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CFD Study of Tube in Tube Heat Exchanger with Helical Insert of Different Height and Helical Groove

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Abstract

In the present study helical groove and helical insert of different height are introduced on the inside surface of inner tube of tube in tube heat exchanger, heat transfer and flow characteristics of the exchanger containing inserts and grooves are analysed and comparison is made. Width of insert and groove is taken 0.5mm and height of groove is taken 0.5mm, although for the inserts three different height are taken which are 0.5mm, 1mm, 1.5mm. Helix of pitch is 100 mm provided to all inserts and grooves. Heat exchangers with all the inserts and grooves are analysed numerically through ANSYS 14.5. Heat and flow parameter were calculated for four different Reynold number 1685.03, 3370.07, 5055.10 and 6740.13 corresponding to the flow rate of 50, 100, 150 and 200 LpH and then comparisons in made. By the study of results it is found that the values of heat transfer coefficient, pressure drop and friction factor are higher for heat exchanger with the insert then the exchanger with the groove. further heat transfer coefficient, friction factor and pressure drop all increases as the height of insert increases.

Keywords: Tube in tube heat exchanger, Helical groove, Helical insert, Nusselt number, Heat transfer coefficient.

INTRODUCTION

Device used for exchanging heat between a number of fluids is known as heat exchanger. Researcher are continuing there efforts in the direction of increasing heat transfer rate between two fluids of heat exchanger and there are many methods available such as introducing swirl at the entry of fluid, use of twisted tapes in fluid flow zone, vibrating the surface and fluid etc which increases heat transfer rate of an heat exchanger. H.F. Elattara et al.[1] studied multi tube in tube helical heat exchanger numerically in which coefficient of heat transfer is found maximum with three inner tubes, effectiveness of exchanger is also found maximum for three tubes. J.I. Córcoles-Tendero et al.[2] investigated numerically that when an elliptical corrugation is introduced in fluid flow region then heat transfer rate, Nusselt number and friction factor are increased in corrugated tube in comparison to simple tube. Erfan Khodabandeh et al [3] studied water graphene is used in spiral tube exchanger as nanofluid with different cross-section such as elliptical, rectangle etc and it is found that average Nusselt number was independent of shape of cross section although local Nusselt number is maximum in elliptical tube. M.A. Gómez et al. [4] investigated water heater tank fitted with helical coils through CFD. Internal movement of flow and transportation of energy inside and outside the fluid are investigated through Navier- stroke equations. Solution is carried by applying approximation of Bousinesq. Corrugated coils were used inside the tank and 150L water is used inside the tank. By the studies of the results it is found that results was closer to that obtained from experimental studies higher temperatures are find satisfactory during heating but have some dissimilarity during cooling and vice versa obtained i.e. lower temperatures are find satisfactory during cooling but not for the heating. Xiaoya Liu et al. [5] analysed the simple straight pipe with the cross shaped twisted tape, the purpose behind the use of tape is to increase turbulence inside the fluid flow zone. Analysis is performed on STAR-CCM+ a computational fluid dynamics software. Analysis is done for the Reynold number range from 40 to 1050, cross-twisted tape is provided four twist ratio of 2.0,3.0,4.0 and 1 with boundary condition of constant wall heat flux. By the studies of the results it has known that for the cross twisted tape local convective heat transfer increase when compared with the simple twisted tube. Nusselt number is increased by 151-195% for cross twisted tapes and performance evaluation criterion in increased by 90-123%. Zhouhang Li et al.[6] numerically analysed a helical tube equipped with four inserts at inner side of the tube in horizontal as well as vertical configuration and found that for both cases heat transfer performance was better than simple tube also heat transfer characteristics were increased further for vertical configuration. Siddhant Singh Yogesh et al.[7] investigated the heat exchanger installed with elliptical fins. Analysis is performed for different ellipticity ratio and for horizontal arrangements through computational fluid dynamics. Comparison is made with result of numerical study performed by Bhuiyan et al. (2013). At inlet section Reynold number is varied under the range starting from 1300 to 1600. Ellipticity ratio 0.6 and 0.8 is considered in order to analyse Colburn factor, friction factor and heat exchanger efficiency index. By the results it has known that Colburn factor decrease as ellipticity ratio and Reynold's number increases. Friction factor is found directly proportional to the inclination angle of the tube and elliptical ratio but is in inversed proportion with the Reynold number. Jundika C Kurnia et al. [8] studied entropy generation and heat transfer characteristics for helical coil. Cross section of helical coil is taken rectangular and analysis is performed for different radius of curvature. Boundary condition of constant value wall temperature and constant heat flux condition both analysed separately for the calculations of entropy generations and of coefficient of heat transfer. By the studies of result it has known that heat transfer coefficient increases as radius of curvature of square helical tube decreases and pumping power increases heat transfer also entropy generation was smaller for the tubes with smaller radius of curvature. Zhenya Duan ei al. [9] studied on the shell and tube exchanger with interrupted baffle plates and plates were joined by using different connection methods such as continuous connection methods or middle connection methods. Baffle plates are arranged at three different angles of 20^0, 30^0 and 40^0. By the analysis of results it is found the rate of heat transfer decreases for larger helix angel of baffles and smaller pressure drop. For per unit pressure drop heat transfer rate is maximum for 40^0 also the performance of continuous connection methods was better than the performance of middle connection methods. Kishor Kumar Sahu et al [10] investigated helical coil numerically as well as experimentally and concluded that Nusselt number pressure have improved then simple tube. R. Maradona et al studied [11] tube in tube heat exchanger through CFD and concluded that performance of helical exchanger was better than that of helical exchanger. Pramod S. Purandare et al. [12] studied heat transfer properties for the conical coil heat exchanger, in which tubes of heat exchanger are arranged in conical fashion with varying diameter, analysis is performed for the cone angle of tubes were taken 45 degree, 90 degree, 135 degree, 180 degree with three tubes of different diameters but for all the tubes average coil diameter is kept constant at 200mm and tube length is kept at 3meter. By the result it is found that as flow rate of fluid inside the tube increases Nusselt number also increases but as flow rate of shell side increases Nusselt number decreases. Effectiveness of heat exchanger depends on diameter of tube, cone angle, Reynold number and effectiveness decreases with the increase in Reynold number for tube side fluid. Effectiveness was found maximum helical coil when compared with the spiral coil and conical coil. Lastly friction factor also depended on cone angle and Reynold number. It decreases as Reynold number increases. Marco Colombo et al.[13] studied mixture of air water flow in helical heat exchanger and found maximum velocity in vicinity of centre of tube where air flows also separation of flow is found in the exchanger. Bodius Salam et al. [14] studied on straight copper tube with internal diameter of 26.6mm and outer diameter of 30 mm and length of test section is kept 900 mm. inside this copper tube a rectangular twisted insert is placed which is made up of stainless steel. Twist ration of 5.25 is provided to the insert. A rectangular section dimensions 8mm by 14 mm is cut from the twisted insert. The uniform heat flus condition are provided over test section. Analysis is carried for Reynold number in the range from 10000 to 19000 with heat flux ranging from 14 to 22 kilowatt per square meter. By the results it is found that Nusselt number is increased by factor 1.4 to 1.8 for the twisted insert when compared with smooth tubes also friction factor is increase 1.9 to 2.3 times in comparison to smooth tubes. S. Eiamsa-ard et al. [15] investigated flow in simple tube equipped with the twin tape type swirl generators in this setup two twisted tapes in one single passage with same twist ratio. Tests are conducted for four different twist ratios at Reynold number ranging from 3700 to 21000 under the boundary condition of uniform heat flux. Results were also compared with single twisted tape. By the studies of the results it is found that as twist ratio is decreases Nusselt number increases friction factor also increases. Counter flow generator have performed better than Cocounter flow swirl generator. For constant pumping power counter flow generator exhibits higher heat transfer rate than that of cocounter flow swirl generators. Also empirical relation is developed between Nusselt number and thermal efficiency index.

Nomenclature

A_c = Cross section area (m^2)

A_s = Surface area of inner tube (m^2)

c_p = Specific heat of hot fluid ($Wkg^{-1} K^{-1}$)

f = Friction factor

h = Heat transfer coefficient of hot fluid

H = Height of insert/ groove

k = Thermal conductivity ($Wm^{-1}K^{-1}$)

l = Length of tube (m)

LpH = Litre per Hour

$\dot{m}$ = mass flow rate of hot fluid(kg/sec)

Nu = Nusselt number

P = Pumping power (Watt)

Q = Rate of heat transfer

Re = Reynold number

t_b = Bulk meat temperature of hot fluid (K)

t_{hi} = Inlet temperature of hot fluid (K)

t_{ho}= Outlet temperature of hot fluid (K)

t_s = Surface temperature of hot fluid (K)

U_i = Velocity in x direction (m/sec)

v = Velocity (m/sec)

$\dot{v}$= Volume flow rate in LpH

$\dot{V}$= Volume flow rate in m3/sec

W = width of insert/ groove

∇P = Pressure difference for hot fluid (Pa)

Greek symbols

ε = Coil effectiveness

ρ_w = Density of water (kg/m^3)

ρ = Density (kg/m^3)

μ = Dynamic viscosity (Pa-s)

ν = Kinematic viscosity (m^2/sec)

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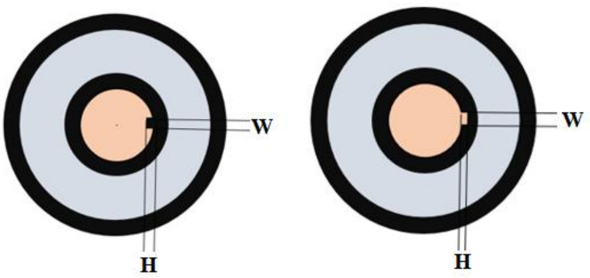
GEOMETRY DESCRIPTION AND MODELLING

Providing inserts in the tubes of heat exchanger has gained a lot of popularity in field of heat transfer enhancement. Hence in this study a helical insert of rectangular cross-section inside a straight tube in tube heat exchanger is investigated. Analysis is carried for varying height of insert and keeping width constant. Different height of insert taken into consideration are shown in the table 1. Fig 1 displays the geometry of insert and Fig 2 displays geometry of inner tube with insert placement. For simplicity material of insert is kept same as material of tubes. In this study a straight tube in tube heat exchanger is considered with ID of inner tube 10.5mm, OD of inner tube 12.5mm, ID of outer tube 27.5mm and OD of outer tube 33.8mm. A insert of rectangular cross-section is introduced inside the inner tube of heat exchanger. This insert is placed in helical arrangement with the pitch of 100mm.



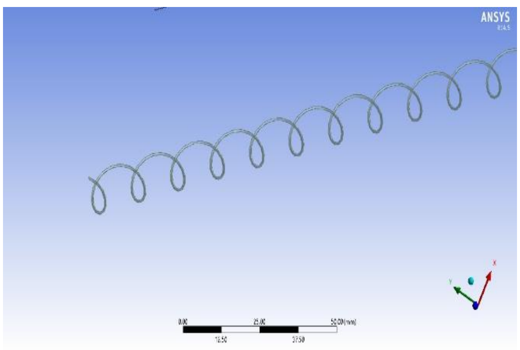
(a) **(b)**

Figure 1. Cross section of (a) tube with insert (b) tube with groove.

Table 1 Dimension of Inserts

S. No.	Height (mm)	Width (mm)
1	0.5	0.5
2	1.0	0.5
3	1.5	0.5

Simulation is performed on CFD software package of ANSYS 14.5. Meshing is performed in ANSYS ICEM-CFD. Sizing of grid is performed by command Body sizing with 1mm for outer tube, outer fluid and for inner tube. Body sizing of 0.5mm is selected for inner fluid. Mapped mashing is done for outer tube and outer fluid. Details of elements of meshing is given in the table 2.



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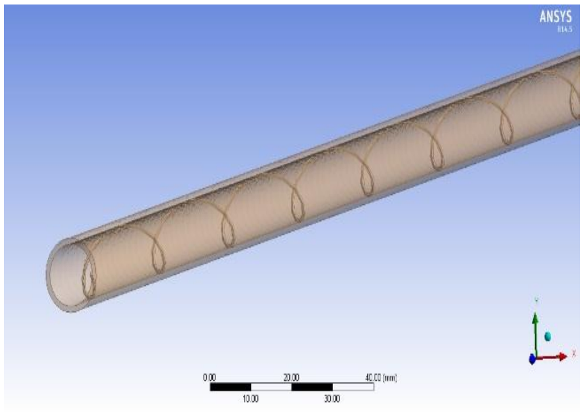

(b)

Figure 2. (a) Helical insert (b) helical insert placed inside the inner tube.

Table 2. Nodes and elements of grids

Type of tube	Nodes in meshing	Elements in meshing
simple tube	1808516	1483573
Tube with insert of ratio 1	1765023	1408086
Tube with insert of ratio 2	1768493	1416299
Tube with insert of ratio 3	1728233	1374139
Tube with groove of ratio 1	1985788	1602116

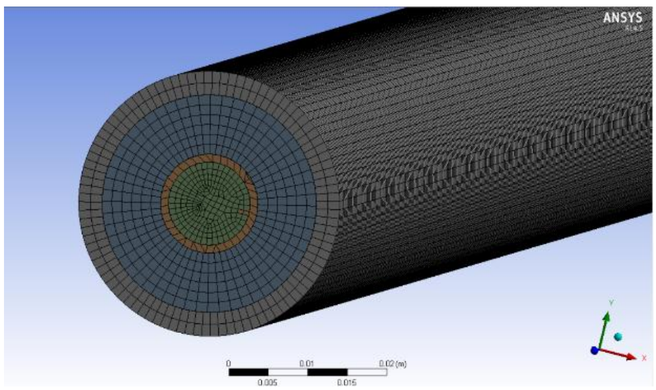

Figure 3. Grid of tube with insert.

NUMERICAL METHODOLOGY AND MATHEMATICAL FORMULATION

Governing equation:-

In present numerical study. 3 Dimensional governing equations are applied with flow specification of turbulent flow. In this study helical inserts are used in order to increase the turbulence which in turns will result in increased rate of heat transfer. Disturbance in fluid caused by the inserts will require k-ε turbulence model to find the heat transfer and flow characteristics. Hence k-ε turbulence model is selected in which two equations one for turbulent kinetic energy and one for dissipation rate is solved. Other than equation of k-ε model following are the governing equations which are used during the analysis:-

1. $$\frac{\partial U_i}{\partial x_i} = 0.$$

2. $$\rho U_i \frac{\partial U_i}{\partial x_j} = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \rho\overline{u'_i u'_j}\right].$$

3. $$\rho c_p U_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\lambda\frac{\partial T}{\partial x_j} - \rho c_p \overline{u'_i T'}\right].$$

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

Where σ_k , σ_ε, $C_{1\varepsilon}$ are simulation coefficient and there values are 1.0, 1.2, 1.9 and 1.44 respectively. $\rho c_p u'_i T'$ represents turbulent heat flux, $\rho u'_i u'_j$ represents average Reynold stress and fluctuating velocities and temperatures of turbulent flow are represented by u'_i, u'_j and T' respectively. G_b represents buoyancy generation and turbulent kinetic energy generation caused by mean velocity gradient is represented by G_k. in compressible turbulence dilation caused by fluctuations to overall dissipation rate is given by Y_M. These governing equations are utilized by FLUENT package of ANSYS. To obtain results nearest to preciseness the solution is made up to convergence where variables seems to be constant against the iterations.

Boundary conditions

Boundary conditions used in the present analysis are as follows: turbulent flow is considered for both hot and cold fluids. Velocity is specified normal to boundary, no slip conditions are applied to the surfaces of tubes at outlet section of tube atmospheric pressure is considered i.e. zero gauge pressure at outlet. Copper is used as material for the tubes. Properties of copper are as follows; Density 8978 kg/m^3, Thermal conductivity 387.6 W/m-K and specific heat capacity is 381 J/kg-K. Working fluid is taken water for both hot and cold conditions. The properties of water are taken as Viscosity 0.001003 kg/m-s, Density 998.2 kg/m^3, Specific Heat Capacity 4182 J/kg-K and Thermal Conductivity is taken as 0.6 W/m-K. Outside surface of outer tube is considered adiabatic during the analysis. Hot flow rate is taken 50, 100, 150 and 200 in LpH as taken by R Maradona et. al [11] this hot fluid is made to flow in inner tube which has provided surface modification and Reynolds number are calculate for each of flow rates which are 1685.03, 3370.07, 5055.10 and 6740.13 which are corresponding to flow rates 50, 100, 150 and 200 in LpH respectively. Flow rate of cold fluid is taken 200LpH and kept constant' Temperature of hot fluid is taken 328K and for cold fluid is taken 305K.

Mathematical formulation for hydraulic and thermal performance parameters:-

1. Volume flow rate (conversion from LpH to m^3/sec)-

   $\dot{V} = \dot{v}/(36*10^5)$

2. Volume flow rate-

   $\dot{V} = A_c v$

3. Mass flow rate-

   $\dot{m} = \rho_w A_c v$

4. Rate of heat transfer by hot fluid-

   $Q = hA_s(t_s - t_b)$

5. Heat lost by hot fluid-

   $Q = \dot{m}c_p(t_{hi} - t_{ho})$

6. Heat transfer coefficient-

   $h = \dot{m}c_p(t_{hi} - t_{ho})/ A_s(t_s - t_b)$

7. Friction factor-

   $f = 2\nabla Pd/\rho_w l v^2$

RESULT AND DISCUSSION

Contours of temperature for tube with insert of height 1.5mm and Reynold number 6740.13 (Flow rate 200LpH)

In figure 4 development of temperature is shown. Contours are drawn at separation of 165mm from each other. These temperature contours are of hot fluid flowing in the tube with insert of height 1.5 mm and with the flow rate of 200LpH (corresponding Reynold number is 6740.13). From the figure 4 it is found that temperature drop initiates at the nearby region of the insert and this drop of temperature grows towards the center. Since fluctuation in flow field and turbulence are maximum in the region of tube near the insert hence in this region temperature drops suddenly and then grows towards center of tube.

Validation

Before analysis of proposed work, a validation is done against some previous work. For this purpose data of h and Nu is collected for a simple straight tube in tube heat exchanger. In figure 5 (a) values of Nu from current study and from the work of R Maradona et al.[11] In figure 5 (b) values of h from current study are compared with the values obtained by R Maradona et al.[11] After analyzing the data obtained it is concluded that that difference was within 3% in the current study and study of R Maradona[11].

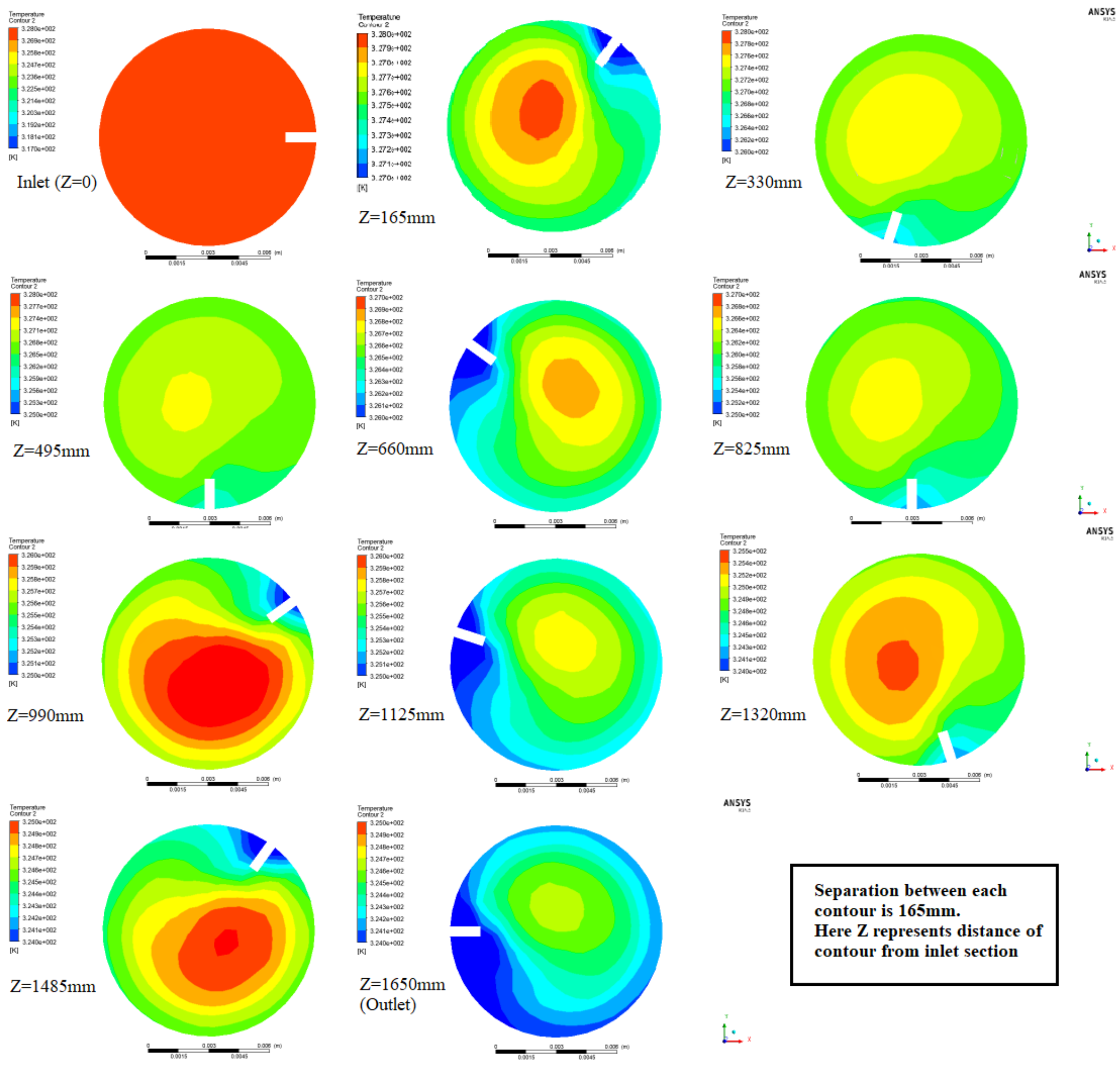

Figure 4. Contours of temperature for tube with insert of height 1.5mm

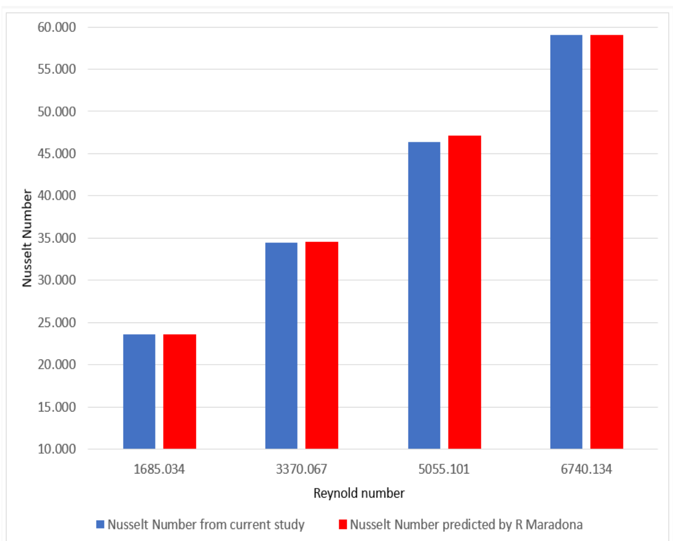

Figure 5 (a) Validation of Nusselt number

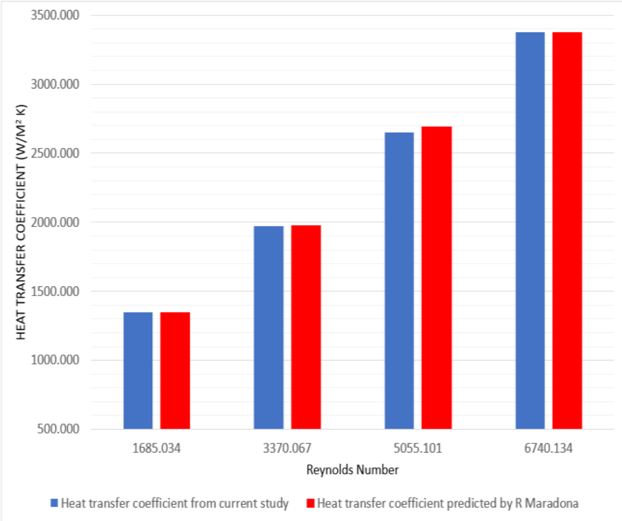

Figure 5 (b) Validation of Heat transfer coefficient

Effect of ratio of height of insert and Reynold number on temperature difference between inlet and outlet section of hot fluid:-

In current study data of temperature of hot fluid are collected for helical inserts of different heights, for four Reynold numbers which are 1685.03, 3370.07, 5055.10 and 6740.13. difference in outlet and inlet temperature are shown in the figure 6. By the study of figures it is found that as Reynold number increases difference in temperature decreases as at fluid higher Reynold number fluid gets less time inside the exchanger. Also as insert height increases temperature drop decreases because of increased turbulence.

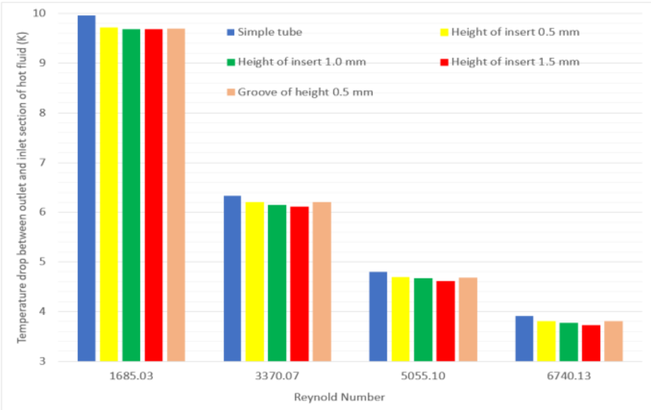

Figure 6 Variation of temperature drop with Reynold number for different tubes.

3.3 Effect of height of insert and Reynold number on heat transfer coefficient

In the present analysis helical inserts are provided to introduce turbulence in the flow field of hot fluid in tube in tube heat exchanger, which results in increase in heat transfer coefficient. The data of heat transfer coefficient is collected for four Reynold numbers 1685.03, 3370.07, 5055.10 and 6740.13 corresponding to the flow rates 50, 100, 150 and 200 LpH respectively. Effect on the heat transfer coefficient are illustrated in the Figure 7 . by the study of the graph it can be said that with increase in Reynold number heat transfer coefficient is also increased because as Reynold number increases turbulence in the fluid also increases which causes fluctuations in momentum and energy between different layers of fluid. Although at smaller Reynold number 1685.03 and 3370.067 viscous effect are still dominating over inertial fluctuations even with the insert hence there is not much difference heat transfer coefficient for all types of tubes. It is also seen that for insert height of 0.5 mm the coefficient of heat transfer is higher for tube with insert than that for the tube with groove of same helix. Further when comparison is made between tubes with inserts of different height than it is found that as height of inserts increases the value of coefficient of heat transfer also increases because higher inserts produces greater disturbance in the flow field which causes better mixing of fluids. At each Reynold number heat transfer coefficient is maximum for the tube with insert of height 1.5 mm.

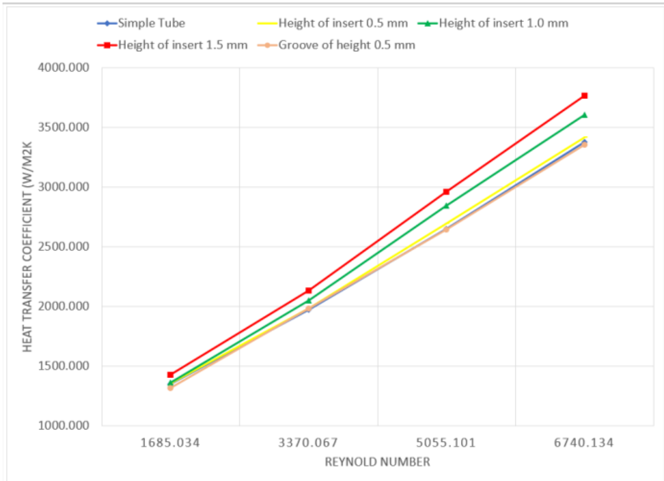

Figure 7 Variation of h with Reynold number for different tubes.

Effect of ratio of height of insert and Reynold number on Pressure drop between inlet and outlet section of fluid:-

Effect on the pressure drop are shown in the Figure 8. At Reynold number 1685.03 there in not much difference in the pressure drop between all types of tubes because at smaller Reynold number even with the presence of insert the viscosity effect dominates and inertial effects of fluid particle could not play any significant role in heat transfer characteristics. As Reynold number increase higher pressure is required to maintain the higher velocity hence in all type of tube pressure at inlet increases with the Reynold number and thus pressure drop also increases. As the data for tube with insert and groove of height 0.5 mm are analyzed then it is found that pressure at inlet is more for tube with insert then the tube with groove of same size and hence pressure drop is more for the same. Further analysis leads to the fact that in between tubes with different insert heights, the pressure drop increase as the height of increases because the maximum resistance to flow is offered by the tube with insert height 1.5mm hence pressure at inlet and pressure difference both are found maximum for tube with insert of height 1.5 mm at each Reynold Number.

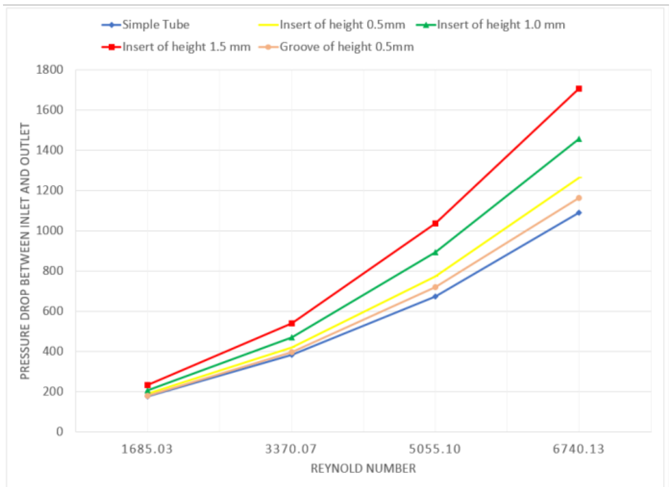

Figure 8 Variation of Nu with Reynold number for different tubes.

Effect of ratio of height of insert and Reynold number on Friction factor:-

Data of friction factor against the Reynold number are shown in the figure 9. From the study of figure it is found that frication factor has decreased with the increase in Reynold number, which also the expected result as friction factor is in inverse proportion with the Reynold number. Also height of insert have affect on the friction factor. As height of insert increases friction factor has increased for each of the Reynold number and friction factor is maximum for insert ratio 3 at Reynold number 1685.03.

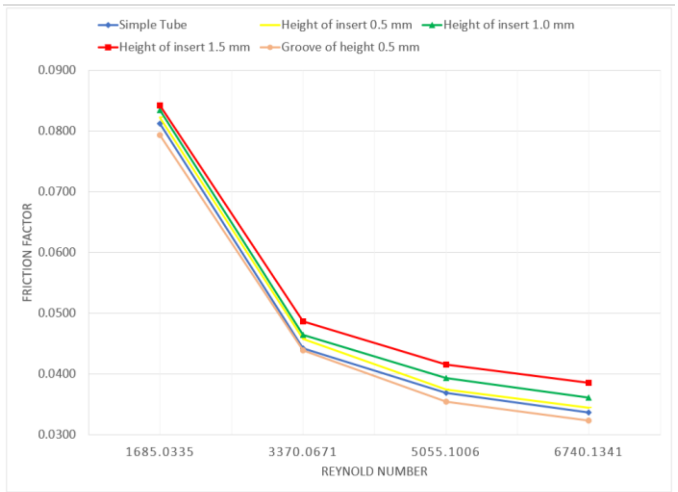

Figure 9 Variation of Friction factor with Reynold number for different tubes.

CONCLUSION

In the present study a tube in tube heat exchanger with modification on inside surface of inner tube is analyzed through CFD package of ANSYS 14.5. Modification on inner surface are introduced in order to increase surface roughness of inner tube which also results in greater turbulence inside fluid flow zone. Modification are done by providing insert of rectangular shape in helical fashion and height of insert is varied also analysis is done for the grooved tube of same shape. Different heat and flow parameters are calculated for all the proposed tubes separately and comparisons is made. By the findings of study it can be concluded that heat transfer coefficient was better for tube with insert then the tube with the groove for same height also as height of inserts increases heat transfer coefficient increases as greater turbulence is created by the insert of increased height. Pressure drop is also found more for the tube with insert than the tube with groove of same size. Further it is also seen that as height of insert is increased pressure drop in inner tube is increased because insert with larger height provides more resistance to flow. This also results in greater value of friction factor for larger height of inserts.

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