International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 2 Issue 8, August - 2013

CFD Analysis of Enhancement of Turbulent Flow Heat Transfer in a Horizontal Tube with Different Inserts

Ramesh Vathare [1*], Omprakash Hebbal[2]

[1*]PG Student, Thermal power Engineering, PDA College of Engineering, Gulbarga-585102, Karnataka (INDIA)

[2]Professors, Department of Mechanical Engineering, PDA College of Engineering, Gulbarga-585102, Karnataka (INDIA)

ABSTRACT

The present work includes the results of CFD analysis of enhancement of turbulent flow heat transfer in a horizontal circular tube with different inserts (Cylinder, diamond and trapezoidal), with air as working fluid. The Reynolds number ranged from 6000 to 14000. Geometry of tube having inner diameter 27.5 mm and length of the tube is 610 mm. The horizontal tube in the presence of different inserts: (Geometry description of diamond: pitch=50mm, core rod diameter =2mm, diagonal length =20mm, thickness=2mm. Geometry description of cylinder: pitch=50mm, core rod diameter =2mm, diameter of cylinder =20mm, thickness of cylinder=2mm.Geometry description of trapezoidal: pitch=50mm, core rod diameter =2mm, bottom length=15mm, top length=10mm, height=10mm). Improvement of average Nusselt Numbers for tube with 87% for cylinder insert, 85% for diamond insert and 28% of trapezoidal insert, with respect to Plain Tube. Similarly friction factor for tube with cylinder insert is 600% more compared with the plain tube. Overall enhancement ratio is high for cylinder (75%) and low for Trapezoidal insert (65%) and pressure drop is considerably more with different inserts when compared to plain tube, pressure drop is high for cylinder insert and low for trapezoidal insert. Finally we compared results with theoretical results and using tool of package of ANSYS-CFX 12.0 version 12.0 version to compare the construction, performance, and economics of tube inserts. Geometries for plain tube and tube with different inserts is developed and meshing in ICEM CFD-13.0 (3d-dimensional) with and exported to ANSYS-CFX 12.0 version12.0 version, then suitable boundary conditions are applied to these models and solved energy momentum and turbulence equations and results obtained are discussed.

Key words : tube, heat transfer, different inserts (Cylinder, diamond and trapezoidal), turbulent flow, pressure drop, augmentation. ICEM-CFD13.0 version software, ANSYS-CFX 12.0 version12.0 version Software,

1 introduction

Heat transfer enhancement or intensification is the study of improved heat transfer performance. Recently adequate energy source and material costs have provided significant resources for the development of enhanced energy efficient heat exchangers. As a result, considerable emphasis has been placed on the development of various augmented heat transfer surfaces and devices. Heat transfer enhancement today is characterized by rigorous research activities both in academic and industrial levels. This can be seen from the exponential increase in world technical literature published on heat transfer enhancement devices, growing patents and hundreds of manufacturers offering products ranging from enhanced tubes to entire thermal systems incorporating enhancement technology. Enhancement of turbulent flow heat transfer in a horizontal circular tube with different inserts has drawn great attention, and some experimental and theoretical research work has been carried out in recent years.

Shou-ShingHsieh ,Feng-Yu Wu,Huang-Hsiu Tsai et al. [1]: The present result of a study of turbulent flow and pressure drop in a horizontal tube with strip type inserts. Experimental data taken for air for a class of strip type inserts (longitudinal, LS and cross, CS inserts) used as a tube side heat transfer augmentative device for a single-phase cooling mode operation are presented. To broaden the understanding of the underlying physical phenomena responsible for the heat transfer enhancement,flow mechanisms through velocity measurements are combined with pressure drop measurements to develop friction factor correlations for 65006 Re 619500 where Re is the Reynolds number. Friction factor increases were typically between 1.1 and 1.5 from low Re 6500 to high Re 19500 with respect to bare tubes.

M Ahmed, L Deju, M. A. R. Sarkar and S. M. Nazrul Islam et al. [2]: Heattransfer and pressure drop characteristics in a circular tube fitted with twisted tape inserts have been investigated experimentally. Experiments were conducted with tape inserts of three different twist ratios (twist ratio, y=23, y=11.5 and y=8).And calculated for tubes with twisted tape inserts to analyze the friction factor, Nusselt Number and the heat transfer coefficient. Reynolds Number was calculated based on inside diameter of the tube and varied from 2.0 × 104 to 5.5 × 104. The result indicated that average heat transfer coefficient is about 1.3 to 3 times higher than that of the smooth tube.

M A R Sarkar, A B M ToufiqueHasan, M Ehsan, M MAlamTalukdar and A MA Huq et al. [3]: An experimental investigation has been carried out to study theconvective heat transfer in a tube with longitudinal inserts with different shapes of strips in turbulent flow. Three different shapes of strip (Y, X and star- configuration) were fabricated from mild steel and inserted into the smooth tube successively. And the flow was varied in the range of Reynolds number 2.0x104 to 5.0x104. At comparable Reynolds number, heat transfer co-efficient in tube with longitudinal strip inserts is enhanced by 1.4 to 3 times, friction factor increased by 1.2 to 2.2 times while the pumping power increased up to 4 times compared to that of smooth tube.

Hiral N. Prajapati et al. [4]: Present work reports CFD studies of Nusselt number,friction factor and overall enhancement efficiency for a tube fitted with wire coil. The result showed that the wire coil with p/d of 0.434, 0.651 and 0.868 can enhance heattransfer up to 4.98, 5 and 4.3 times respectively and friction factors up to 5.82, 4.06 and 3.3 times respectively, in comparison with plain tube. Wire coil inserts with e/d of 0.038, 0.128 and 0.171 provide heat transfer enhancement around 4.99, 5.92 and 7.6 times respectively and friction factor enhancement up to 5.82, 19.97 and 35.7 times respectively than plain pipe.

S.N. Sarada, A.V.S.R. Raju and K.K. Radha et al. [5]: The present work focuseson experimental and numerical investigations of the augmentation of turbulent flow heat transfer in a horizontal circular tube by means of mesh inserts with air as the working fluid. Sixteen types of mesh inserts with screen diameters of 22 mm, 18 mm, 14 mm and 10 mm for varying distance between the screens of 50 mm, 100 mm, 150 mm and 200 mm in the porosity range of 99.73 to 99.98 were considered for experimentation. The horizontal tube was subjected to constant and uniform heat flux. The Reynolds number varied from 7,000 to 14,000. The results are compared with the clear flow case when no porous material was used. Computational fluid dynamics (CFD) techniques were also employed to perform optimization analysis of the mesh inserts. The horizontal tube along with mesh inserts was modelled in Gambit 2.2.30 with fine meshing and analyzed using FLUENT 6.2.16. CFD analysis was performed initially for plain tube and the results are compared with experimental values for validation.

2 Methodology.

2.1Mathematical Model

Finite Volume Formulation

ANSYS-CFX 12.0 version uses a finite-element-based finite volume method. The governing equations namely the conservation for mass; momentum and energy are expressed in equations (2.1, 2.2 and 2.3).

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_j}\left(\rho u_j\right)=0 \tag{2.1}$$

$$\frac{\partial}{\partial t}\left(\rho u_i\right)+\frac{\partial}{\partial x_j}\left(\rho u_j u_i\right)=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu_{eff}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right]+S_{ui} \tag{2.2}$$

$$\frac{\partial}{\partial t}\left(\rho \phi\right)+\frac{\partial}{\partial x_j}\left(\rho u_j \phi\right)=\frac{\partial}{\partial x_j}\left[\Gamma_{eff}\left(\frac{\partial \phi}{\partial x_j}\right)\right]+S_{\phi} \tag{2.3}$$

Where the energy conservation has been replaced by a generic scalar transport equation. The finite volume method proceeds by integrating these equations over a fixed control volume, which, using Gauss Theorem, results in equation (2.4, 2.5 and 2.6).

$$\frac{\partial}{\partial t}\int_v \rho dv + \int_s \rho u_j dn_j = 0 \qquad (2.4)$$

$$\frac{\partial}{\partial t}\int_v \rho u_i dv + \int_s \rho u_j u_i dn_j = -\int_s P dn_i + \int_s \mu_{eff}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) dn_j + \int_v S_{u_i} dv \qquad (2.5)$$

$$\frac{\partial}{\partial t}\int_v \rho \phi dv + \int_s \rho u_j \phi dn_j = \int_s \Gamma_{eff}\left(\frac{\partial \phi}{\partial x_j}\right) dn_j + \int_v S_\phi dv \qquad (2.6)$$

where v and s denote volume and surface integrals respectively and dn_j represents the differential Cartesian component of the outward normal surface vector. The equations represent a flux balance in a control volume. The above equations are applied to each control volume or cell in the computational domain. These continuous equations are approximated numerically using discrete functions. Discretisation of the above equations yields the following in equations (2.7, 2.8 and 2.9).

$$\rho V\left(\frac{\rho - \rho^o}{\Delta t}\right) + \sum_{ip}\left(\rho u_j \Delta n_j\right)_{ip} = 0 \qquad (2.7)$$

$$\rho V\left(\frac{u_i - u_i^o}{\Delta t}\right) + \sum_{ip} m_{ip}(u_i)_{ip} = \sum_{ip}(P\Delta n_i)_{ip} + \sum_{ip}\left(\mu_{eff}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\Delta n_j\right)_{ip} + \bar{S}_{u_i} V \qquad (2.8)$$

$$\rho V\left(\frac{\phi_i - \phi^o}{\Delta t}\right) + \sum_{ip} m_{ip}\phi_{ip} = \sum_{ip}\left(\Gamma_{eff}\frac{\partial \phi}{\partial x_j}\Delta n_j\right)_{ip} + \bar{S}_\phi V \qquad (2.9)$$

Where

$$m_{ip} = (\rho u_j \Delta n_j)^o_{ip}$$

and V is volume of the control volume, the subscript ip denotes the integration point, the summation is overall the integration points of the surface,$\Delta n_j$is the discrete outward surface vector,Δt is the time step, the superscript to refers to the old time level, and the overbar on the source terms indicate an average value for the control volume. The fluxes are evaluated at integration points, which are shared by adjacent control volumes and exactly the same flux that leaves one control volume enters the next.

The Standard k-ε Turbulence Model: The k-ε turbulence model utilizes the eddy-viscosity assumption to relate the Reynolds stress and turbulent flux terms to the mean flow variables. For the general Reynolds stress tensor is given by the equations (2.10 and 2.11).

Continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho \bar{U}_j) = 0 \qquad (2.10)$$

Momentum equation

$$\frac{\partial}{\partial t}(\rho \bar{U}_i) + \frac{\partial}{\partial x_j}\left(\rho \overline{U_i U_j}\right) = -\frac{\partial P^*}{\partial x_i} + \frac{\partial}{\partial x_j}\left\{\mu_{eff}\left[\frac{\partial \bar{U}_i}{\partial x_j} + \frac{\partial \bar{U}_j}{\partial x_i}\right] - \frac{2}{3}\mu_{eff}\left(\frac{\partial \bar{U}_i}{\partial x_i} + \frac{\partial \bar{U}_j}{\partial x_j} + \frac{\partial \bar{U}_k}{\partial x_k}\right)\delta_{ij}\right\} + \rho f_i \quad (3.19) \qquad (2.11)$$

Here $\mu_{eff} = \mu + \mu_t$ μ_t= turbulent viscosity and $P^* = P + 2/3\rho k$
Turbulent viscosity is related to turbulent kinetic energy k and
dissipation rate ε a sin equation 2.12

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon} \qquad (2.12)$$

The value so f k and ε come directly from the differential, transport equations for turbulent kinetic energy and turbulent dissipation rate as in equations (2.13 and 2.14).

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho \bar{U}_j k)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\Gamma_k \frac{\partial k}{\partial x_j}\right)+P_k-\rho\varepsilon \tag{2.13}$$

$$\frac{\partial(\rho \varepsilon)}{\partial t}+\frac{\partial(\rho \bar{U}_j \varepsilon)}{\partial x_j}=\frac{\partial}{\partial x_j}\left(\Gamma_\varepsilon \frac{\partial \varepsilon}{\partial x_j}\right)+\frac{\varepsilon}{k}(c_{\varepsilon 1}P_k-\rho c_{\varepsilon 2}\varepsilon) \tag{2.14}$$

Where diffusion coefficients are given by

$$\Gamma_k=\mu+\frac{\mu_t}{\sigma_k}$$

$$\Gamma_\varepsilon=\mu+\frac{\mu_t}{\sigma_\varepsilon}$$

Where$\sigma_k$and $\sigma_\varepsilon$are model constants. Production rate of turbulent kinetic energy P_k is given by equation 2.15.

$$P_k=\mu_t\left(\frac{\partial \bar{U}_i}{\partial x_j}+\frac{\partial \bar{U}_j}{\partial x_i}\right)\frac{\partial \bar{U}_i}{\partial x_j}-\frac{2}{3}\left[\rho k+\mu_t\left(\frac{\partial \bar{U}_i}{\partial x_i}+\frac{\partial \bar{U}_j}{\partial x_j}+\frac{\partial \bar{U}_k}{\partial x_k}\right)\right]\frac{\partial U_k}{\partial x_k} \tag{2.15}$$

The values of constants in the model are c_μ= 0.09, $c_{\varepsilon 1}$= 1.44, $c_{\varepsilon 2}$=1.92,σ_k=1.0,σ_ε= 1.3 and
Turbulent Prandtl number Pr_t = 0.9

2.2 Sequence of operation:

Description of the problem and geometry: The primary goal of the present work is to enhance the heat transfer in a tube employing various inserts. Also determine average Nusselt number and friction factors for Reynolds number ranging from 6000-14000 in the turbulent region. Average Nusselt numbers, friction factors of working fluid (air) flowing in the plain tube are compared with average Nusselt numbers, friction factors of working fluid (air) flowing in tube with diamond, cylinder and trapezoidal inserts which enhance heat transfer. Enhancement Efficiencies of the different inserts also compared.

The geometry of the tube and dimensions of the insert used in this study are listed:
Inner Diameter of the test pipe, D = $2.75*10^{-2}$ m
Cross-sectional area of the pipe, A_p = $5.939572*10^{-4}$ m^2
Length of the Heating Zone, L = $61*10^{-2}$m
Diameter of the Orifice, d_0 = 0.014m
Cross-sectional area of the Orifice, A_o = $1.54*10^{-4}$ m^2
Coefficient of discharge, C_d = 0.64 Thickness of the insert, t=2 mm, Average wall Temperature, Tw

Nusselt numbers and friction numbers are calculated for Reynolds number Ranging from 6000-14000.Equations for calculating Nusselt number & friction factor are given below:
$Nu=hD_i/k$
$h= ((Q/A)/(T_w-T_b))$
Where T_w= Average Surface Temperature
$T_b= (T_i+T_0)/2$
$f= (\Delta P)/(L/D)\rho V/2)$
Enhancement Efficiency is calculated using the following equation:
Friction factor ratio = $(Nu/Nu_0)/(f/f_o)^{1/3}$

Theoretical calculations:
Nusselt number and friction factor are calculated for Reynolds numbers ranging from 6000-14000 using **DITTUS-BOELTER** results are tabulated below:

DITTUS-BOELTER for Nusselt number:

Nu=0.023 x $Re^{0.8}$ x Pr^{n}for6000≤Re≤14000 , Where n = 0.4 for Heating fluids.
Friction factor:f= $[1.82 \log10 Re-1.64]^{-2}$for6000≤Re≤1400

Analysis of problem : CFD techniques used to perform the overall performance and optimization analysis of the fluid flow transfer of the tube with/without insert was performed using ANSYS-CFX 12.0 version.

Without inserts: Initially the CFD experiment is carried out with air as the moving fluid through pipe section without any inserts. This also termed as PLAIN TUBE CFD experiment.

With insert: In this type case is taken: Trapezoidal, Diamond and Cylinder.

Numerical Investigations in Plain Tube:

- Initially numerical analysis was performed on the horizontal plain tube for the estimation of Nusselt number and friction factor of air by using commercially available ANSYS-ICEM-13.0version, and ANSYS-CFX 12.0 version software.
- Nusselt numbers and friction factors of air obtained from the numerical analysis under turbulent flow are validated with the correlations available in the literature for tube internal flow.
- Conducting simulation at a constant heat flux of 759.010 w/m2 at different mass flow rates for calculation of Nusselt numbers, friction factors and pressure drop (across the test section of PLAIN TUBE CFD experiment) of air at different Reynolds numbers by varying the mass flow rate of air.
- Nusselt numbers and friction factors values obtained from the numerical analysis are compared with the correlations available in the literature and the percentage deviation is reported.

Numerical Investigations in Plain Tube with Inserts

- Simulation of horizontal tube test section (using numerical analysis) in the presence of different inserts like diamond, trapezoidal and cylinder to predict the enhancement of heat transfer rate and generation of pressure drop data across the test section by using commercially available ANSYS ICEM13.0version and ANSYSCFX12.0 version, software.
- Simulation of horizontal tube test section (using numerical analysis) in the presence of different inserts: (Geometry description of diamond - pitch=50mm, core rod diameter =2mm, diagonal length =20mm, thickness=2mm. Geometry description of cylinder - pitch=50mm, core rod diameter = 2mm, diameter of cylinder =20mm, thickness of cylinder=2mm. Geometry description of trapezoidal- pitch=50mm, core rod diameter = 2mm, bottom length=15mm, top length=10mm, height=10mm) were considered for numerical analysis. CFD techniques are employed to perform optimization analysis of the different inserts. The horizontal tube along with different inserts were modelled and meshing and analysed using ICEM-13.0version and ANSYS-CFX 12.0version, software.
- Numerical investigations are conducted in plain tube fitted with different inserts of circular, diamond and trapezoidal geometry at a constant heat flux of 759.010 w/m2 to find the Nusselt number and friction factor of air in the presence of different inserts. The diameter of the core rod is 2 mm. The distance between two adjacent insert (pitch) is fixed at 50 mm. The results of investigations using different inserts are compared with those of plain tube to estimate the rate of heat transfer enhancement of air in the presence of different inserts.
- Nusselt number ratio (Nui/Nu: Ratio of Nusselt number with insert to that of without insert) for all the inserts like diamond, trapezoidal and cylinder) is calculated.
- Friction factor ratio (fi/f: Ratio of friction factor with insert to that of without insert) for all the inserts like diamond, trapezoidal and cylinder) is calculated.
- Based on Nusselt number ratio and friction factor ratio, overall enhancement ratio is calculated individually for each insert.
- Overall enhancement ratio is calculated to determine the optimum geometry of inserts that could provide the maximum heat transfer enhancement rate with lesser friction factor.
- Based on oveall enhancement ratio, optimum insert geometry is recommended

ISSN: 2278-0181
Vol. 2 Issue 8, August - 2013

2.3 Model geometry, Boundary conditions and mesh generation

For setting up any CFD problem, the geometry has to be modelled with required details, mesh has to be generated optimally to obtain the results correctly and flow parameters and boundary conditions are to be set up for solving the problem. The discretized domain is solved using solver and results are analyzed in post processor. In the present investigation, CAD model and ANSYS-ICEM 13.0version software is used for the geometry modelling and mesh generation. ANSYS-CFX 12.0version, software is used for defining boundary conditions, solving and post processing.

Table 2.1:Boundarycondition specification for ANSYS-CFX 12.0version.

Zone	Type	Boundary Conditions
Tube	Wall	Heat flux = 759.010 w/m^2
Inlet temperature	Heat inlet	Temperature = 53.3^{0}c
Mass Flow rate	Inlet	0.00334 Kg/sec
Pressure	Outlet	101395.8 Pa

.

Fig. 2.2: Grid for the Tube with trapezoidal insert configuration
Grid Info:
Cells: 833004
Pitch = 50 mm
Core rod diameter = 2mm
Bottom length = 15 mm
Top length = 10 mm
Height = 10 mm
Thickness = 2mm

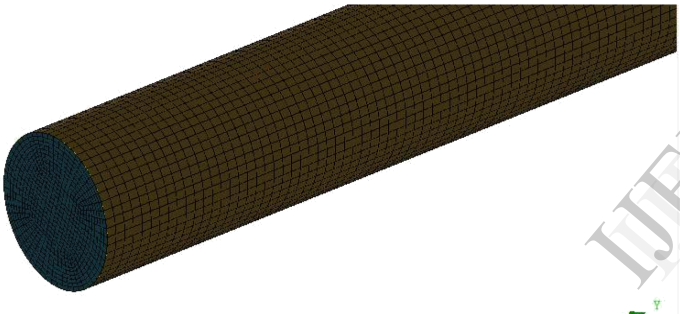


Fig. 2.1: Grid for the Plain Tube configuration

Grid Info: cells 232092

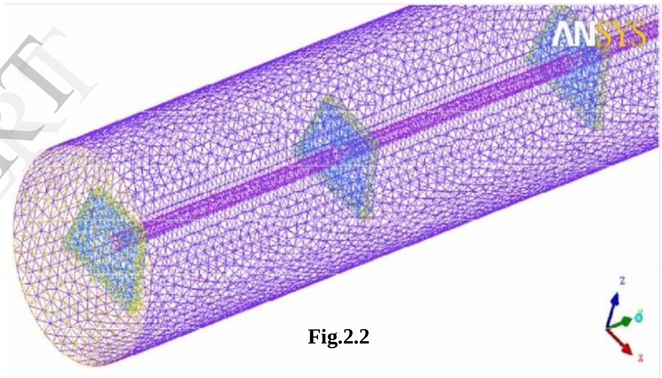
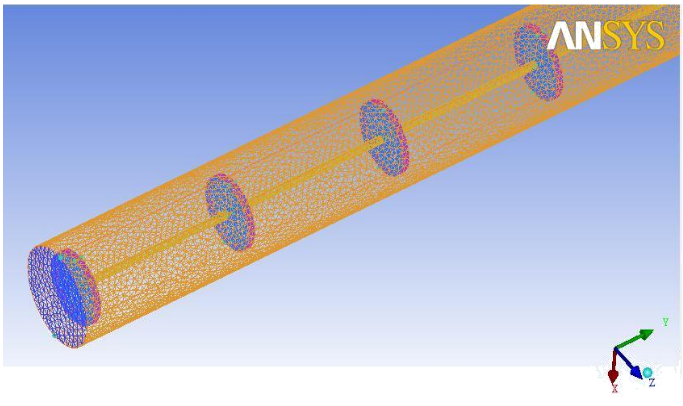


Fig.2.2



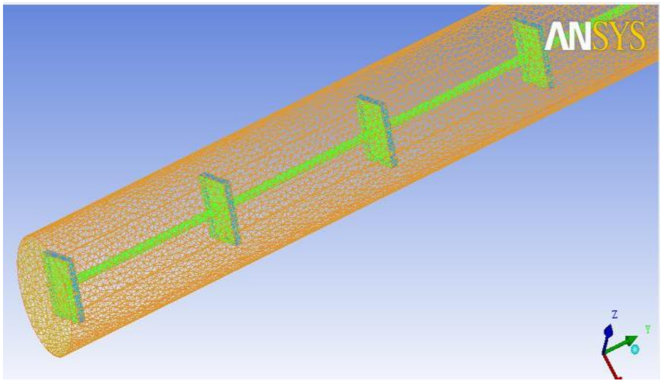


Fig.2.3: Grid for the Tube with cylinder insert configuration **Fig2.4: Grid for the tube with diamond insert configuration**

Grid Info:
Cells: 852465
Pitch = 50 mm
Core rod diameter = 2mm
Diameter of cylinder = 20 mm
Thickness of cylinder = 2mm

Grid Info:
Cells: 833722
Pitch = 50 mm
Core rod diameter = 2mm
Diagonal length = 20 mm
Thickness = 2mm

2.4 Simulation Scheme

Analysis of problem in ANSYS-CFX12 version

Sequence of steps involved in ANSYS-CFX 12.0 version and analysis:

- Determination of Mean Velocity (V) of working fluid:
- Using Reynolds Number considered from [17] the Mean Velocity of working fluid is determined by the following Equation. **$Re=\rho VD_i/\mu$**
- Selection of solver in ANSYS-CFX 12.0version: For the current study the type of solver chosen is upwind discretization model
- Under model Turbulence model: k-epsilon model selected
- Under materials tab in ANSYS-CFX 12.0version: The working fluid is air.
- Defined suitable Operating conditions.
  Define Boundary Conditions:
  Inlet-Velocity
  Constant Heat Flux of **759.010** w/m ^{2}on outer surface
  Pressure Outlet 101395.8 Pa
- Assigned suitable Residual values ranging up to 10^{-6}
- Then solve the following equations:
  Flow
  Turbulence
  Energy

3. Results and Discussions.

Each case was run using higher order residual schemes for each governing equations. It was ensured that residuals dropped to at least 10^{-6} for each case. Nusselt number and friction factor for plain tube are validated with theoretical relations [Table:3.1] and then they are determined for trapezoidal, diamond and cylinder insert. Nusselt number and friction factor calculated for the plain tube and plain tube with different insert for 6000<Re<14000. The Nusselt number and friction factor obtained for the tube with insert are compared with Nusselt number and friction factor of Plain tube.Each case is solved for 3 equations Energy, Momentum and Turbulence and results are plotted on corresponding graphs as shown below.

Table: 3.1- Theoretical Calculations

case. No	Mass flow rate	Re	F	Nu
1	0.003334	7757.98	0.0338	25.79
2	0.004108	9557.67	0.0316	30.48
3	0.00474	11028.08	0.0303	34.17
4	0.005304	12341.18	0.0295	37.39

ANSYS-CFX 12.0 version Calculations:

In ANSYS-CFX 12.0version also cases are analyzed separately one for plain tube, other for tube with trapezoidal, diamond and cylinder inserts respectively. The calculations for these cases are tabulated below.

Table 3.2: Plain Tube Calculations in ANSYS-CFX 12.0 version

case. NO	Mass flow rate	Tw(Avg. Wall temperature) K	Tb (K)	Q/A	Re	Nu	Pressure drop (Pa)	h (w/m^2-K)	F
1	0.003334	363.983	335.406	759.01	7757.98	23.8017	11.4221	24.7266	0.038733
2	0.004108	357.656	334.118	759.01	9557.67	28.4486	16.876	29.5541	0.037694
3	0.00474	354.168	332.848	759.01	11028.08	31.4305	21.7285	32.6519	0.036453
4	0.005304	351.706	332.193	759.01	12341.18	34.2683	26.5954	35.6000	0.035634

Table3.3: Plain Tube with Trapezoidal inserts Calculations in ANSYS-CFX 12.0 version

case.NO	Mass flow rate	Tw(Avg. Wall temperature) K	Tb (K)	Q/A	Re	Nu	Pressure drop (Pa)	h (w/m^2-K)	f
1	0.003334	355.796	331.630	759.01	7757.98	30.6313	84.207	31.8216	0.2855
2	0.004108	351.179	328.120	759.01	9557.67	35.9875	123.731	37.3859	0.2763
3	0.00474	348.821	326.920	759.01	11028.08	40.2445	161.399	41.8083	0.2707
4	0.005304	346.460	326.194	759.01	12341.18	43.9761	199.066	45.6849	0.2667

Table3.4: Plain Tube with Diamond inserts Calculations in ANSYS-CFX 12.0 version

case.NO	Mass flow rate	Tw(AvgWall temperature) K	Tb (K)	Q/A	Re	Nu	Pressure drop (Pa)	h (w/m^2-K)	f
1	0.003334	348.356	340.591	759.01	7757.98	44.7463	210.720	46.4851	0.7145
2	0.004108	344.797	336.692	759.01	9557.67	52.7770	311.489	54.8279	0.6957
3	0.00474	342.685	334.513	759.01	11028.08	59.1738	407.711	61.4732	0.6840
4	0.005304	341.191	333.100	759.01	12341.18	64.7826	504.144	67.3000	0.6754

Table3.5: Plain Tube with Cylinder inserts Calculations in ANSYS-CFX 12.0 version

case.NO	Mass flow rate	Tw(AvgWall temperature) K	Tb (K)	Q/A	Re	Nu	Pressure drop (Pa)	h (w/m^2-K)	f
1	0.003334	342.745	343.580	759.01	7757.98	68.8613	641.083	71.5372	2.1739
2	0.004108	339.982	339.640	759.01	9557.67	81.7430	953.987	84.9194	2.1308
3	0.00474	338.357	337.360	759.01	11028.08	91.9827	1253.98	95.5570	2.1037
4	0.005304	337.215	335.842	759.01	12341.18	101.046	1552.22	104.973	2.0797

Table3.6: Enhancement Efficiency calculations for different inserts

Mass flow rate	Re	Trapezoidal (Nu/Nu_0)	Diamond (Nu/Nu_0)	Cylinder (Nu/Nu_0)
0.003334	7757.98	1.286936106	1.879960804	2.893119698
0.004108	9559.023	1.264998522	1.855166685	2.873349743
0.00474	11029.64	1.280426341	1.882684053	2.926539091
0.005304	12342.03	1.283285181	1.890450434	2.948689579

Mass flow rate	Re	Trapezoidal $(f/f_0)^{1/3}$	Diamond $(f/f_0)^{1/3}$	Cylinder $(f/f_0)^{1/3}$
0.003334	7757.98	1.946265665	2.642327227	3.828741986
0.004108	9559.023	1.942687752	2.642760047	3.837876364
0.00474	11029.64	1.95114868	2.657301876	3.864443433
0.005304	12342.03	1.95612613	2.666343054	3.878962515

- **Flow Simulation of Enhancement of heat transfer in tube with different varying Reynolds numbers such as case1 = 7757.98, case 2 = 9559.023, case 3 = 11029.64, case 4 = 12342.03**

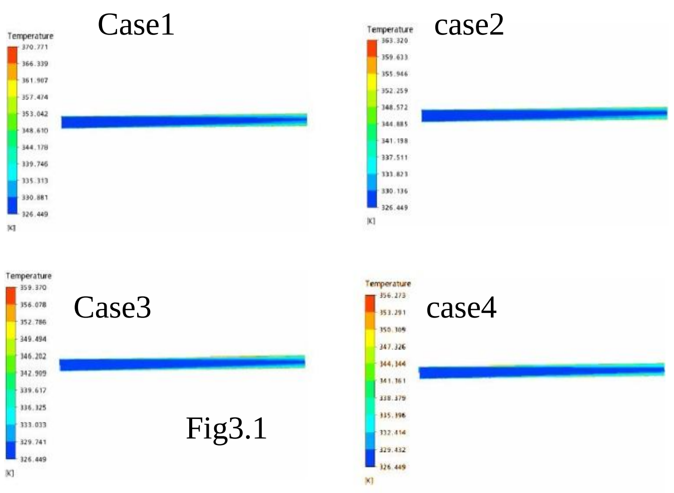


Fig3.1

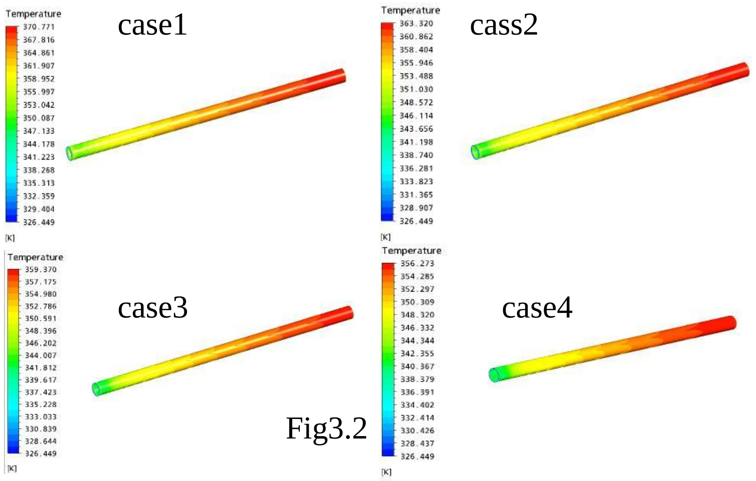


Fig3.2

Fig3.1 and Fig3.2 show that the static temperature variation for the plain tube configuration and on walls configuration.

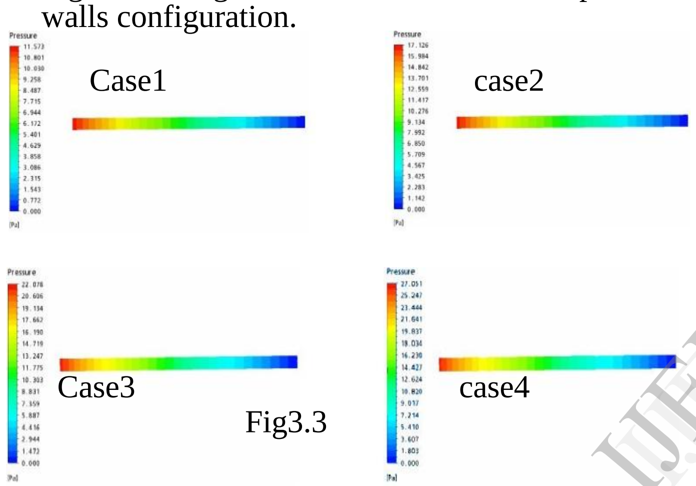


Fig3.3

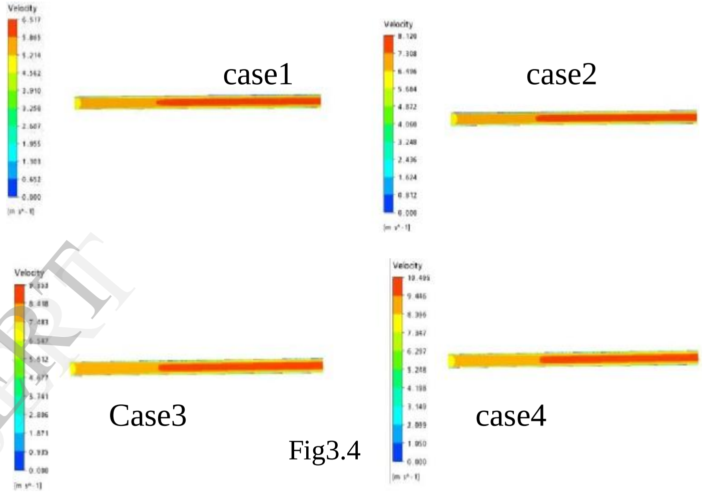


Fig3.4

Fig3.3 and Fig3.4 show that pressure and velocity variation for the plain tube configuration.

- **Flow Simulation of Enhancement of heat transfer in tube with Trapezoidal insert, with varying Reynolds number such as case1 = 7757.98, case 2 = 9559.023, case 3 = 11029.64, case 4 = 12342.03**

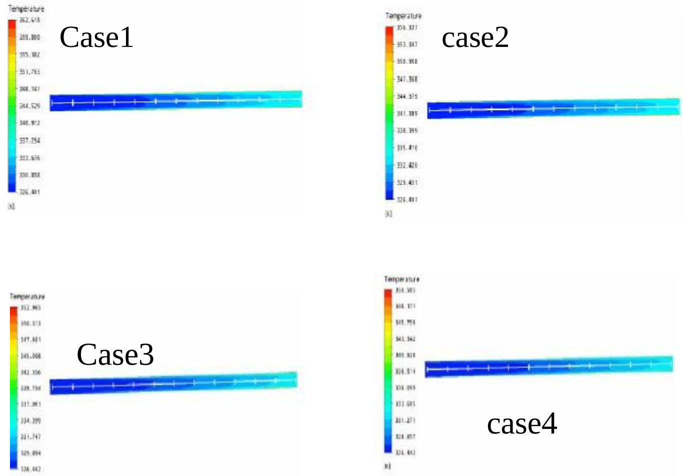


Fig3.5

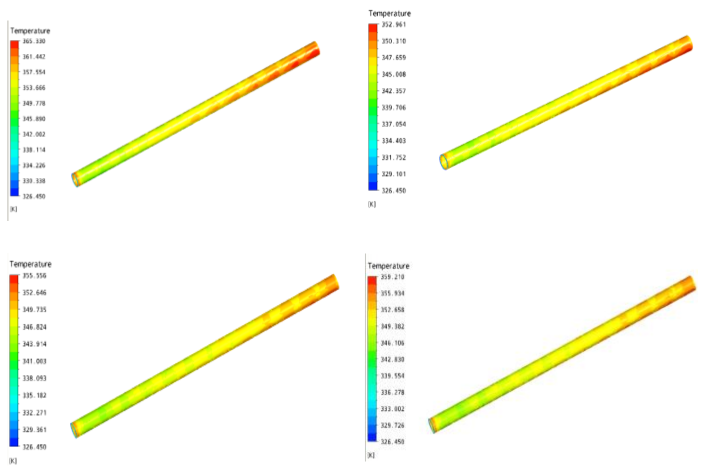


fig3.6

Fig3.5 and fig3.6 show that the static temperature variation for the tubewith trapezoidal insert

Configuration and on walls configuration.

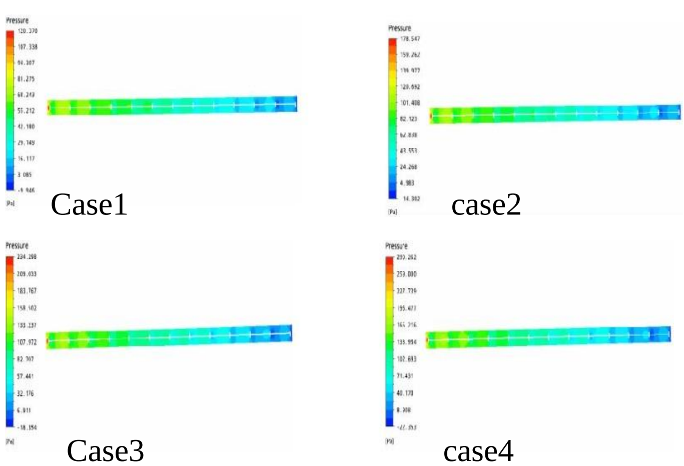


Fig3.7

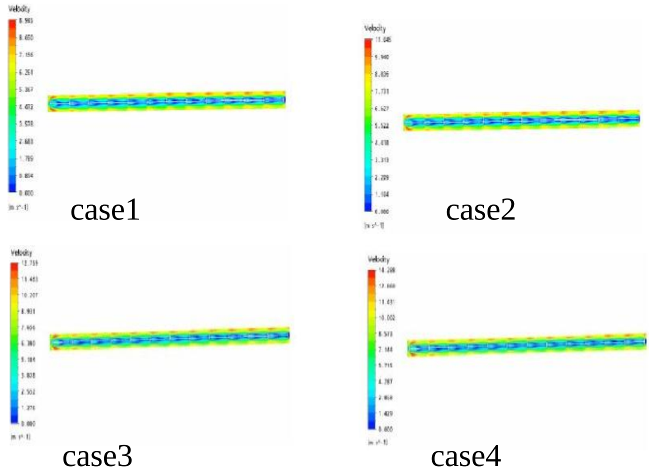


Fig3.8

Fig3.7and Fig3.8 show that pressure and velocity variation for the tube with trapezoidal insert configuration.

- **Flow Simulation of Enhancement of heat transfer in tube with Diamond insert, with varying Reynolds number such as case1 = 7757.98, case 2 = 9559.023, case 3 = 11029.64, case 4 = 12342.03**

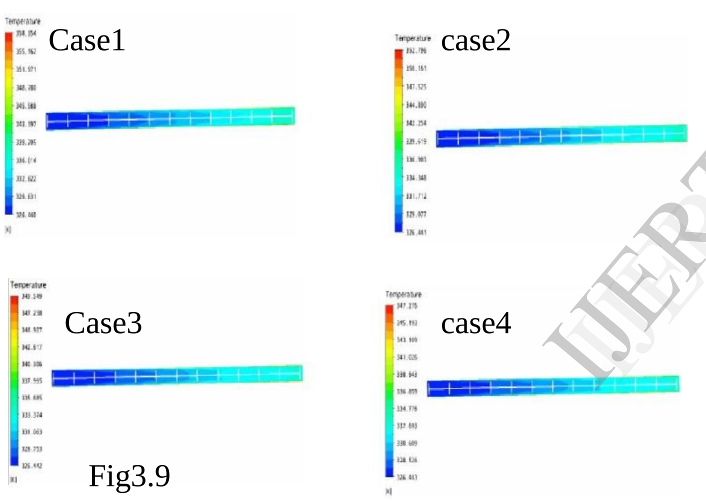


Fig3.9

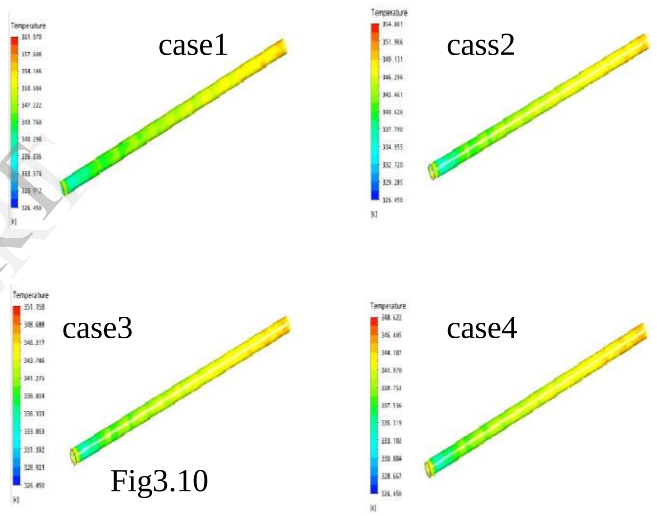


Fig3.10

Fig3.9 and Fig3.10 show that the static temperature variation for the tube with diamondinsert configuration and on walls configuration.

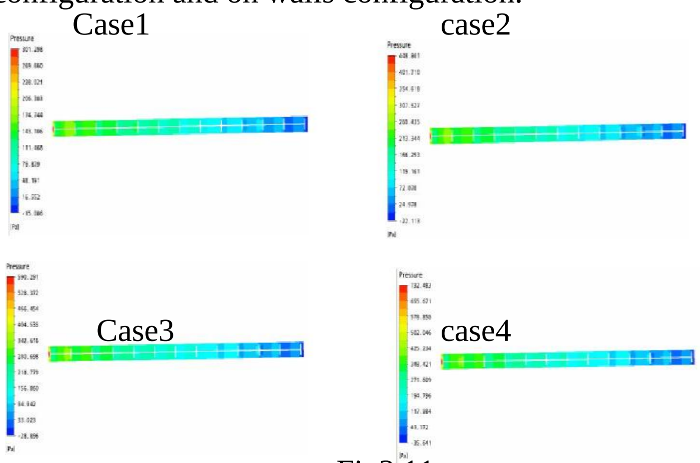


Fig3.11

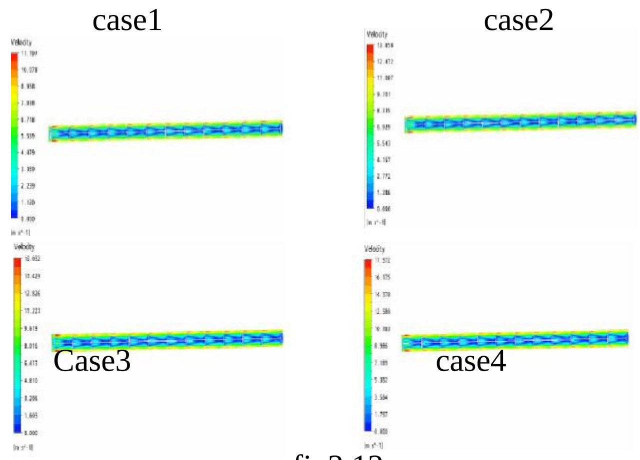


fig3.12

Fig3.11and Fig3.12 show that pressure and velocity variation for the tube with diamond insert configuration.

- **Flow Simulation of Enhancement of heat transfer in tube with cylinder insert, with varying Reynolds number such as case1 = 7757.98, case 2 = 9559.023, case 3 = 11029.64, case 4 = 12342.03**

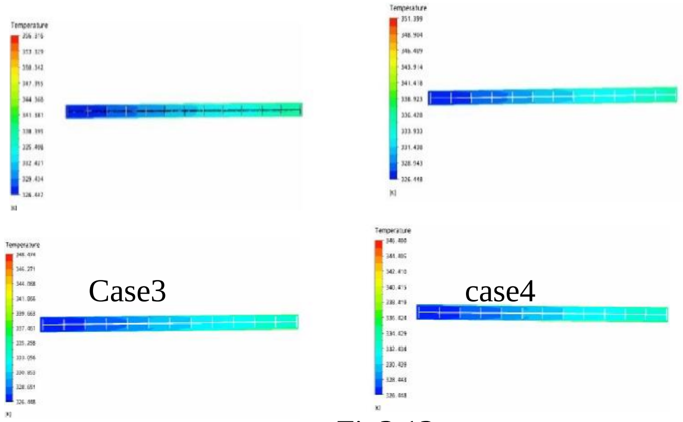


Fig3.13

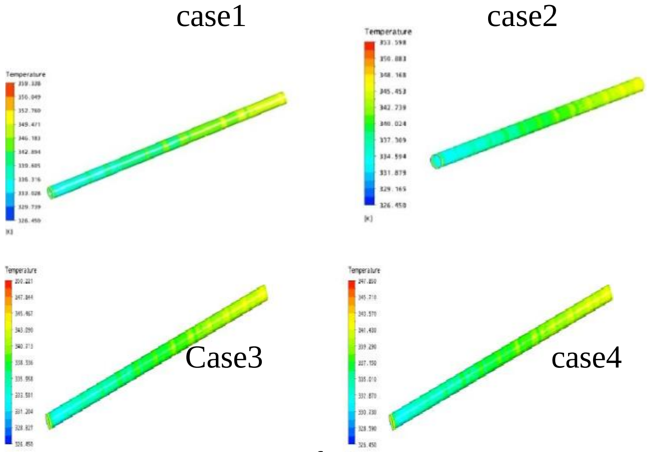


fig3.14

Fig3.13and Fig3.14show that static temperature variation for the tube with cylinder insert configuration and on walls.

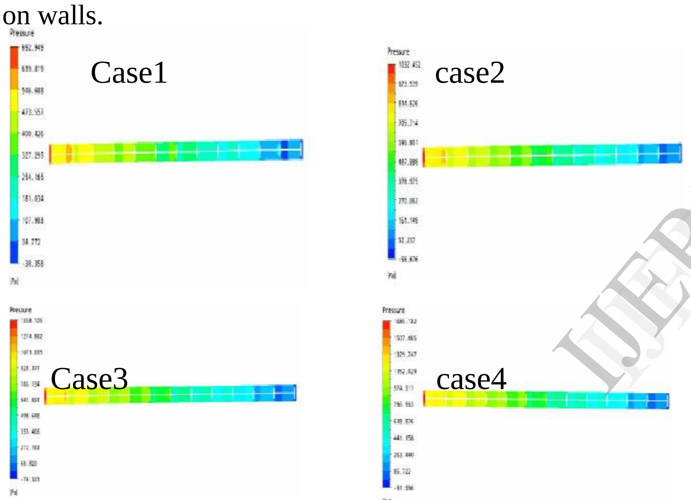


Fig3.15

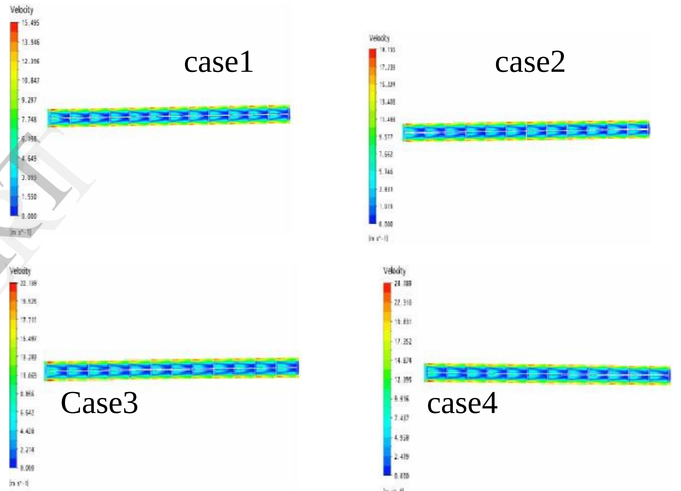
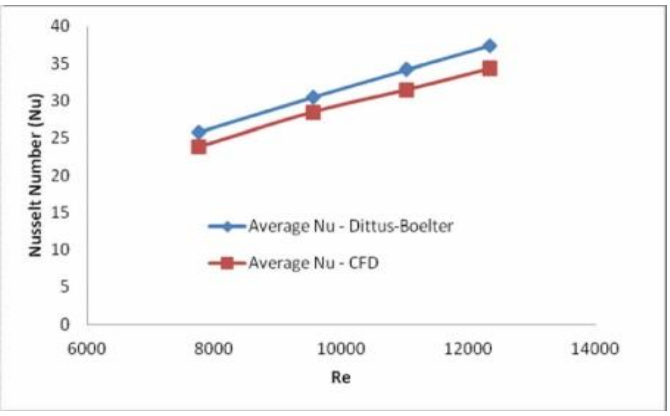


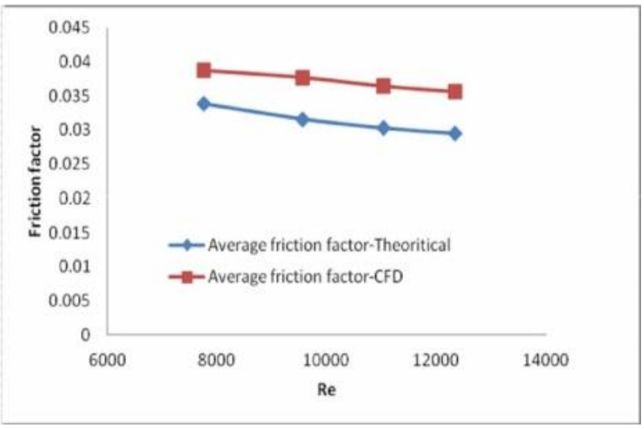
fig3.16

Fig3.15and Fig3.16 show that pressure and velocity variation for the tube with cylinder insert configuration.

Graph3.1



Graph3.2



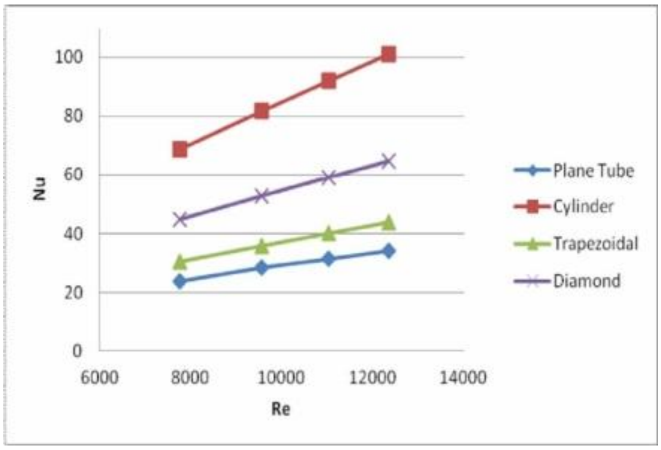
Graph3.1 show that the Comparison between Theoretical and CFD Analysis for Variation of Average Nusselt Number with Reynolds number in Plain Tube.

Inference from Graph3.1: the graph3.1 reveals that results obtained through ANSYS-CFX 12.0 version are almost identical with values thorugh DITTUS-BOELTER for Nusselt number. The Maximum % variation of CFD results for Nusslet number is found to be 6.1% with respect to theoretical values.

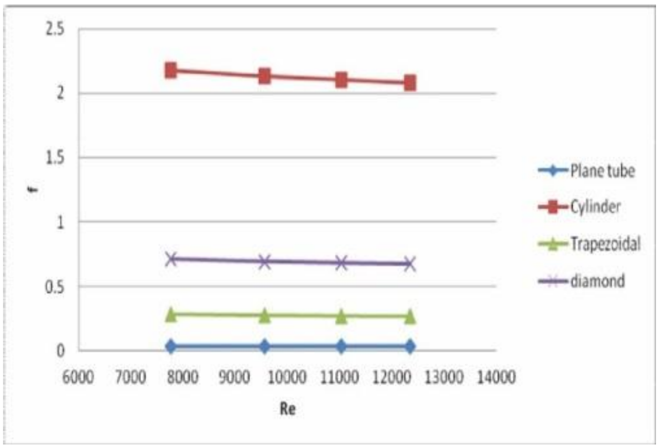
Graph3.2 show that the Comparison between Theoretical and CFD Analysis for Variation of Friction Factorwith Reynolds number in Plain Tube.

Inference from Graph3.2: the graph3.2 reveals that results obtained through ANSYS-CFX 12.0 version are well within the range of values obtained through DITTUS-BOELTER for Friction Factor. The Maximum % variation of CFD results for Friction Factor is found to be 10.3% with respect to theoretical values.

Graph3.3



Graph3.4



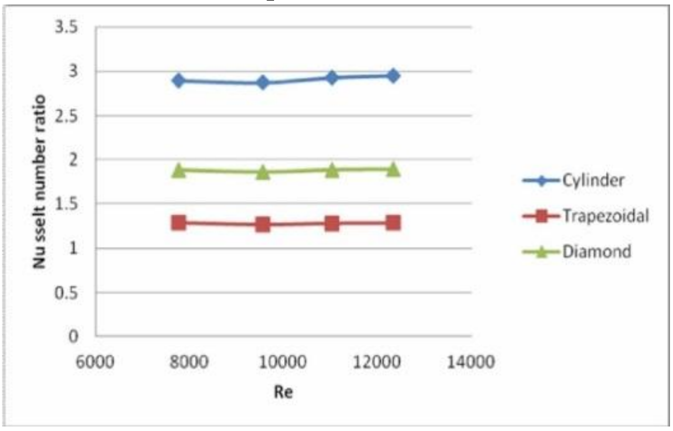
Graph3.3 show that Comparison between CFD Analysis for Variation of Nusselt Number with Reynolds number in Plain Tube Vs different insert.

Inference from Graph3.3: the graph3.3 reveals that results obtained through ANSYS-CFX 12.0 version. The maximum % variation of CFD results for Nusselt Number is found to 87% for cylinder insert, 85% for diamond insert and 28% of trapezoidal insert, with respect to Plain Tube.

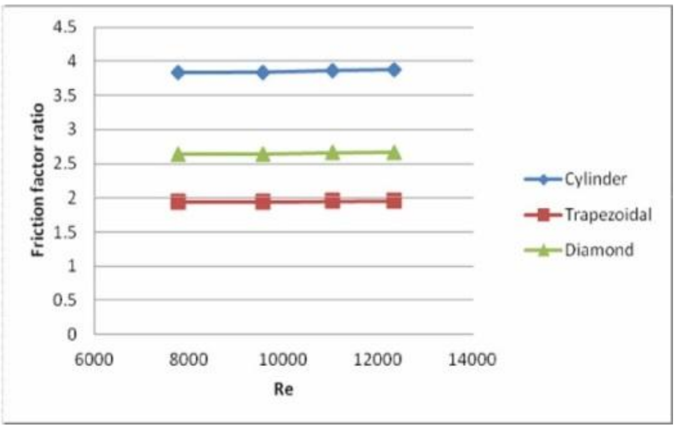
Graph3.4 show that Comparison between CFD Analysis for Variation of Friction factor with Reynolds number in Plain Tube Vs different inserts.

Inference from Graph3.4: the graph3.4 reveals that results obtained through ANSYS-CFX 12.0 version. The max 600 % variation of CFD results for Friction factor is found for cylinder insert with respect to Plain Tube.

Graph3.5



Graph3.6



Graph3.5 show that Comparison between the Nusselt Number ratio Vs Reynolds Number of CFD values of different inserts through ANSYS-CFX 12.0 version.

Inference from Graph3.5:the graph3.5 reveals that if Reynolds number increases Nusselt number ratio also increasing. Maximum Enhancement of Nusselt Number for cylinder insert is 2.8.

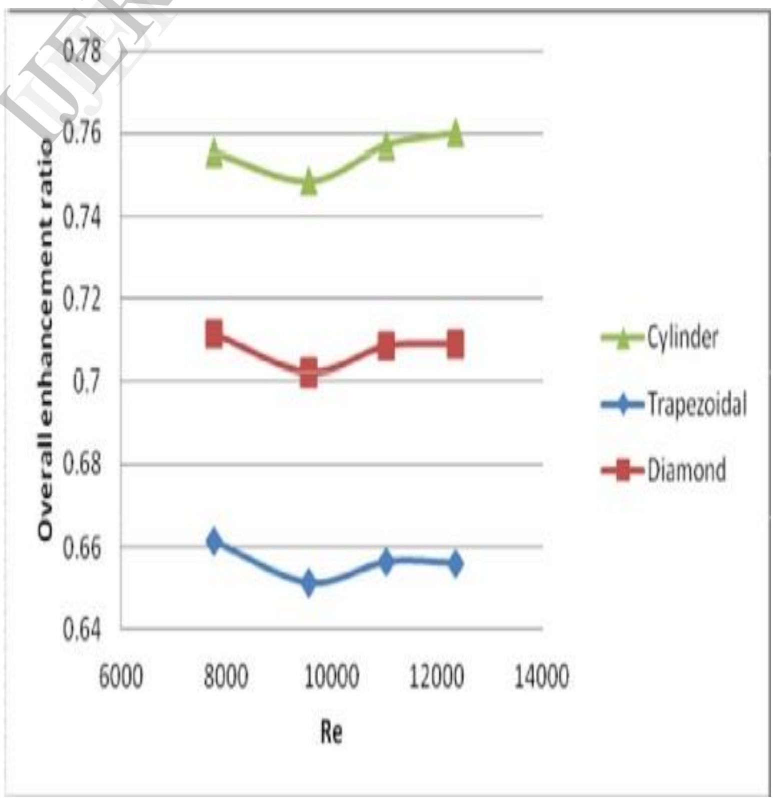
Graph3.6 show that Comparison between the Friction Factor ratio Vs Reynolds Number of CFD values of different inserts through ANSYS-CFX 12.0version.

Inference from Graph3.6:the graph3.6 reveals that if Reynolds number increases friction factor is decreasing. Minimum friction factor ratio for trapezoidal insert is 1.95.

NOMENCLATURE

Re	Reynolds No
P	Density in Kg/m^3
D_i	Inner diameter of tube in m
Do	outer diameter of tube in m
A	Surface area in m^2
V	Mean velocity of fluid in m/s
L	Length of tube in m
M	Dynamic viscosity of fluid
Nu	Nusselt number
F	Friction Factor
K	Thermal Conductivity in W-m/k
Q	Heat supplied in w
H	Heat transfer coefficient in W/m^2-k
T1	Inlet temperature in K
T0	outlet temperature in K
Tb	Bulk temperature in K
T_w	Inner surface temperature in K
Δp	Pressure Drop
N_{uo}	Average Nusselt Number for Plain Tube
N_{ut}	Average Nusselt Number for Tube with insert
F	Friction Factor for Plain tube
f_o	Friction Factor for Tube with insert

Graph3.7



Graph3.7 show that Comparison between the Overall enhancement ratio Vs Reynolds Number of CFD values of different inserts through ANSYS-CFX 12.0version.

Inference from Graph: Above graph reveals that overall enhancement ratio is high for cylinder (75%) and low for Trapezoidal insert (65%)

4 CONCLUSIONS

In the present work CFD Analysis of enhancement of heat transfer of different inserts for improving heat transfer in horizontal tube has been carried out with boundary conditions such as mass inlet and pressure outlet defined with constant heat flux. Mesh is created ANSYS-ICEM 13.0version(3- dimensional). The variations of Temperatures, average Nusselt Numbers, friction factor and pressure drop on with inserts like diamond, trapezoidal and cylinder has been studied.

- Results revelaed that average Nusselt Numbers and friction factors are considerably more with different inserts when compared to plain tube.
- Improvement of average Nusselt Numbers for tube with 87% for cylinder insert, 85% for diamond insert and 28% of trapezoidal insert, with respect to Plain Tube.
- Similarly friction factor for tube with cylinder insert is 600% more compared with the plain tube. Overall enhancement ratio is high for cylinder (75%) and low for Trapezoidal insert (65%).
- The pressure drop is considerably more with different inserts when compared to plain tube, pressure drop is high for cylinder insert and low for trapezoidal insert.

FUTURE WORKCFD simulation can work with different inserts as well as different Reynolds number. Also can carry out the experimental studies to validate the present results

REFERENCE

[1] shou-Shing Hsieh ,Feng-Yu Wu,Huang-Hsiu Tsai “Turbulent heat transfer and flow characteristics in a horizontal circular tube with strip-type inserts” Part I. Fluid mechanics. International Journal of Heat and Mass Transfer 46 (2003) 823–835.

[2] M Ahmed, L Deju, M. A. R. Sarkar and S. M. Nazrul Islam “heat transfer in turbulent flow through a circular tube with twisted tape inserts” ICME05-TH-08, International Conference on Mechanical Engineering 2005 (ICME2005) 28-30 December 2005, Dhaka, Bangladesh.

[3]M A R Sarkar, A B M ToufiqueHasan, M Ehsan, M MAlamTalukdar and A M A Huq “heat transfer in turbulent flow through tube with longitudinal strip inserts” ICME05-TH-46, International Conference on Mechanical Engineering 2005 (ICME2005) 28- 30 December 2005, Dhaka, Bangladesh.

[4]Hiral N. Prajapati “numerical study on heat transfer and pressure drop characteristics of a tube equipped with wire coil insert” 37th National & 4th International Conference on Fluid Mechanics and Fluid Power ,December 16-18, 2010, IIT Madras, Chennai, India.

[5]S. Naga Sarada, A.V. Sita Rama Raju, K. KalyaniRadha, L. Shyam Sunder “Enhancement of heat transfer using varying width twisted tape inserts” International Journal of Engineering, Science and Technology Vol. 2, No. 6, 2010, pp. 107-118 © 2010 MultiCraft Limited.

[6]A.E Bergle, “some perspective on enhanced heat transfer, second generation heat transfer technology” ASME journal of heat transfer 110 (November) (1988) 1082_1096.

[7]J.P. Holman, “Heat Transfer” 6 th edition, McGraw-Hill Book company, 1986.

[8]A.E. Bergles, “Techniques to augment heat transfer” in W.M Rohsenow, a.p. Hartnett, E Ganie (Eds.), Handbook of Heat Transfer Application, , McGraw-Hill, NEW YORK,1985.

[9]Shou-ShingHsieh , Ming-Ho Liu, Huang-Hsiu Tsai “Turbulent heat transfer and flow characteristics in a horizontal circular tube with strip-type inserts” Part II. Heat transfer. International Journal of Heat and Mass Transfer 46 (2003) 837–849.

[10]S. Eiamsa-ard and P. Promvonge “Enhancement of Heat Transfer in a Circular Wavy-surfaced Tube with a Helical-tape Insert” International Energy Journal 8 (2007) 29-36

[11]P. Promvonge , S. Eiamsa-ard “Heat transfer behaviors in a tube with combined conical-ring and twisted-tape

insert” Mahanakorn University of Technology, Bangkok 10530, Thailand. 21 May 2007 Elsevier Ltd.

[12]S. Eiamsa-ard , K. Wongcharee , P. Eiamsa-ard , C. Thianpong “Heat transfer enhancement in a tube using delta-winglet twisted tape inserts” Applied Thermal Engineering 30 (2010) 310–318 © 2010 Elsevier Ltd.

[13]S. Eiamsa-ard , P. Nivesrangsan , S. Chokphoemphun , P. Promvonge “Influence of combined non-uniform wire coil and twisted tape inserts on thermal performance characteristics” International Communications in Heat and Mass Transfer 37 (2010) 850–856 © 2010 Elsevier Ltd.

[14]V. Kongkaitpaiboon, K. Nanan, S. Eiamsa-ard “Experimental investigation of convective heat transfer and pressure loss in a round tube fitted with circular-ring turbulators” International Communications in Heat and Mass Transfer 37 (2010) 568–574 © 2010 Elsevier Ltd.

[15]P. Murugesan , K. Mayilsamy, S. Suresh “Heat Transfer and Friction Factor in a Tube Equipped with U-cut Twisted Tape Insert” © 2011 Jordan Journal of Mechanical and Industrial Engineering, Volume 5, Number 6, Dec. 2011, ISSN 1995-6665,Pages 559 – 565.

[16]P.K.Nagarajan and P.Sivashanmugam “Heat Transfer Enhancement Studies in a Circular Tube Fitted with Right-Left Helical Inserts with Spacer” World Academy of Science, Engineering and Technology 58 2011.

[17]R.C. Prasad, J. Shen, Performance evaluation of convective heat transfer enhancement devices using exergy analysis, International Journal of Heat and Mass Transfer 36 (1993) 4193–4197.