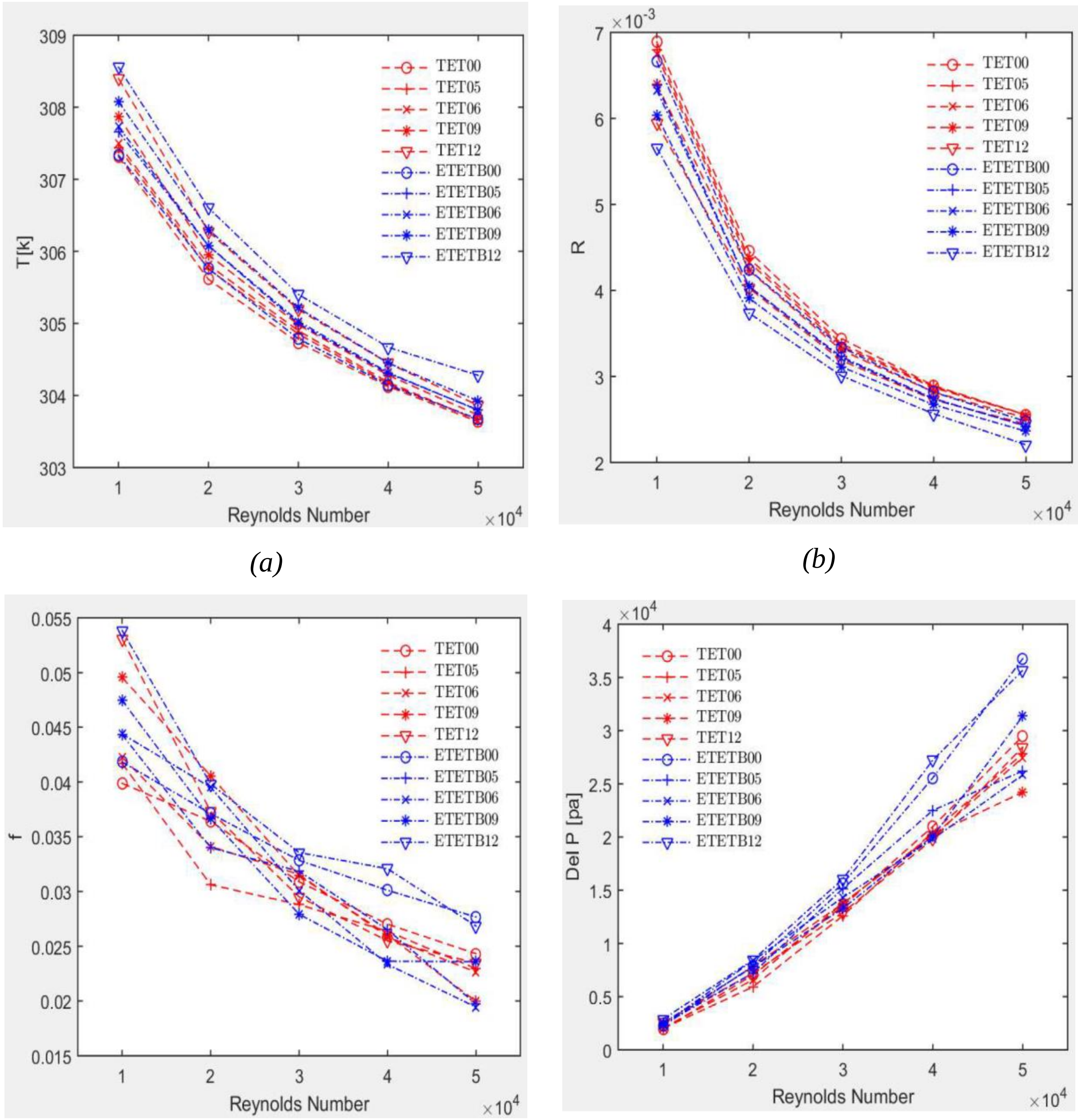


(c)



(d)

Figure 8 (a) Average Nusselte Number (b) Heat Flux (c) Heat Transfer Coefficient and (d) Temperature Difference VS Reynolds Number



(a) *(b)*

(c) *(d)*

Figure 9 (a) Temperature at point 02 (b) Thermal Resistance (c) hydraulic Friction and (d) Pressure drop against Reynolds number.

Following Figure 10 Ratio of average Nusselte number for enhanced to smooth tube. shows the caparison of all tube with respect to the smooth elliptical smooth. ETET 12 has the best thermal performance for Re. No. 10000 to 50000. For increasing Reynolds number thermal performance decreases till Re No. 40000. Then it starts increasing with further increase in Reynolds No. Till 30000 Reynolds number each tube sows better thermal performance than smooth tube. For Reynolds 40000 and onward TET 05 and 06 shows worse results. TET 09 shows worse performance than Smooth tube for Reynolds number 50000.

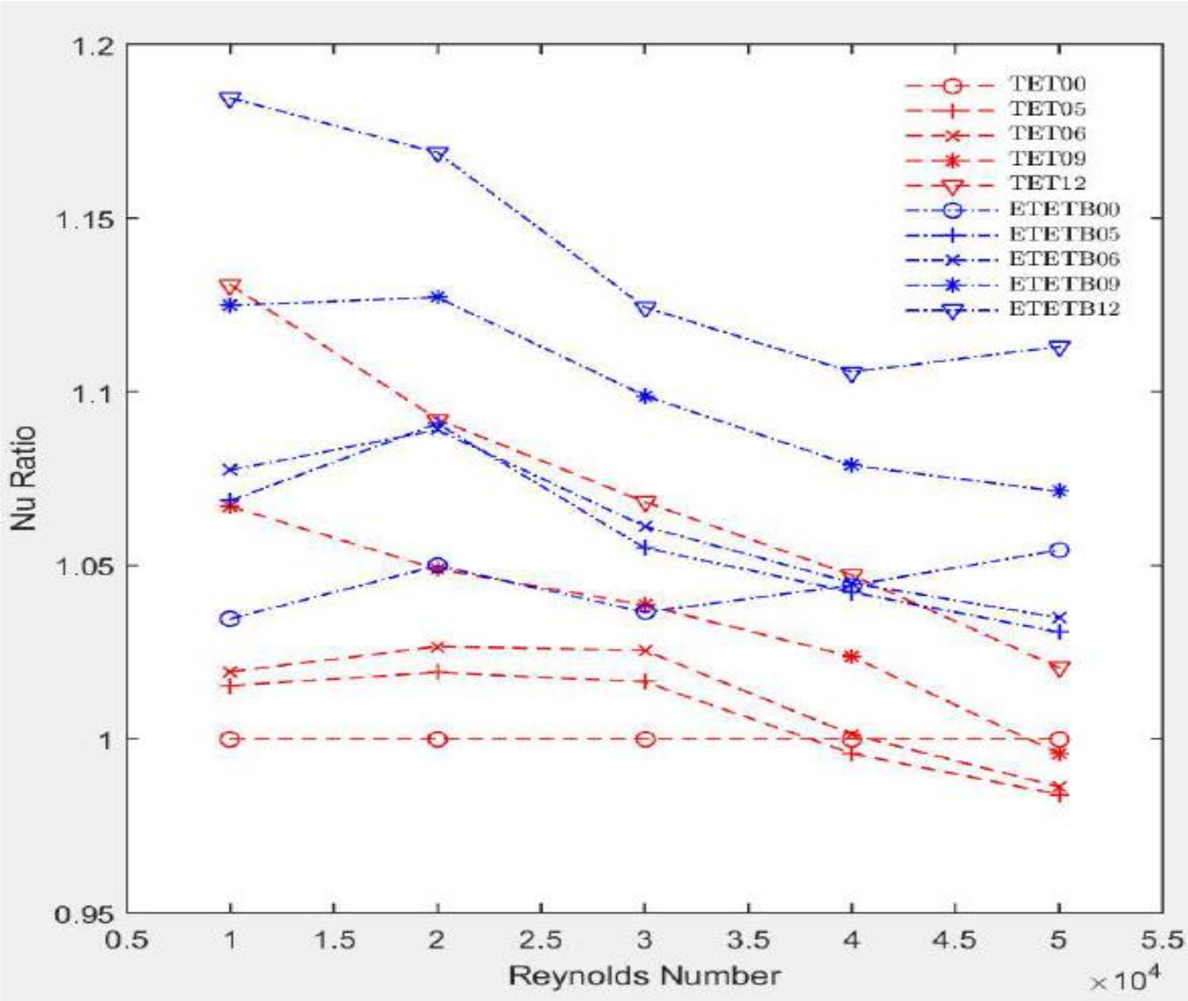


Figure 10 Ratio of average Nusselte number for enhanced to smooth tube.

ETET 12 also show the highest pressure drop ratio with respect to smooth tube till 40000 Re. No. For 50000 Reynolds number ETETB 00 has highest pressure drop ratio. For Re. No. 20000 TET 05 for Re. No. 30000 TET 05 for Re. No. 40000 TET 12 and for Re. No. 50000 TET 09 has the lowest pressure drop ratio. For TET 09 and ETETB 06 pressure drop ratio decreases with increasing Re. No. shown in following Figure 11 Pressure drop ratio of enhanced to smooth tube.

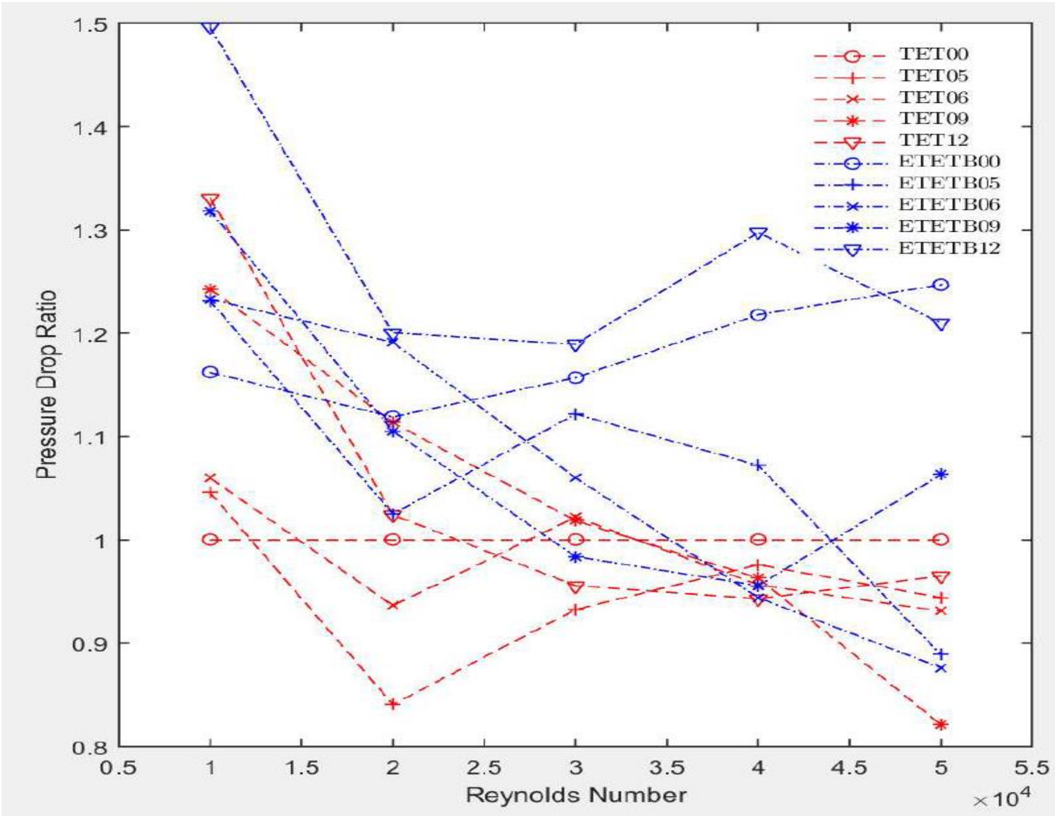


Figure 11 Pressure drop ratio of enhanced to smooth tube.

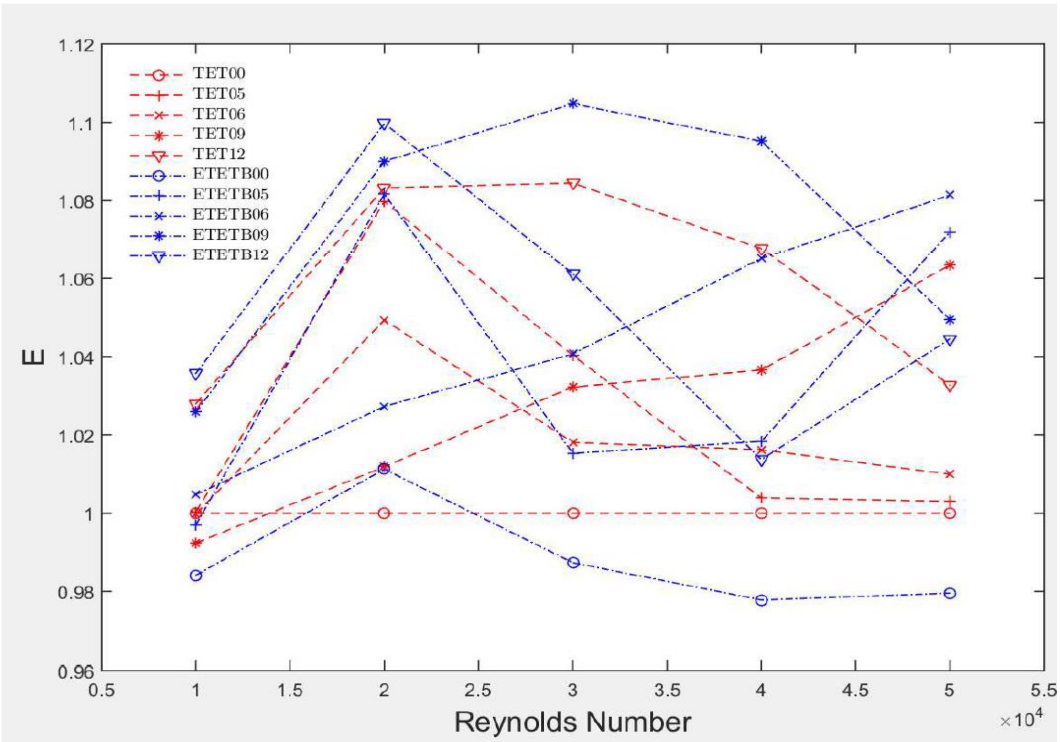


Figure 12 Thermal Hydraulic performance relative to smooth elliptical tube.

The above Figure 12 Thermal Hydraulic performance relative to smooth elliptical tube. indicates the thermal hydraulic performance of all tubes considered in the present study. It is clear from figure that ETETB 12 shows best thermal hydraulic performance for Reynolds numbers 10000 and 20000. ETETB 9 shows best performance for Reynolds number 30000 and 40000. While ETETB 6 shows best performance for Re. No. 50000. While ETETB 00 Shows worse performance compare to the other tubes expect for Reynolds number 20000 where TET 00 shows the worse performance. For Reynolds number 1000 TET 09 and ETETB 05 show worse performance than TET 00.

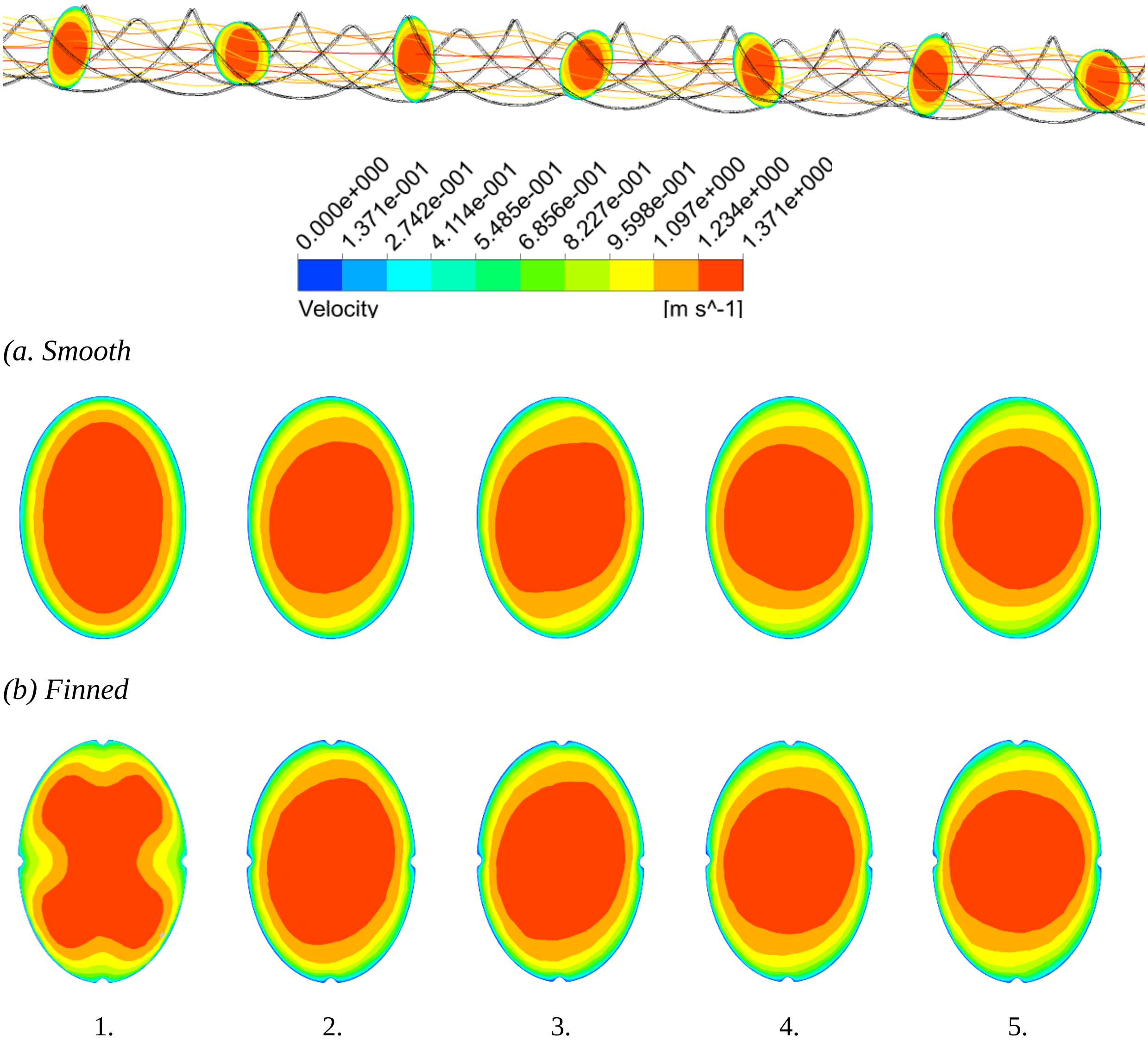


Figure 13 stream lines and velocity distribution at middle of tube 1) for smooth Tube and for twisted tube with pitch 2) 150 mm 3) 125mm 4) 83.33 mm 4) 62.5 mm respectively

Figure 13 stream lines and velocity distribution at middle of tube 1) for smooth Tube and for twisted tube with pitch 2) 150 mm 3) 125mm 4) 83.33 mm 4) 62.5 mm respectively shows the velocity distribution in ETETB and stream lines. And depict the x velocity component at the middle of test section of all tubes. Velocity at the center of tube is higher as secondary flow increases with twist pitch decreases which increase the heat transfer. Velocity at the center of tube increases as number of twist increases in tube. **Error! Reference source not ound.**describe the y velocity component for ETETB 12 at different sections from inlet to outlet. Fluctuation in transverse velocity in-between minimum to maximum increase. This secondary flow enhance the heat transfer as mixing increases.

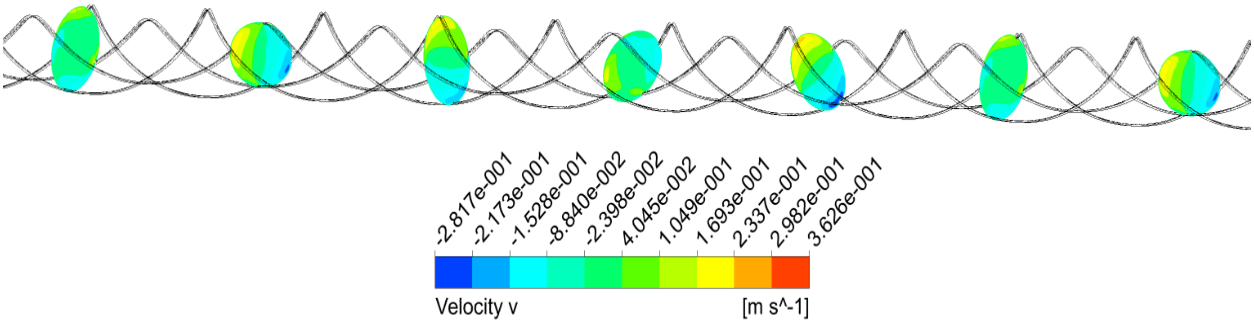


Figure 14 Secondary flow velocity distribution.

1)

2)

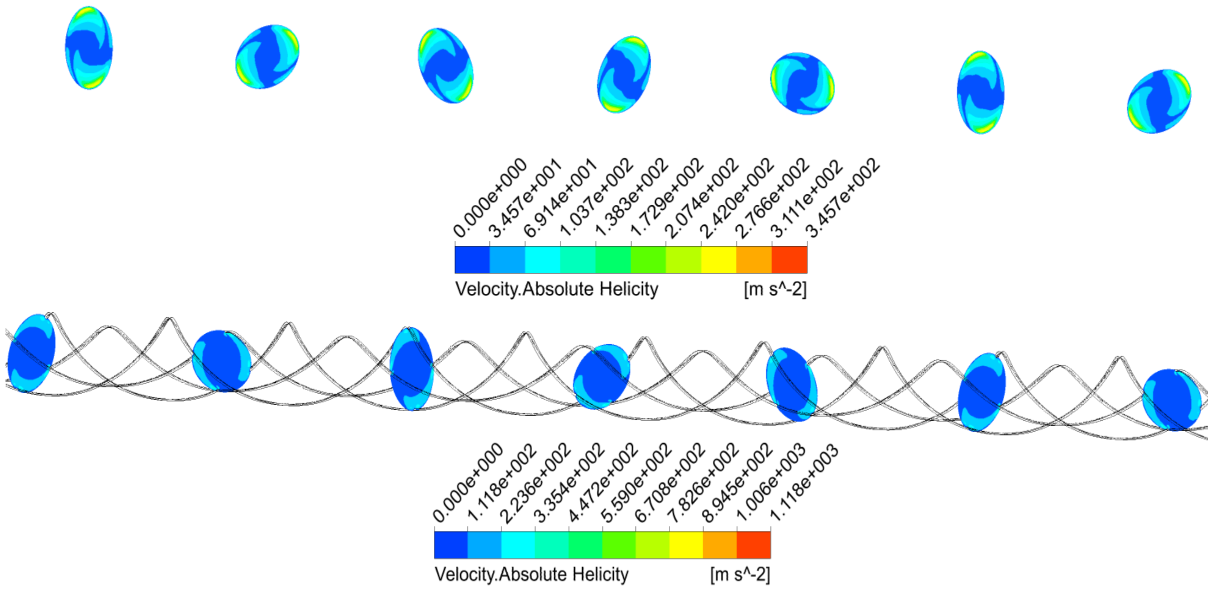
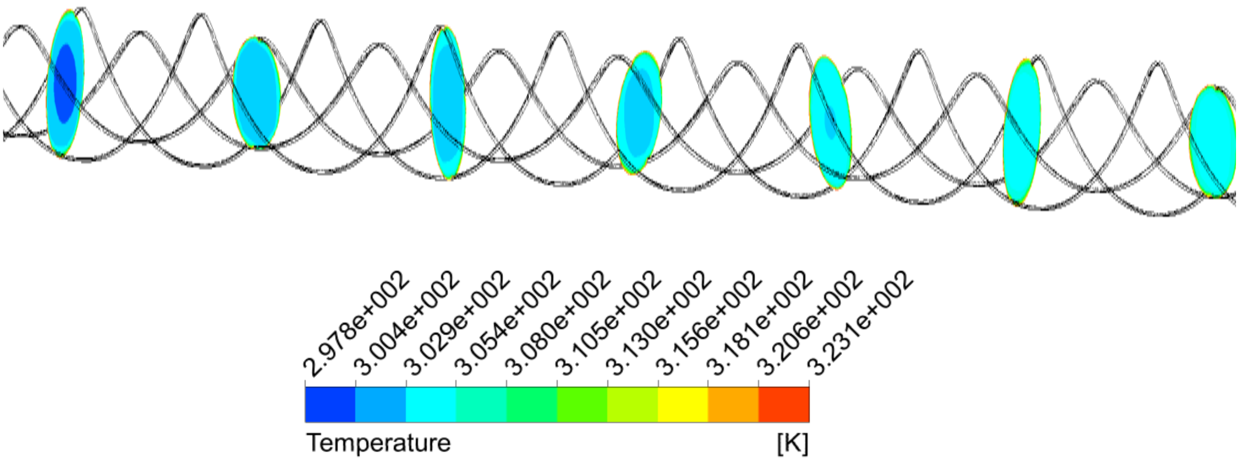


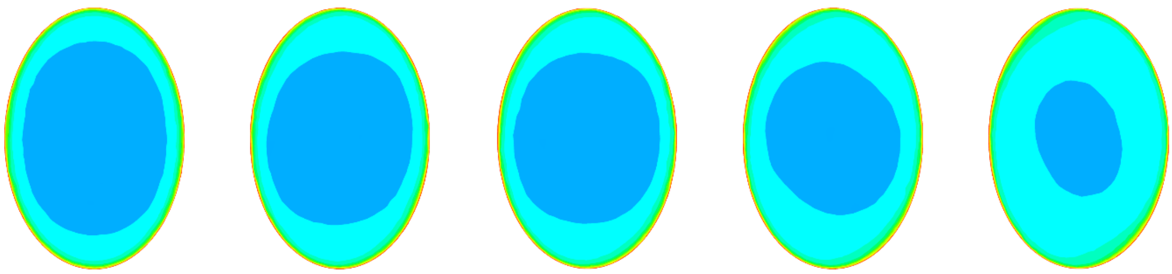
Figure 15 velocity absolute helicity 1)TET 2)ETETB

Above Figure 15 velocity absolute helicity 1)TET 2)ETETB shows that velocity absolute helicity is higher in ETETB 12 than TET 12. Which is scalar and the dot product of the velocity and vorticity.

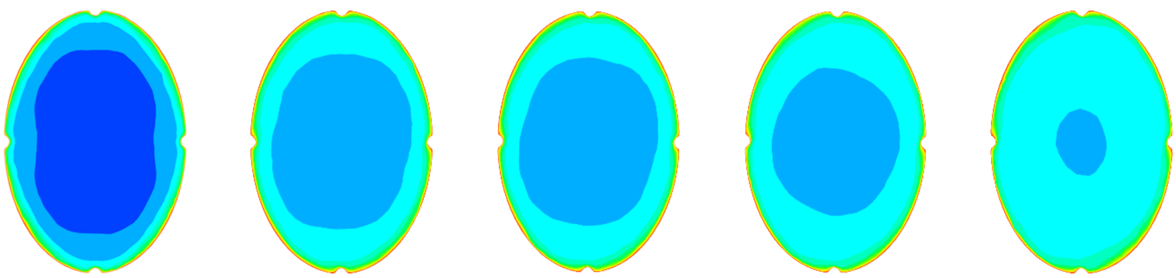
The following contours showed in Figure 16 *Temperature distribution at middle of tube*at the temperature distribution at the center of test section of all tubes at Reynolds no 10000. It is clear for the figure as twist pitch decreases the temperature distribution changes significantly. Variation in temperature distribution in TET starts early by changing pitch but its intense in enhanced tube. Twisted wall produce secondary flow and cold fluid moves from core of tube the outer hot surface.



(a. Smooth



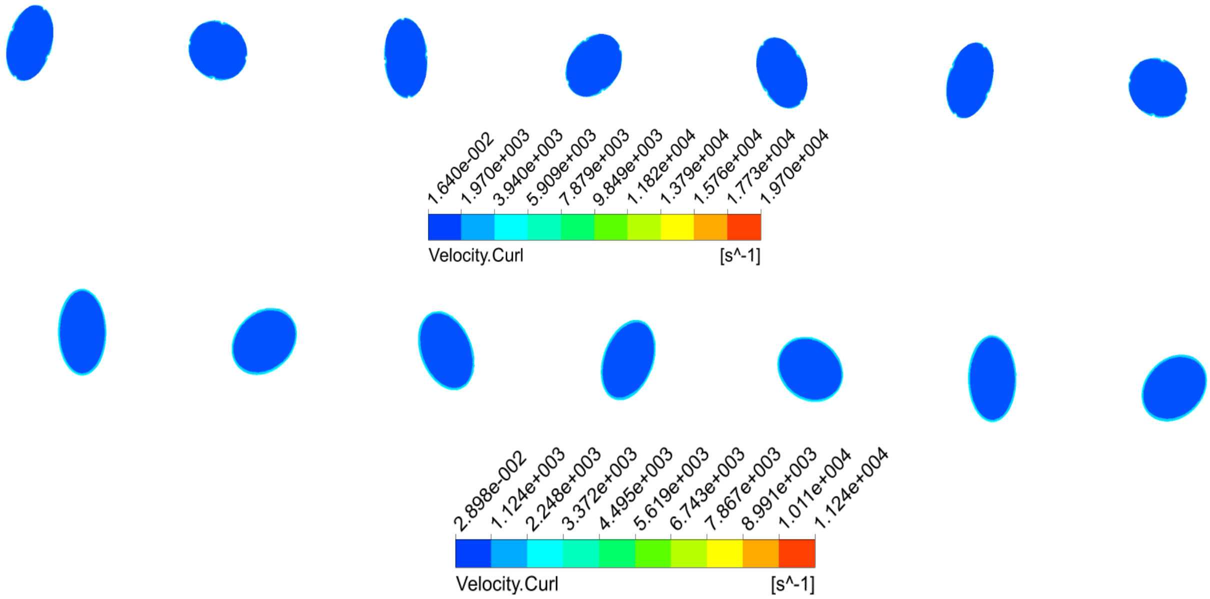
(b) Finned



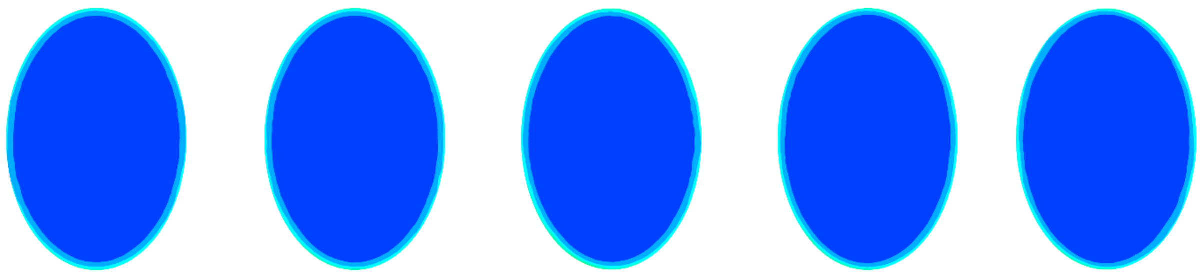
1. 2. 3. 4. 5.

Figure 16 *Temperature distribution at middle of tube 1) for smooth Tube and for twisted tube with pitch 2) 150 3) 125 4) 83.33 4) 62.5 mm respectively.*

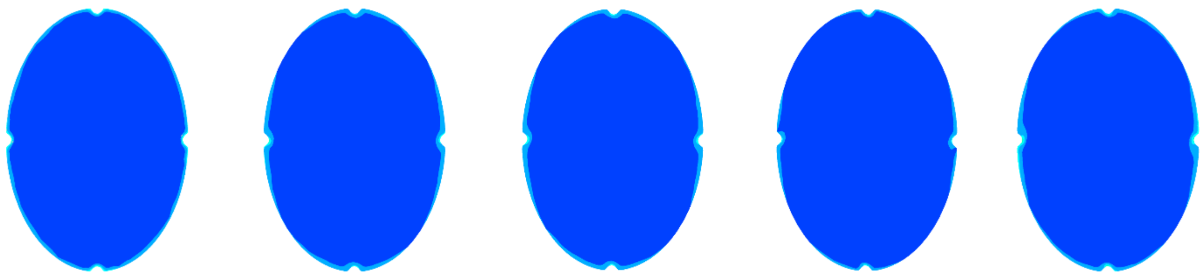
Figure 17 vorticity in ETETB and TET at different sections and at middle of tube 1) for smooth Tube and for twisted tube with Pitch 2) 150 mm 3) 125 mm 4) 83.33 mm 4) 62.5 mm shows the vorticity (velocity curl) curl in ETETB and TET at different sections and also a the middle of each tube.



(a. Smooth



(b) Finned

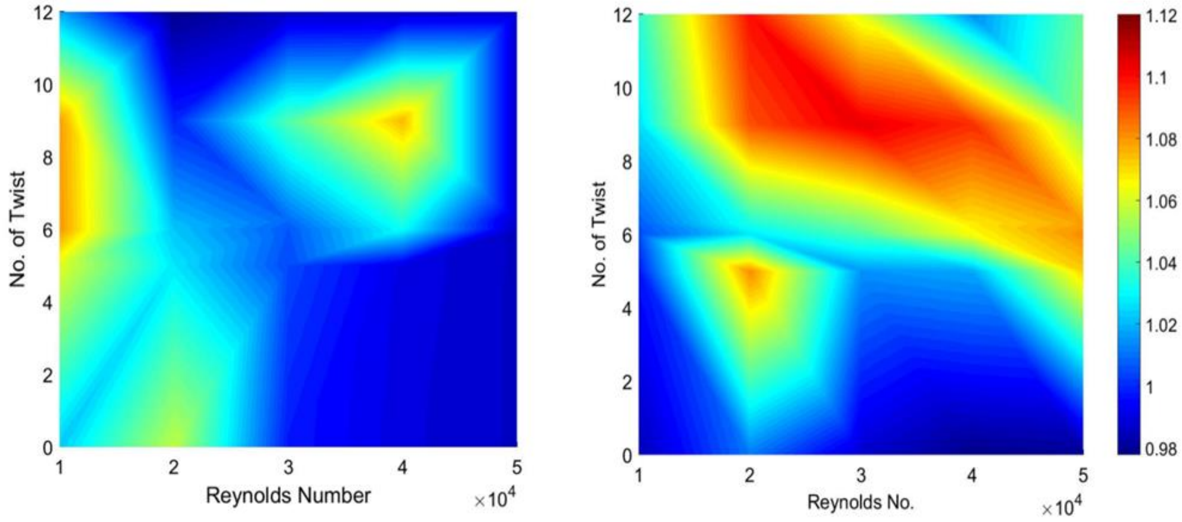


1. 2. 3. 4. 5.

Figure 17 vorticity in ETETB and TET at different sections and at middle of tube 1) for smooth Tube and for twisted tube with Pitch 2) 150 mm 3) 125 mm 4) 83.33 mm 4) 62.5 mm

3.2 Conclusion and discussion

This study is carried out to find the thermal and hydraulic behavior of fluid flow in twisted elliptical tube and enhanced twisted tube numerically. The effect of twist pitch, introducing fins and Reynolds number were studied.



(a) (b)

Figure 18 Thermal Hydraulic Performance Contour for (a) TET and (b) ETETB

Results in Figure 18 Thermal Hydraulic Performance Contour for (a) TET and (b) ETETB**Error! Reference source not found.** shows that Pressure drop increase with increasing Reynolds number. ETETB 12 with 62.5 mm has highest value for Reynolds number 40000 and less. While ETETB 00 has highest pressure drop for Reynolds number 50000. For Re. No. 20000 and 30000 TET 05 for Re. No. 40000 TET 12 and for Re. No. 50000 TET 09 has the lowest pressure drop. Pressure drop ratio varies from 0.82 to 1.5 relative to smooth tube. Pressure drop increase to twisted wall.

Average Nusselte number increases with increasing Reynolds number. ETETB 12 has the highest value of average Nusselte number against each Reynolds number. On the basis of average Nusselte no thermal performance varies from 1.18 to 0.98 comparative to smooth tube.

ETETB 12 shows best thermal hydraulic performance for Reynolds numbers 10000 and 20000. ETETB 9 shows best performance for Reynolds number 30000 and 40000. While ETETB 6 shows best performance for Re. No. 50000. This is due to decrease in severity of pressure drop with higher Reynolds number for lower value of pitch.

Index A

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Uncategorized References

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