CFD Investigation for Counter Flow on Heat Transfer Characterstics in Concentric Tube Heat Exchanger with Varying Mass Flow Rate Using ANSYS FLUENT 14.5

Himanshu Bhist[1], Shivasheesh Kaushik[1], Sumit Kumar[2], Mayank Bhola[2], Harsh Kumar Yadav[3], Satyendra Singh[3]

Abstract— Cylindrical pipes are used very extensively in a lot of heat transfer and engineering applications. They have found extensive use in various types of Heat Exchangers, in Automobile in thermal power plants. There are three modes of heat transfer namely Conduction, Convection, Radiation. The heat transfer in Heat Exchanger usually involves convection on each side of fluids and conduction through the wall separating the two fluids. In this research paper a circular double pipe heat exchanger with inserts which mounted on the outerside of inner tube at different locations is modeled and CFD analysis is performed on it. In the present research work, the heat transfer and friction factor in circular tube channel with or without insert has been analyzed, and its dependency on geometrical spacing is reported for counter flow concentric tube heat exchanger in this study. The heat transfer effect is primarily due to induced turbulence which gives higher heat transfer rate. As shown from the study, different latitudinal spacing like y=15 and y=45 with same geometry give different results at varying mass flow rate (Reynolds number/inlet velocity) and thermal (heat flux) conditions. In this present research it is determine after all sort of investigation regarding heat transfer characteristics of various parameters for different geometrical linear spacing between two inserts with varying mass flow rate that as the spacing increase the quality of heat transfer characteristics for various parameters like Nusselt number, Reynolds number, Friction factor And Pressure drop will also decreases, as here we were using spacing y=15 and y=45. We achieve best results for linear geometrical spacing y=15 as compared to y=45.

Index Terms— Pentagonal Shape Insert, Nusselt Number, Heat Transfer Characterstic, Mass Flow Rate, Geometrical Spacing, Counter Flow

1 Introduction

The heat transfer in Heat Exchanger usually involves convection on each side of fluids and conduction through the wall separating the two fluids. In this research a circular double pipe heat exchanger with inserts at varying linear spacing is modeled and CFD analysis is performed with different mass flow rate on it. A large number of research analyses have been carried out on the circular double pipe heat exchanger during the recent years Giakward et al, (2014) investigated thermal performance of double pipe heat exchanger for laminar flow using twisted wire brush insert which are fabricated by winding a 0.2 mm diameter of the copper wires over a 2 mm diameter two twisted iron core-rods and concluded that the Nusselt number for the tube with twisted wire brush insert varied from 1.55 to 2.35 times in comparison of those of the plain tube. Sarada et al, (2010) investigated heat transfer in a horizontal circular tube using mesh insert in turbulent region and concluded that maximum Nusselt number obtained at smallest pitch of larger mesh diameter using CFD analysis which is 2.15 times that of plain tube. Jamra et al, (2012) investigated heat transfer enhancement in double pipe heat exchanger using simple pattern of rectangular insert and observed that the heat transfer coefficient varied from 1.9 times the smooth tube values. Pardhi et al, (2012) investigated performance improvement of double pipe heat exchanger by using two different twisted tape as turbulator. Conclusion of this work was that the heat transfer coefficient increased by 61% for twisted tape 1 and 78% for twisted tape 2. Patil et al, (2011) investigated thermohydraulic performance of tube in tube heat exchanger using twisted tape with winglets. Author concludes that twisted tape insert mixes the bulk flow well and therefore performs better in laminar flow, because in laminar flow the thermal resistant is not limited to a thin region. Omkar et al, (2014) investigated double pipe heat exchanger with helical fins on the inner rotating tube and concluded that the Nusselt number increased up to 64% at 100 rpm compared to stationary inner tube with helical fins. Al-Kayim et al, (2011) analytically investigated the thermal performance of double pipe heat exchanger with ribbed inner tube, an enhancement of 4 times in the heat transfer in terms of Stanton number is achieved. Pachegaonkar et al, (2014) investigated the performance of double pipe heat exchanger with annular twisted tape insert. Concluded that swirl flow helps decrease boundary layer thickness of the hot water flow and increase residence time of water in the outer tube.

Md. Julker Nine, Gyeong Hwan Lee, Hanshik Chung, Myoungkuk JI, and Hyomin Jeong (2014) The article

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represents an experimental investigation on friction and turbulent flow characteristics of free airflow through a rectangular duct fitted with semicircular ribs of uniform height (e = 3.5 mm) on one principle wall. The aspect ratio of the rectangular duct was AR= 5 where the duct height was 30 mm. Four different rib pitches of 28 mm, 35 mm, 42 mm, and 49 mm were examined with constant rib height to hydraulic diameter ratio ($e/Dh = 0.07$) and constant rib height to channel height ratio ($e/H = 0.116$). The experimental results show some significant effects of pressure drop as well as turbulent characteristics at various configurations among different numbers of rib arrangements varying Reynolds number in the range of 15000 to 30000. Experimental results have been compared with numerical analysis and it can be seen a good agreement. The result explains the phenomena elaborately between two periodic ribs and enables to optimize the rib pitch ratio in terms of turbulence kinetic energy for maximum heat transfer. The article analyzes friction and turbulent flow characteristics both in experimentally and numerically. For the convenience of analysis somewhere both results are compared. The prediction had a good similarity in case of pressure drop and aerodynamic analysis.

Naveen Sharma, Andallib Tariq and Manish Mishra (2015) The present experimental investigation examines the local heat transfer and friction factor characteristics of pentagonal ribs mounted on the bottom heated wall of a rectangular channel (AR = 4:1). The emphasis is towards assessing and analysing the potential impact of varying chamfering angle and rib pitch to height ratio on heat transfer and friction factor characteristics for turbulent flow. Kanade Rahul H., Kailash B A, Gowreesh (2015) higher rate of heat transfer and higher thermal efficiency are the main goals to improve the efficiency of heat exchanger.

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The present study investigates the effect of internal aluminium baffles on heat transfer enhancement and pressure drop in counter flow Double Pipe Heat Exchanger (DPHE) using computational fluid dynamic analysis (CFD).. Comparative study were employed for heat exchanger without baffles and inner pipe equipped with semicircular and quarter-circular baffles, parameters such as total residence time, pressure drops, overall HTC and Heat transfer rate are investigated.

Munish Gupta (2016) Over the past decade different research groups have worked on heat transfer characteristic in a tube type heat exchanger with nanofluids as working fluids. The heat transfer augments with the use of nanofluids due to improved thermophysical properties with addition of nanoparticles in the base fluid. Further the enhancement can be attained by using inserts inside the tube. This review article presents the works done by different researchers using inserts and nanofluids. The heat transfer is greatly improved with combination of nanofluids and inserts.

2 OBJECTIVE

The main aim is to investigate the varying effects on heat transfer characteristics or parameters of concentric heat exchanger with inner tube having pentagonal shape insert at different spacing for counter flow with varying mass flow rate and axial velocity. The other major objectives are as follows:

1- To calculate Nusselt number with varying mass flow rate for different spacing inserts.
2- To calculate friction factor with varying Reynolds number for different spacing inserts.
3- To calculate Pressure Drop with varying mass flow rate for different spacing inserts.
4- To calculate Average wall temperature with varying mass flow rate for different spacing inserts.
5- To develop various contours for optimum mass flow rate for different spacing inserts.

3 GOVERNING EQUATION

The behavior of the flow is generally governed by the fundamental principles of the classical mechanics expressing the conservation of mass and momentum. Here the considered steady, incompressible, turbulent flow is modeled by the momentum and continuity equations. The continuity and the momentum equations are as follows.

3.1 CONTINUITY EQUATION

Continuity Equation also called conservation of mass. Consider fluid moves from point 1 to point 2. The overall mass balance is Input – output = accumulation. Assuming that there is no storage the Mass input = mass output. However, as long as the flow is steady (time-invariant), within this tube, since, mass cannot be created or destroyed then the above equation. According to continuity equation, the amount of fluid entering in certain volume leaves that volume or remains there and

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according to momentum equation tells about the balance of the momentum. The momentum equations are sometimes also referred as Navier-Stokes (NS) equation. They are most commonly used mathematical equations to describe flow. The simulation is done based on the NS equations and then K-Epsilon model.

$$\frac{\delta(\rho\bar{u})}{\delta x}+\frac{1}{r}\frac{\delta(\rho r\bar{v})}{\delta r}=0 \qquad (2.1)$$

3.2 MOMENTUM EQUATIONS

Axial component (z-component)

$$\rho\bar{v}\left[\frac{\delta\bar{u}}{\delta r}+\bar{u}\frac{\delta\bar{u}}{\delta x}\right]=\frac{\delta\bar{p}}{\delta x}+\frac{\delta}{\delta x}\left(\mu_{eff}\frac{\delta\bar{u}}{\delta x}\right)+\frac{1}{r}\frac{\delta}{\delta r}\left(\mu_{eff}\frac{\delta\bar{u}}{\delta r}\right)+\frac{\delta}{\delta x}\left(\mu_{eff}\frac{\delta\bar{u}}{\delta x}\right)+\frac{1}{r}\frac{\delta}{\delta r}\left(\mu_{eff}\frac{\delta\bar{u}}{\delta r}\right) \qquad (2.2)$$

Radial component (r-component)

$$\rho\left[\bar{v}\frac{\delta\bar{v}}{\delta r}+\bar{u}\frac{\delta\bar{v}}{\delta x}\right]=-\frac{\delta\bar{p}}{\delta r}+\frac{\delta}{\delta x}\left(\mu_{eff}\frac{\delta\bar{v}}{\delta x}\right)+\frac{1}{r}\frac{\delta}{\delta r}\left(r\mu_{eff}\frac{\delta\bar{v}}{\delta r}\right)+\frac{\delta}{\delta x}\left(\mu_{eff}\frac{\delta\bar{u}}{\delta r}\right)+\frac{1}{r}\frac{\delta}{\delta r}\left(r\mu_{eff}\frac{\delta\bar{v}}{\delta r}\right)-2\mu_{eff}\frac{\bar{v}}{r^2}+\rho\frac{\bar{w}^2}{r} \qquad (2.3)$$

Tangential Component (□- component)

$$\rho\left[\bar{v}\frac{\delta\phi}{\delta r}+\bar{u}\frac{\delta\phi}{\delta x}\right]=\frac{\delta}{\delta x}\left[\mu_{eff}\frac{\delta\phi}{\delta x}\right]+\frac{1}{r}\frac{\delta}{\delta r}\left[r\mu_{eff}\frac{\delta\phi}{\delta r}\right]-\frac{2}{r}\frac{\delta}{\delta r}\left[\mu_{eff}\phi\right] \qquad (2.4)$$

Here $\bar{u}$, $\bar{v}$ and $\bar{w}$ are the mean velocity components along z, r and θ directions respectively and the variable $\phi = r\bar{w}$.

The total effective viscosity of the flow is given by,

$$\mu_{eff}=\mu_l+\mu_t \qquad (2.5)$$

Here μ_l and μ_t stand for molecular or laminar viscosity and eddy or turbulent viscosity respectively. The molecular or the laminar viscosity is the fluid property and the eddy viscosity or the turbulent viscosity is the flow property. By using dimensional analysis, the eddy viscosity μ_t can be expressed as,

$$\mu_t \sim \rho V_t l$$

here V_t , is the turbulent velocity scale and l is the turbulent length scale. It was postulated by Prandtl and Kolmogorov and later adopted in the standard k-ε model that

$$l=\frac{\kappa^{3/2}}{\varepsilon} \text{ and } V_t \sim \sqrt{k} \qquad (2.6)$$

From the equation (6) the eddy viscosity is obtained and it is given by

$$\mu_t=\frac{\rho c\mu k^2}{\varepsilon} \qquad (2.7)$$

The modelling constant, C_μ in the eddy viscosity formulation, as shown in equation (2.8), is empirically tuned for the simple shear layer. Meanwhile, there is no mechanism in the k-ε model which can either amplify the turbulent intensity or eddy viscosity in the presence of concave or convex curvature. Therefore, the expression for eddy viscosity in the standard k-ε model is considered to be inadequate to account for the streamline curvature effect. It is evident that modifications to the standard k-ε model are necessary to include the curvature effects. Therefore the constant C_μ is considered. The constant, C_μ is given by

$$C_\mu=\frac{-k_1k_2}{\left[1+8k_1^2\frac{k^2}{\varepsilon^2}\left(\frac{\delta U_s}{\delta n}+\frac{U_s}{R_c}\right)\frac{U_s}{R_c}\right]} \qquad (2.8)$$

In the eq. (2.8) $U_s=\sqrt{\bar{u}^2+\bar{v}^2}, R_c$ is the radius of curvature of the streamline concerned (Ψ constant). The value of k1 and k2 are taken as 0.27 and 0.3334 respectively.

3.3 THE TURBULENT MODELING

3.3.1 KAPPA-EPSILON MODEL:

The K-epsilon model is most commonly used to describe the behavior of turbulent flows. It was proposed by A.N Kolmogrov in 1942, then modified by Harlow and Nakayama and produced K-Epsilon model for turbulence. The Transport Equations for K-Epsilon model are for k, Realizable k-epsilon model and RNG k-epsilon model are some other variants of K-epsilon model. K-epsilon model has solution in some special cases. K-epsilon model is only useful in regions with turbulent, high Reynolds number flow.

K − Equation

$$\rho\left[\bar{u}\frac{\partial k}{\partial x}+\bar{v}\frac{\partial k}{\partial r}\right]=\frac{\partial}{\partial x}\left[\left(\mu_l+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x}\right]+\frac{1}{r}\frac{\partial}{\partial r}\left[r\left(\mu_l+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial r}\right]+\rho g-\rho\varepsilon \qquad (2.9)$$

Where, G is the production term and is given by

$$G=\mu_t\left[2\left\{\left(\frac{\partial\bar{v}}{\partial r}\right)^2+\left(\frac{\partial\bar{u}}{\partial x}\right)^2+\left(\frac{\bar{v}}{r}\right)^2\right\}+\left(\frac{\partial\bar{u}}{\partial r}+\frac{\partial\bar{v}}{\partial x}\right)^2\right] \qquad (2.10)$$

The production term represents the transfer of kinetic energy from the mean flow to the turbulent motion through the interaction between the turbulent fluctuations and the mean flow velocity gradients.

ε- Equation

$$\rho\left[\bar{u}\frac{\partial\varepsilon}{\partial x}+\bar{v}\frac{\partial\varepsilon}{\partial r}\right]=\frac{\partial}{\partial x}\left[\left(\mu_l+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x}\right]+\frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_l+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial r}\Big]+C_{S1}G\frac{\varepsilon}{k}-C_{s2}\frac{\varepsilon^2}{k} \qquad (2.11)$$

Here C_{s1}, C_{s2}, σ_k and σ_ε are the empirical turbulent constant. The values are considered according to the Launder et al., 1974. The values of Cμ, C_{s1}, C_{s2}, σ_k and σ_ε are 0.09, 1.44, 1.92, 1.0 and 1.3 respectively.

3.4 DITTUS-BOELTER EQUATION

Although the heat transfers in fluids are all forced convection. In addition, the heat transfer inside the tube is an internal convection, such that another important characteristic is that the fluid is confined by the surface wall of the tube. As a result, the heat transfer phenomena inside tube are thus closely associated with the characteristic of the hot fluid and the di-

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mension of the tube. For heating water, the Dittus-Boelter equation as a classical expression for computing the Nusselt number of fully developed turbulent flow in a tube is given as,

$$N_u = 0.023\,R_e^{.8}\,P_r^{\,n} \tag{2.12}$$

Where n = 0.4 for heating and n = 0.3 for cooling.

For calculating the heat transfer coefficient

$$N_u = \frac{hD}{k} \tag{2.13}$$

Where D is the diameter of the tube

3.5 FRICTION FACTOR

The Fanning friction factor is the ratio of wall shear stress to the flow kinetic energy. It is related to pressure drop in tube heat exchanger as:

$$f = \frac{D_h}{L}\,\frac{\Delta p^*}{(1/2)\rho V^2} \tag{2.14}$$

3.6 BOUNDARY CONDITIONS

A turbulent flow is considered. The quantities U, k, ε are obtained by using numerical calculations based on the k-ε model for high Reynolds Number. The boundary conditions are listed below:

1) At the inlet of the channel:

$$u = U_{in,}\, v = 0 \tag{2.15}$$

$$k_{in} = 0.005 U_{in}^{\;2} \tag{2.16}$$

$$\varepsilon_{in} = 0.1 K_{in}^{\;2} \tag{2.17}$$

Kin stands for the admission condition for turbulent kinetic energy and ε_{in} is the inlet condition for dissipation.

2) At the walls:

$$u = v = 0 \tag{2.17}$$

$$k = \varepsilon = 0 \tag{2.18}$$

3) At the exit:

$$P = P_{atm} \tag{2.19}$$

The Reynolds number based on circular diameter in case of circular tube and hydraulic diameter Dh in case of rectangular tube.

$$\mathrm{Re} = \frac{\rho . U_0 . D_h}{\mu} \tag{2.20}$$

For fully development turbulent flow, the centerline velocity Vc is given by

$$\frac{u}{Vc} = \left(1 - \frac{r}{R}\right)^{\frac{1}{n}} \tag{2.21}$$

Where u is the time mean average of x- component of instantaneous velocity, V_c is the centreline velocity or axial velocity, R is the radius of pipe, r is the radius of elementary ring and n is a function of the Reynolds number.

4 MATERIAL AND METHEDOLOGY

4.1. GEOMETRY OF MODEL

For the numerical simulations presented in this work, we refer to the numerical and experimental work done by Sarmad A. et al, (2015) who studied the insert with a simple pattern. The geometric dimensions of the system are listed below.

The geometry of the problem is presented on Fig. 4.1.a, 4.1.b, 4.1.c. The system consist of air at 299 K flowing through inner pipe of circular double tube parallel flow heat exchanger in plain tube and plain tube with insert condition and analyzed. The insert are of rectangular type with 1mm width and are fitted in the direction parallel to the fluid flow.

Table 4.1 Geometry description of model

Length of the heat exchanger L (mm)	2200mm
Diameter of inner pipe d (mm)	53mm
Diameter of annulus space D (mm)	53mm
Thickness of inner pipe t (mm)	3mm
Thickness of outer pipe T (mm)	8mm
Diameter of rod fixed with insert din (mm)	22mm
Width of insert w (mm)	12.5mm
Height of insert h (mm)	26mm
Length of insert lin (mm)	1mm
Length of heat exchanger upon which inserts are acting l (mm)	2000mm
Insert shape	Pentagonal
Distance between two insert pattern ldi (m)	150mm
Distance between two similar insert pattern lsi (m)	300mm

4.2 GEOMETRICAL DISCRIPTION

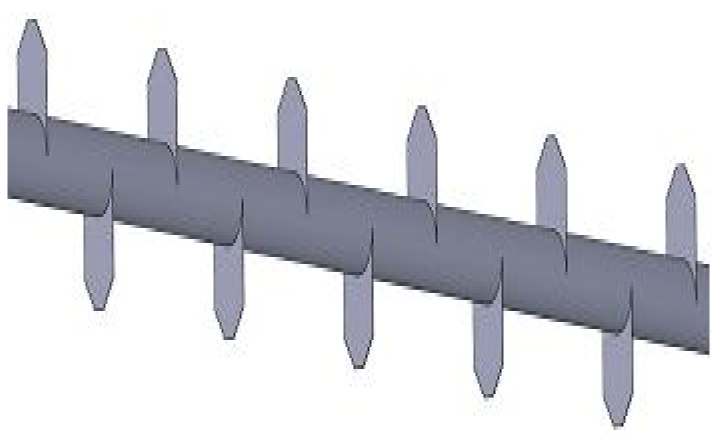

Figure 4.1 Inner Circular Tube having Pentagonal Shape Inserts Having Spacing y =15cm

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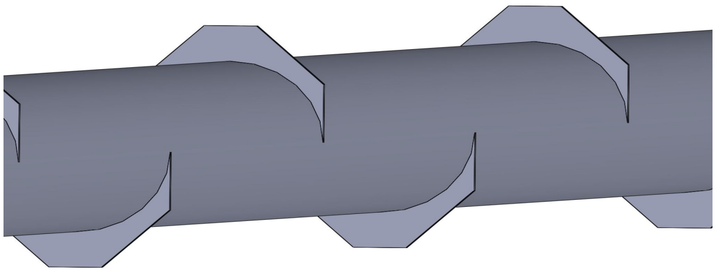


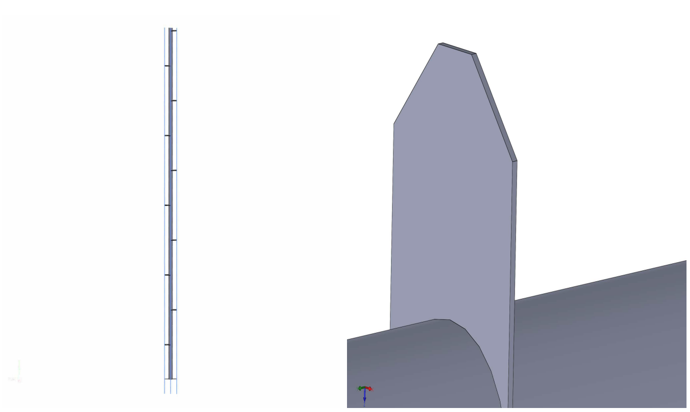
Figure 4.2 Inner Circular Tube having Pentagonal Shape Inserts Having Spacing y = 45cm

Table 4.2 Properties of working fluid

Density (kg / m3)	998.2-995
Specific heat Cp (J / kg-k)	4182
Thermal conductivity (W / m-k)	0.6
Viscosity (kg / m-s)	0.001003
Inlet temperature of cold fluid(k)	293
Inlet temperature of cold fluid(k)	333

Table 4.3 Mesh setting domain for fluid.

Physics preference	CFD
Solver preference	fluent
Smoothing	high
Transition	fast
Minimum size	0.001
Maximum face size	0.001
Maximum size	0.001
Growth rate	1.850
Inflation	Program control
Maximum layers	5
Growth rate	1.2



(a) (b)

Figure 4.3 Top View and Isometric View of Inner Circular Tube having Pentagonal Shape Inserts

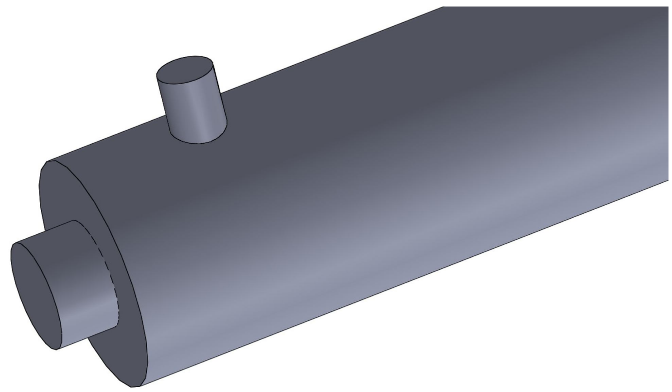


Figure 4.4 Assemble View of Concentric outer and Inner Circular Tube having Pentagonal Shape Inserts

4.3 MESHING DETAILS

Computational grid for fluid domain of circular double tube heat exchanger and circular double tube heat exchanger with insert is obtained using ANSYS FLUENT 14.5.

A tetrahedron mesh elements are obtained throughout the fluid volume using following mesh settings.

Table 4.4 Mesh details for plain tube with insert heat exchanger at y=45 spacing

Nodes	82984
Elements	266859
Orthogonal quality	0.51
Skewness	0.71
Min. aspect ratio	0.2
Max. aspect ratio	0.761

Table 4.5 Mesh details for plain tube heat exchanger

Nodes	82263
Elements	177679
Orthogonal quality	0.51
Skewness	0.67
Min. aspect ratio	0.4
Max. aspect ratio	0.866

Table 4.6 Mesh details for plain tube with insert heat exchanger at y=15 spacing

Nodes	86552
Elements	186819
Orthogonal quality	0.49
Skewness	0.57
Min. aspect ratio	0.38
Max. aspect ratio	0.896

Using the following mesh setting, no. of nodes and element, orthogonal quality, skewness and aspect ratio for circular double tube and circular double tube with insert are tabulated below

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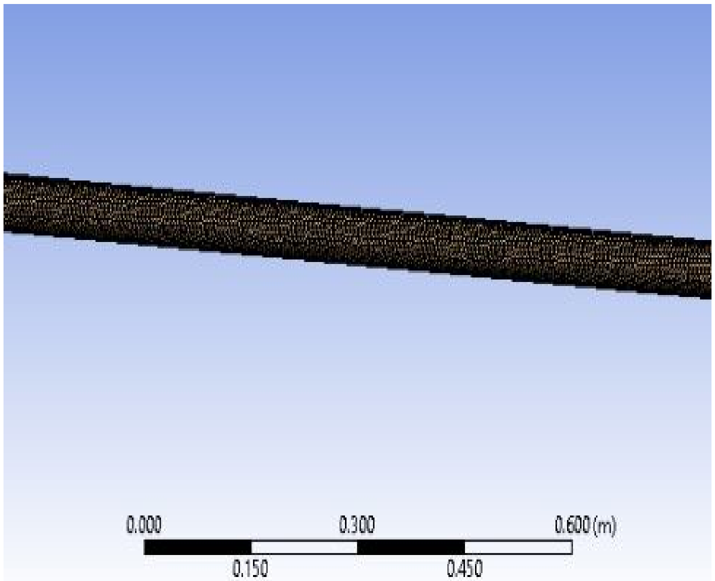

Figure4.5 Mesh View for plain tube heat exchanger

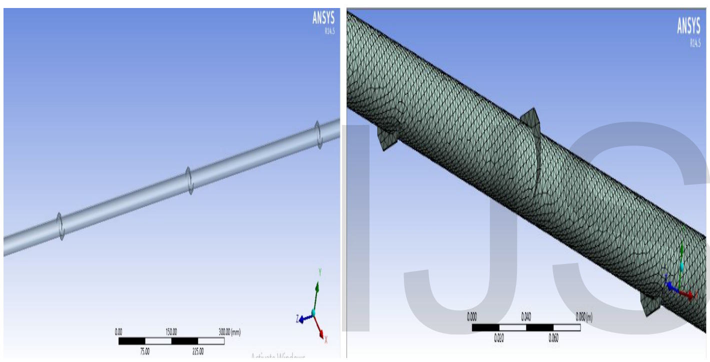

Figure 4.5 Mesh View for plain tube with insert heat exchanger at y=15 spacing

Table 4.7 Mesh details for plain tube with insert heat exchanger at y=45 spacing

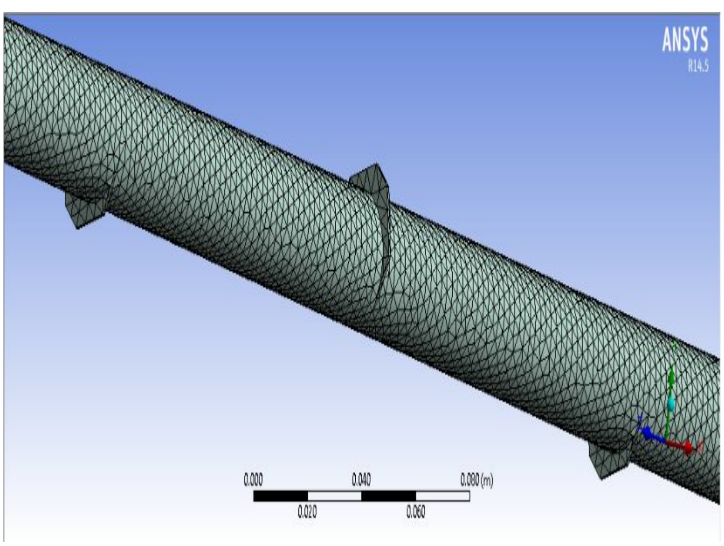

Figure 4.6 Mesh View for plain tube with insert heat exchanger at y=45 spacing

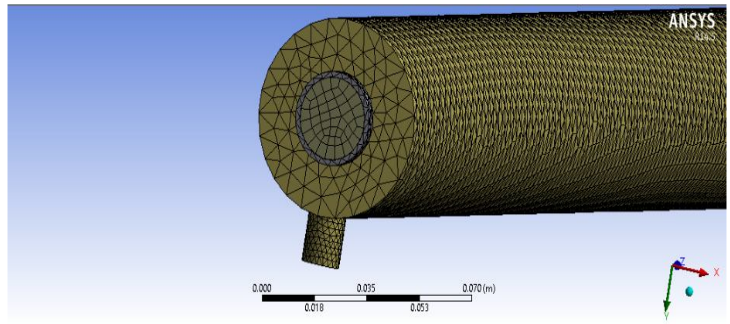

Figure 4.7 Assemble Mesh View for plain tube with concentric outer tube heat exchanger

4.4 BONUDARY CONDITIONS

Computational domain is solved using ANSYS FLUENT 14.5 CFD code. Following are the boundary conditions for the domain:-

Table 4.7 Boundary conditions

Inlet temperature of air	298
Mass flow rate of air (kg/s)	0.07-0.213
Inlet temperature of water	333
Mass flow rate of water (kg/s)	0.06
Pressure	1 atm

4.5 DATA REDUCTION

The data reduction of the measured results is summarized in the following procedures:

Rate Heat transfer to the hot water, Q_h, can be calculated from

$Q_h = m_h C_{p,w} T_{hi} - T_{ho}$ (2.1.1)

The rate heat transfer from the cold water, Q_c, can be calculated from;

$Q_c = m_c C_{p,w} T_{co} - T_{ci}$ (2.1.2)

The average rate heat transfer, $Q_{ave.}$, of the hot and cold water can be determined from

$Q_{ave.,} = Q_h + Q_c / 2$ (2.1.3)

The overall heat transfer coefficient, U, is determined from the following equation;

$U = Q_{ave}.Ai.LMTD$ (2.1.4)

Where,

The tube side heat transfer coefficient is than determined using;

$1/U = 1/h_o + 1/U_i$ (2.1.5)

Where the annulus side of heat transfer coefficient (h_o) is estimated using the correlation of Dittus-Boelter [8].

$Nu_o = h_o.D_h/k \quad = 0.023Re^{0.8}Pr^{0.3}$ (2.1.6)

Where,

The Reynolds number is based on the different flow rate at the inlet of the concentric tubes heat exchanger.

$Re = \rho\, v\, di/\mu$ (2.1.7)

Thus the experimental value of Nusselt number is evaluated by,

$Nu_{exp.} = hi.di/k$ (2.1.8)

Theoretical value of Nusselt number without semicircular disk baffles (smooth tube) can be determined by using the correlation of Dittus-Boelter [8].

$Nu_o = 0.023Re^{0.8}Pr^{0.3}$ (2.1.9)

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Experimental friction factor can be written as:

$f_{exp.} = \Delta P.\ d_i / 2\, \rho\, L\, v^2$ (2.1.10)

$v = m\ \rho.A_i$ (2.1.11)

Theoretical friction factor for smooth tube is calculated from the correlation of Blasius [9];

$f_{theo.} = 0.046\ Re^{0.2}$ (2.1.12)

All of thermo physical properties of the water are determined at the overall bulk temperature. The performance ration or enhancement efficiency (ψ) is defined as the ratio of the heat transfer coefficient for the tube fitted with inserted semi circular disc baffles (hc) to that for the smooth tube (hs) at a constant Reynolds number (Re) as follows [10 and 11]:

$\Psi = (hchs).Re$ (2.1.13)

4.6 Methedology Used:

The methodology of the present study can be divided into four stages of process flow which are geometry modeling, pre-processing, processing and post-processing. Various steps in adopted methods are:

- Mathematical modeling of the system considered in present study.
- Developed the model in SOLIDWORKS.
- Validation of present work with previous research.
- Calculation of heat transfer parameters.
- Run program to obtain the plots with different geometrical spacing of same shape insert inner tube parameters.
- Plotting & analysis of obtained plots.
- Optimization of the system.

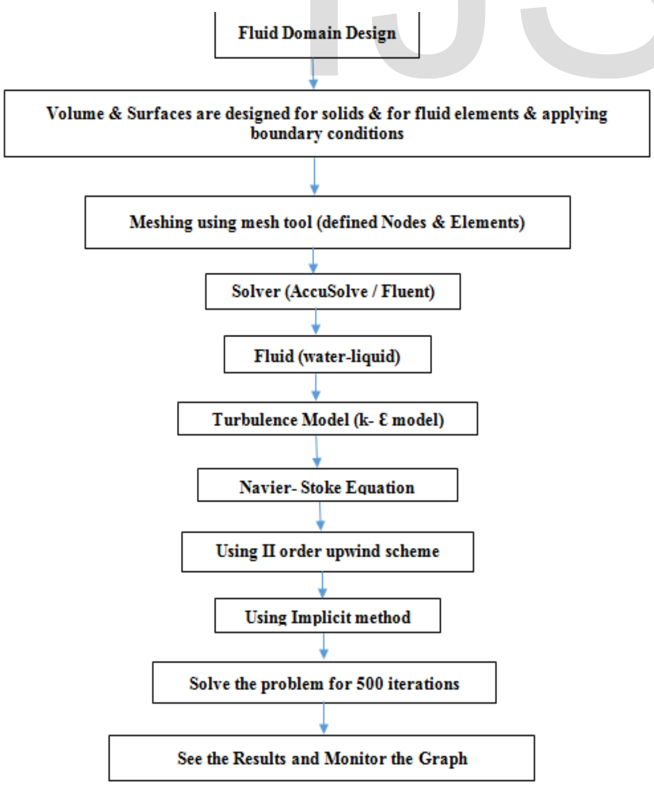

Figure 5.8: Methodology flow chart.

5 RESULT AND DISCUSSION

In the simulation work conducted on the circular double pipe heat exchanger, the mass flow rate of the air entering the circular tube is varied from 0.079kg/s to 0.213Kg/s corresponding to Reynolds number 5000 to 0.213 kg/s corresponding to a Reynolds number 15000 and a temperature of 298 K. Convective boundary condition is applied on inner tube of the heat exchanger. Temperature of hot water entering the tube is 333 K.

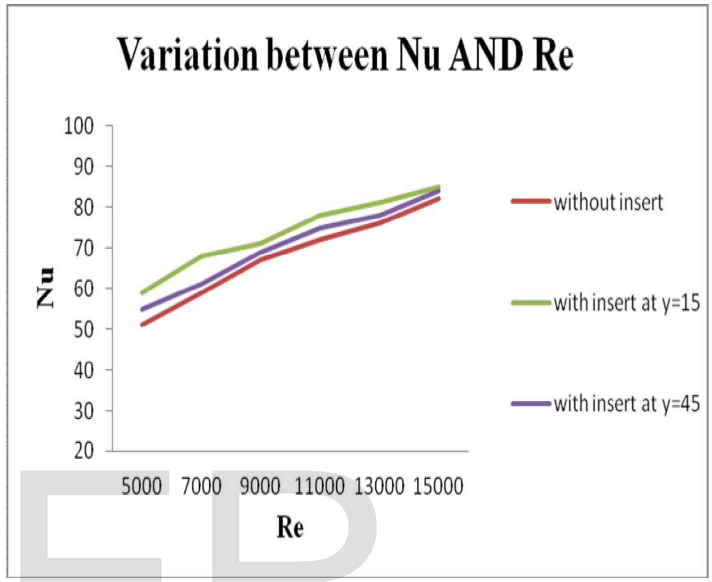

Figure 5.1 Graphical variations between Nusselt number and Reynolds number with varying latitudinal spacing for counter flow.

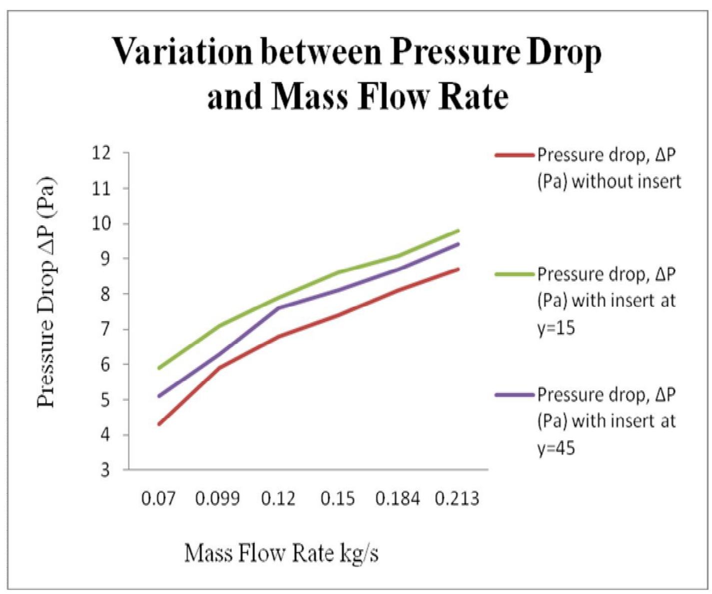

Figure 5.2 Graphical variations between Pressure Drop and Mass Flow Rate with varying latitudinal spacing for counter flow.

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A comparison of the average Nusselt number and heat transfer coefficient is shown in Fig.5.1 shows the heat transfer data without insert/ plane tube with varying mass flow rate of water. Fig.5.1 shows the heat transfer data with insert having linear geometrical spacing between two inserts y=45cm and Fig.5.1 shows the heat transfer data with insert having linear geometrical spacing between two inserts y=15cm. From table as the mass flow rate of water increases from 0.007 to 0.213 kg/s, which is responsible for increment of average Nusselt number value rises from 30 to 77 and the corresponding pressure drop (ΔP) increases from 9.2 to 11.6 Pa for plane tube case as shown in Fig.5.2. The corresponding friction factor is also calculated and the results showed the general trend of increase of heat transfer with decrease in friction factor as mass flow rate increases which shown in Fig. 5.4.

Fig.5.1 and 5.2 as the mass flow rate of water increases from 0.007 to 0.213 kg/s, and Reynolds number increases from 5000-15000 which is responsible for value increment of average Nusselt number rises from 51 to 93 and the corresponding pressure drop (ΔP) increases from 9.89 to 12.13 Pa for inner tube having insert with linear geometrical spacing y=45. It is evidently shown from figure that as the mass flow rate of water increases from 0.007 to 0.213 kg/s, and Reynolds number increases from 5000-15000 which is responsible for value increment of average Nusselt number rises from 55 to 98 and the corresponding pressure drop (ΔP) increases from 10.19 to 12.61 Pa for inner tube having insert with linear geometrical spacing y=15.

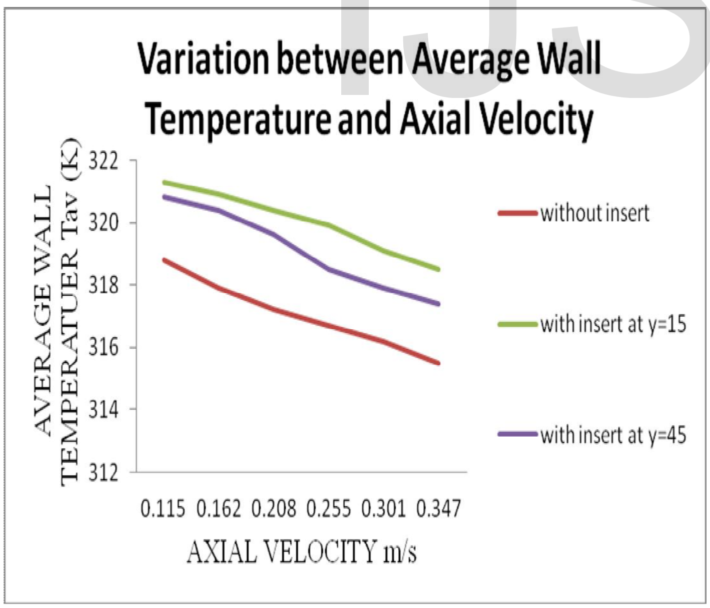

Figure 5.3 Graphical variations between Average Wall Temperature and Axial Velocity with varying latitudinal spacing for counter flow.

Fig. 5.1, 5.2, 5.3, and 5.4 shows the heat transfer data with insert with varying linear geometrical spacing. Heat transfer coefficient and average Nusselt number are calculated for varying inlet mass flow rate of air, as the mass flow rate increases from 0.007 to 0.213 kg/s, the value of average.

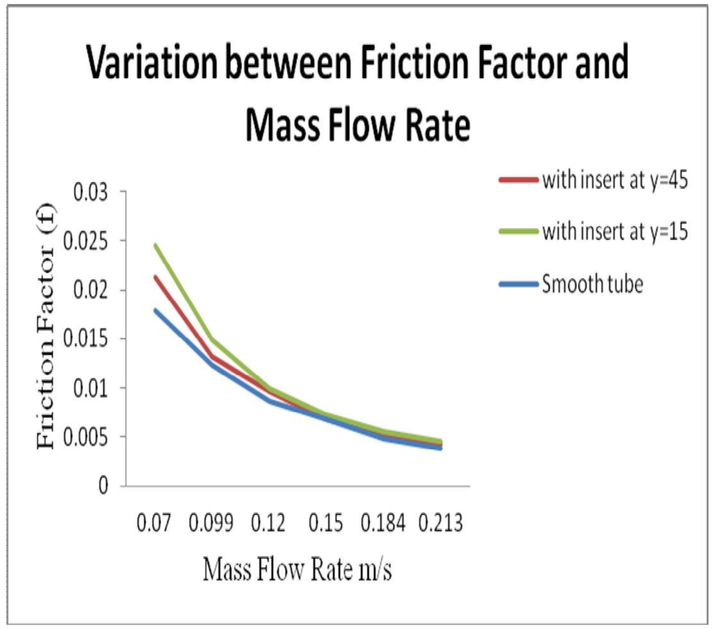

Figure 5.4 Graphical variations between Friction Factor and Mass Flow Rate with varying latitudinal spacing for counter flow.

Thus from all above graph of Nusselt number, pressure drop, it is evident that as mass flow rate, Reynolds number increases Nusselt number, pressure drop will always increases and graph of Average Wall Temperature, Friction Factor, it is evident that as mass flow rate, Reynolds number increases, the values of friction factor and wall temperature will always decreases. Thus that's why the above results show a trend of increase in heat transfer with the provision of insert on the heat exchanger as compared to plane tube for parallel flow. The heat transfer was found to increase as the Reynolds number was varied over the range. The results obtained show that the effect of insert on the enhancement of heat transfer depends on both the pattern of insert and the Reynolds number of the flow.

The results showed a trend of increase in heat transfer with the provision of insert on the heat exchanger. The heat transfer was found to increase as the Reynolds number was varied over the range. The results obtained show that the effect of insert on the enhancement of heat transfer depends on both the pattern of linear geometrical spacing of insert as well as consider shape of insert.

6 Conclusions

The heat transfer and friction factor in circular tube channel with or without insert has been analyzed, and its dependency on geometrical spacing is reported in this research study.

The heat transfer effect is primarily due to induced turbulence which gives higher heat transfer rate. As shown from the study, different latitudinal spacing like y=15 and y=45 with same geometry give different results at varying mass flow rate

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(Reynolds number / inlet velocity) and thermal (heat flux) conditions. The choice of appropriate geometry does not always depend on high Nusselt number values, pumping power available and constant behavior for all parameters plays an important role.

According to study, convective properties shown by different spacing are: pentagonal geometry at spacing y=45 showed approximately 49% more convective transfer, pentagonal geometry at spacing y=15 showed 79% more convective transfer when compared to smooth channel.

For counter flow heat exchanger the heat transfer enhancement and friction factor depend on different spacing. Therefore, while deciding the geometry to be employed, we must consider the pumping power available for flows the fluid through the pipe and the purpose of the geometry. The decided geometry that provides the maximum heat transfer enhancement for a particular range of Reynolds number need not be the geometrical spacing that provides the lowest friction factor. Therefore, the channel needs to be analyzed with different geometrical spacing between inserts of decided pentagonal geometries for varying mass flow rate and the most suitable geometrical spacing based on whether we require an enhancement of heat transfer or an decrease in friction factor for the Reynolds number range of operation.

In this present research it is determine after all sort of investigation regarding heat transfer characteristics of various parameters for different geometrical linear spacing between two inserts with varying mass flow rate that as the spacing increase the quality of heat transfer characteristics for various parameters like Nusselt number, Reynolds number, Friction factor And Pressure drop will also decreases, as here we were using spacing y=15 and y=45. We achieve best results for linear geometrical spacing y=15 as compared to y=45 for counter flow.

The results of the inner Nusselt number from these values were compared to work done by experimental work results for plane tube.

 The friction factor decreases with increasing Reynolds number and the heat transfer coefficient increases with increase in Reynolds number in plane tube.

 The friction factor increased with inserting pentagonal shape inserts in concentric pipe heat exchanger as compared with plane type.

 Heat transfer enhanced by 25% to 38% in temperature outlet at various Reynolds numbers with using inserts at y=45 linear geometrical spacing.

 Heat transfer enhanced by 18% to 55% at various Reynolds number by using inserts at y=15 linear geometrical spacing.

 Increase of pressure drop is more by using y=15 geometrical spacing pentagonal shape inserts in cold pipe of concentric tube type heat exchanger compared to plane tube. Present study deals with effect on heat augmentation by the use of different latitudinal spacing with same geometry for similar thermal boundary condition in which water is used as a fluid and flow internally through the channel. If future scope is concerned then heat augmentation can be used:

- For air or any other fluid we use inside the heat exchanger.
- Analysis can be performed experimentally.
- Analysis can be performed for active and compound methods used for heat augmentation.
- Different thermal boundary conditions can be used for same geometries.

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