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Numerical Simulation of Tube inTube Heat Exchanger withTwisted Tape Insert

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Abstract: *In this article without application of external power we can enhance the heat transfer rate by modifying the design by providing the twisted tape insert in tubes, extended surface or swirl flow devices. The turbulent flow model with counter flow heat exchanger is considered for analysis purpose. The P/d ratio is 4, 5, 7. Nusselt number, friction factor, pumping power required and LMTD variation of inner fluid with respect to Reynolds number is found out for different P/d ratio. The optimize Reynolds number for maximum heat transfer and minimum power loss is found out by graph intersection methods. From the results complicated behavior of fluid flow is captured for both the fluids flowing inside the tube. With increases in P/d ratio the Nusselt number will decreases and the outer wall boundary condition does not have any significant effect on the inner Nusselt number. The Darcy friction factor decreases with increase in Reynolds number. The Pumping power increases with increase in Reynolds number for all the condition of P/d ratio and for all the boundary conditions. Log mean temperature difference (LMTD) increases at a steady rate with increase in Reynolds number. The optimization point between Nu and pumping power with respect to Re; shifts toward the higher Reynolds number with increase in P/d ratio.*

Keywords: *Nusselt number, Reynolds number, friction factor, pumping power, LMTD and pitch diameter ratio (P/d ratio).*

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I. Introduction

Effective utilization, conservation and recovery of heat are critical engineering problems faced by the process industry. A majority of heat exchangers used in thermal power plants, chemical processing plants, air conditioning equipment, and refrigerators, petrochemical, biomedical and food processing plants serve to heat and cool different types of fluids. In recent years, many experimental and numerical works has been done to enhance the heat transfer rate. The twisted tape inserts are new addition to the family of inserts for enhancement of heat transfer. For the twisted tape insert, the swirl moves in one direction along the twisted tape and induce swirl in the flow, which increase the retention time of the flow and consequently provide better heat transfer performance over twisted tape insets. The high heat transfer with twisted tape inserts is also accompanied by a higher pressure drop across the flow.**Manglik et al.[1]** has studied and developed the Laminar flow correlations for friction factor 'f' and Nusselt number 'Nu' based on experimental data for water and ethylene glycol with tape inserts of three different twist ratio.Uniform wall temperature condition has been considered which represents practical heat exchangers in the chemical and process industry.**Rao et al. and others [2-7]** has analyzed the experimental and numerical data of double pipe heat exchanger with helical coil and twist tape insert and compared both the resultnumerical and experimental work on considering fluid to fluid heat transfer.**Mun-oz-Esparza et al. [8]** employed the CFD simulation package and investigated the heat transfer and fluid flow performance inside a circular pipe with the helical wire coils inserts. **Jamshidi et al. and Xing et al. [9-10]** they performed experimental work to enhance the heat transfer in shell and helical tube heat exchanger. In the helical tube section of the heat exchanger hot water flows. **Srbislav et al. [11]** has performed the experimental work to predict the performances of warmth exchangerswith turbinatetube coils.**Yang et al. [12]** has performed experimental work to predict the characteristics of convective heattransfer in turbinate coil device.

Nomenclature

A = cross sectional area, m^2
C_p =Specific heat, J/kg-K
d_i = d = inner diameter of inner tube, mm
d_o = outer diameter of inner tube
D_i = inner diameter of outer tube
D_o = outer diameter of outer tube
f = friction factor
h = heat transfer coefficient, W/m2k
k = thermal conductivity, W/m-K
L = length of pipe, mm
m = mass flow rate, kg/s
Nu_x = Local Nusselt Number
Nu = Average Nusselt number
P = pitch of twisted tape, mm
ΔP = pressure drop, N/m2
Pr = Prandtl Number
Q = heat flux, W/m2
Re = Reynolds Number
Re_{sw} = Reynolds Number corresponding to swirl parameter
Sw = Swirl parameter
T_{hi} = temperature of hot fluid at inlet, K
T_{he} = temperature of hot fluid at outlet, K
T_{ci} = temperature of cold fluid at inlet, K
T_{ce} = temperature of cold fluid at outlet, K
T_w = wall temperature, K
T_f = fluid mean temperature, K
u_i ,$u_j$and u_k= velocity in x, y, z directions
V = velocity of fluid flowing in tube, m/s

Greek symbol

μ = dynamic viscosity, Kg/ms
μ_w = dynamic viscosity at wall, Kg/ms
δ = density
ρ= thicknes of tape.

Hossain et al. [13] has performed the Finite Element based model study of the heat transfer problem. The investigated without insert and with insert i.e. combination of horizontally and vertically arranged rectangular boxes of 5 mm thickness are being fitted perpendicular to the flow direction respectively at equal distance from each other along the length.**Rahimi et al. [14]** has performed the numerical simulation of the warmth transfer and friction issue characteristics of a circular tube fitted with V-cut twisted tape. **Paisarn et al. [15]** has performed the experimental simulation on spiral coil device underneath cooling and dehumidifying conditions. **Pawar et al. [16]**has performed the experiment on equal steady state and non-isothermal unsteady state conditions in spiral coils for Newtonian put together as for non-Newtonian fluids. **Bharadwaj et al. [17]**has investigated the heat transfer and fluid flow characteristics of a rounded tube fixed with lots of inserts arranged in co-swirl and counter-swirl orientations. **Lia et al. [18]**has worked on new tube insert, named centrally hollow slender twisted tape, is developed and its impact on the warmth transfer improvement performance.**Tang et al. [19]** has experimentally compared and performed numerical simulation for turbulent flow characteristics and heat transfer performances of twisted tri-lobed tube.**Zhi-jiang Jin et al. [20]** has performed numerical simulation on spiral corrugated tube used in coaxial heat exchangers and has given a correlation of the six-start spirally corrugated tube for heat transfer calculation. **Chunbao Liu et al. [21]**has done theoretical optimization to develop a plate-fin heat exchanger for the hydraulic retarder. **Azmiet al. [22**] has worked on a numerical model for turbulent flow of Nano-fluids in a tube with twisted tape inserts.**Vashistha et al. [23]**has an experiment determined the pressure drop and heat transfer characteristics of flow of water in a passing 75-start spirally grooved tube with twisted tape insert.**Gorman et al. [24]**has performed numerical simulation on fluid flow in a double-pipe heat exchanger in which the wall of the inner pipe is helically corrugated.

II. Objective Of The Work

To determine and compare the Nusselt number, pumping power, friction factor for plane tube and tube with twisted insert for different p/d ratio in laminar and turbulent region.
To find out the optimization point for different p/d ratio between Nusselt number and Pumping Power.

III. Problem Formulation

In the literature survey it was found that so much attention has been done to enhance the heat transfer rate in heat exchanger. However, no work has been reported to optimize the heat transfer rate with respect to pumping power consumption. In this work heat transfer rate has been optimized using twisted tape insert, keeping in mind that it should produce maximum heat transfer rate with minimum power consummation.
In recent past few years, study of heat exchanger with parallel flow and counter flow has been done to predict the heat transfer behaviour. In this study counter flow heat exchanger has been considered as it is known that counter flow heat exchanger is an efficient heat exchanger as compared to all other heat exchanger. In counter flow heat exchanger the cold fluid and the hot fluid flow in opposite directions in their respective tube as shown in fig (3.1). Both the hot fluid and cold fluid flows with different velocity as per different Reynolds number. The flow velocity of hot fluid is remained constant and the cold fluid flow rate is varied to find the heat transfer rate, friction factor and pumping power.

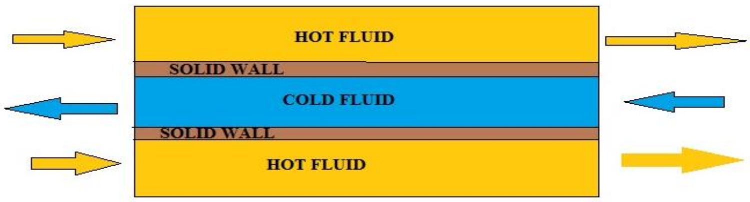

Fig 3.1: The fluid flow arrangement of counter flow heat exchanger.

Problem specification
(3.1) the specifications of tube in tube heat exchanger has been tabulated below in table

Length of tube, L	2.2 m
Inner diameter of inner pipe, d_i	0.022m
Outer diameter of inner pipe, d_o	0.026m
Inner diameter of outer pipe, D_i	0.054m
Outer diameter of outer pipe, D_o	0.058m
Material	Copper
Inner pipe fluid	Cold water
Annulus fluid	Hot water

Table 3.1: Specifications of heat exchanger without insert

3.1 Boundary Conditions:

Velocity inlet condition was considered for inlet cold fluid. Here the velocity of cold fluid has been estimated corresponding to following Reynolds number as 800, 1200, 1600, 2000,4000, 6000, 8000, 10000 respectively. The cold fluid temperature at inlet was considered as 300 K.
Similarly for the hot fluid **velocity inlet** condition was considered. Thehot fluid flow rate was assumed to be constant corresponding to the Reynolds number as 10000 for all the condition of fluid flow. The temperature of the hot fluid at the inlet is considered as 353 K.

Pressure outlet boundary condition is considered for outlet hot and cold fluid. At the outlet the gauge pressure was taken as zero atmospheric pressure.

Insulated wall condition has been considered for the side wall of heat exchanger as there is no heat transfer takes place to and fro from the side wall of the heat exchanger.
The fluid properties of the working fluid (water) were assumed to be constant throughout the analysis with respect to temperature and are presented in the table (3.2)

VISCOSITY	0.001003	kg/m-s
DENSITY	998.2	kg/m3
SPECIFIC HEAT CAPACITY	4182	J/kg-K
THERMAL CONDUCTIVITY	0.6	W/m-K

Table 3.2: Properties of working fluid (Water)

The tube of the heat exchanger was made up of copper to maximize the heat transfer, because copper has good thermal conductivity. Also the properties of the copper were also remains constant throughout the analysis. It is represented in table (3.3)

DESCRIPTION	VALUE	UNITS
DENSITY	8978	kg/m3
SPECIFIC HEAT CAPACITY	381	J/kg-K
THERMAL CONDUCTIVITY	387.6	W/m-K

Table 3.3: Properties of Solid pipe material (Copper)

Basic assumptions
For the analysis following assumption have been considered
1. Outer wall thickness is neglected for simplifying the numerical calculation.
2. Steady state heat transfer conditions have been assumed.
3. Incompressible fluid with constant fluid property.
4. Natural convention and Radiation heat transfer was neglected.
5. Conjugate heat transfer between the two fluids was considered.
6. Counter flow heat exchanger was considered.

3.2 Governing Equations:

The mathematical equations used to describe the flow of fluids are the continuity and momentum equations, which describe the conservation of mass and momentum. The momentum equations are also known as the Navier-Stokes equations. For flows involving heat transfer, another set of equations is required to describe the conservation of energy.
The problem under consideration is assumed to be steady, three dimensional, & incompressible. The CFD modeling involves numerical solutions of the conservation equations for mass, momentum and energy. These equations for incompressible flows, under all the assumptions can be written are as follows:

Mass Conservation: $\frac{\partial}{\partial x_i}(\rho u_i) = 0$

Momentum conservation: $\frac{\partial(\rho u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_k}{\partial x_k}\right)\right]$

Energy conservation: $\frac{\partial}{\partial x_j}\left(\rho u_j C_p T - k\frac{\partial T}{\partial x_j}\right) = 0$

Heat transfer coefficient is obtained by equating the conduction heat transfer to the convection heat transfer; $q_{cond} = q_{conv}$

$$h = \frac{-K\frac{\partial T}{\partial x}}{T_w - T_f}$$

Local Nusselt number is given by; $Nu_x = -\frac{\frac{\partial T}{\partial x}D_h}{T_w - T_f}$

Then the average Nusselt number can be found by following relation; $N_{avg} = \frac{1}{L}\int_0^L Nu_x\, dx$

Friction factor is given by: $f = \frac{2\times\Delta P\times d}{\rho\times L\times V^2}$

Length of pipe is given by following relation; $L = \pi\sqrt{H^2 + (\pi D)^2}$

Log Mean Temperature Difference for counter flow heat exchanger can be presented by following relation and the figure shows in fig 3.2.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}}$$

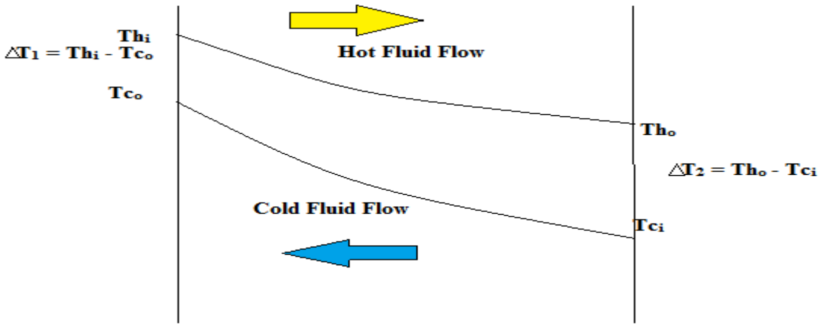

Fig 3.2: Temperature distribution curve of counter flow heat exchanger.

Where $\Delta T_1 = T_{hi} - T_{co}$ & $\Delta T_2 = T_{ho} - T_{ci}$
To find the outlet fluid temperature we can use the energy balance equation,

$$Q = mC_p[T_1 - T_2] = hA_s[T_w - T_f]$$

3.3 CFD Procedure

Geometry creation for tube in tube heat exchanger

1. Select a plane and sketch a circle of 22mm as diameter.
2. Extrude that circle with thickness 2mm to a depth of 2200mm.
3. Draw another circle with diameter 54mm.
4. Extrude this circle up to 2200mm as frozen solid.
5. Use Boolean operation and subtract pipe from solid cylinder by applying preserve tool bodies we get 3 solids. And the figure 3.3 shows the geometry of tube in tube heat exchanger.

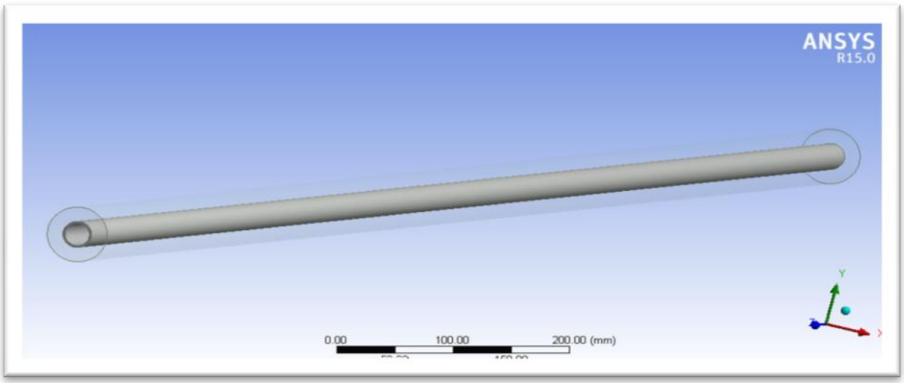

Fig 3.3: The geometry of tube in tube heat exchanger.

3.4 Grid Generation

In grid generation(shows fig. 3.4) first go to the MESH option located in FLUENT tree, and then press the GENERATE MESH button. It will create automatic grid. As the mesh which is created automatically is not as fine as we require so we have to modify the grid or make the grid finer so that accurate results will come. Proximity and curvature is used for advanced size function, course is used for relevance center. Triangular surface masher is used under patch conforming. Face refinement for both inlet and outlet face was done separately as both the faces were not fine. Number of nodes and elements were found to be 457945 and 1885995.

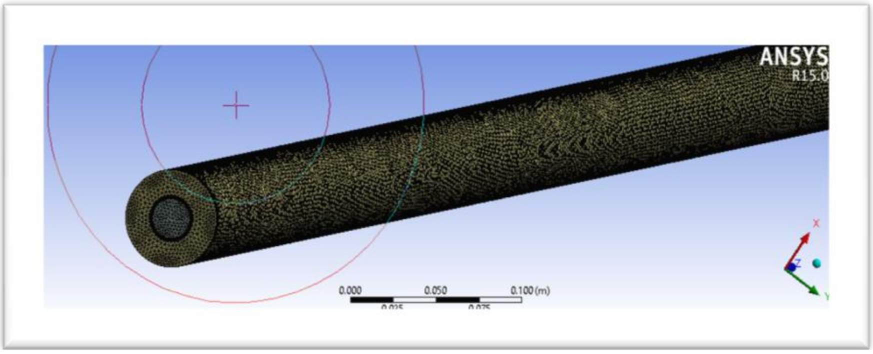

Fig 3.4: Grid generation for the tube in tube heat exchanger

Boundary condition

At **inlet cold**, temperature at inlet cold is constant that is 300k and turbulent intensity is given by $0.016R_e^{-1/8}$ and hydraulic diameter is the inner diameter of inner pipe.

At **inlet hot**, velocity based boundary condition is provided in which velocity is constant as the Reynolds number is constant that is 10000. Turbulent intensity is given by $0.016R_e^{-1/8}$ and hydraulic diameter is the inner diameter of inner pipe.

For **outlet cold and hot**, pressure based boundary condition is considered that is 0 gauge pressure and same intensity and hydraulic diameter is given that of inlet.

Solution methods

Simple pressure velocity coupling scheme is used, Least squares cell based gradient is used, and Standard pressure is taken in account. Second order upwind is used for momentum and first order upwind for turbulent kinetic energy and turbulent dissipation rate.

The criteria for convergence are 0.001 for continuity, x-velocity, y-velocity, z-velocity, k and epsilon and 10^{-6} for Energy. Standard Initialization is used and we are computing the solution from the inlet cold boundary condition and finally we provide the number of iteration for the convergence.

Grid independence test

Grid independence or convergence test (shows in fig. 3.5) is an important task for any numerical simulation. In order to find the accuracy of the result obtained one needs to do grid independence test. It is done by gradually refining the mesh and then is simulated using a commercial CFD package. If the results of the simulations do not alter much as compare to previous value, then the results are considered as independent of the mesh created.

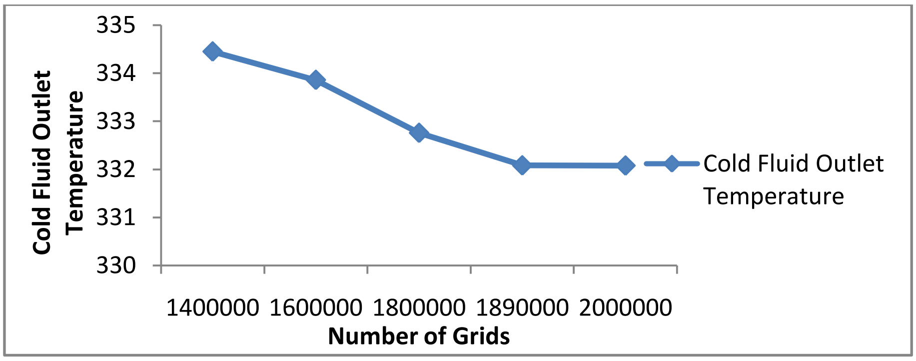

Fig 3.5: Grid Independence Test of Model based on Cold Fluid Outlet Temperature.

Geometry creations for tube in tube heat exchanger with twisted tape insert

1. Select a plane and sketch a circle of 22mm as diameter.
2. Extrude that circle with thickness 2mm to a depth of 2200mm.
3. Draw another circle with diameter 54mm.
4. Extrude this circle up to 2200mm as frozen solid.
5. Use Boolean operation and subtract pipe from solid cylinder by applying preserve tool bodies we get 3 solids.
6. Select plane on the circular face of the tube and draw a horizontal line of 20mm

7. Select plane on the axis of pipe and draw a line of length 2200mm.
8. Using sweep tool and selecting 20mm line as profile and 2200mm line as path and entering pitch value we get the twisted tape. (shows in fig. 3.6 and fig. 3.7)

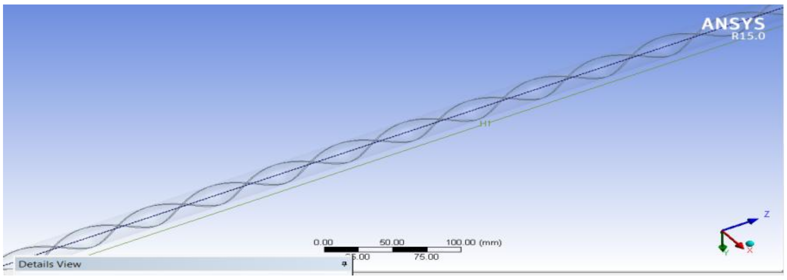

Fig 3.6: Showing the geometry of twisted tape insert created in ANSYS 15 work bench.

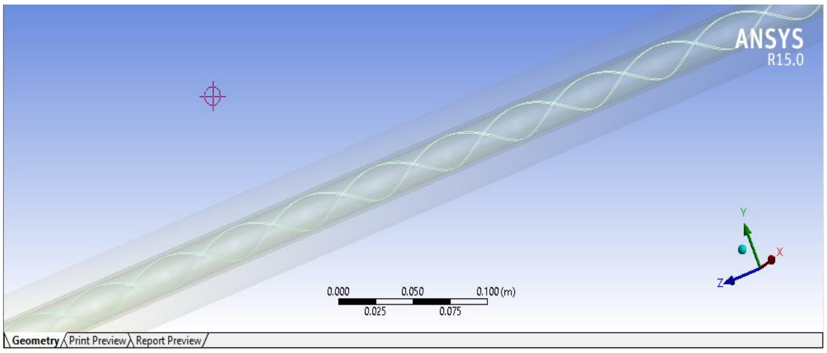

Fig 3.7: Showing the geometry of tube in tube heat exchanger with twisted tape insert created in ANSYS 15 work bench.

Grid generation

In fig. 3.8 Proximity and curvature is used for advanced size function, coarse is used for relevance center, medium smoothing is used .triangular surface masher is used under patch conforming .as the mesh was not fine so I have done face refinement for both inlet and outlet face .number of nodes and elements were found to be 645993 and 2766859.

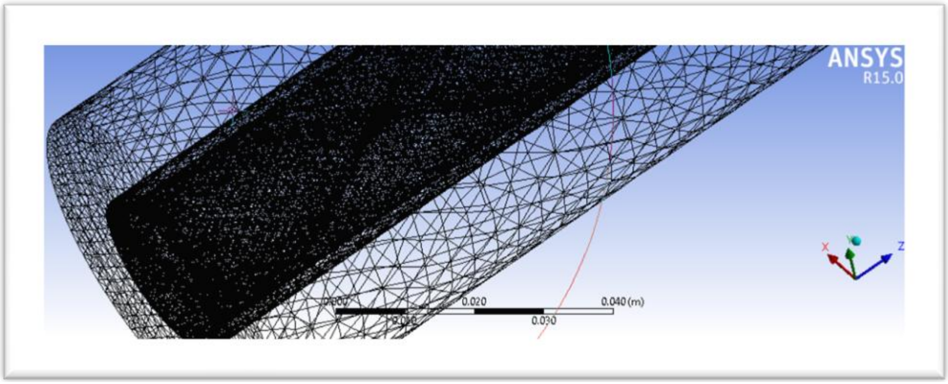

Fig 3.8: Grid generation for the double tube heat exchanger with twisted tape insert

Solution methods

Criteria for the convergence are 0.001 for continuity, x-velocity, y-velocity, z-velocity, k and epsilon and 10^{-6} for Energy. Standard Initialization is used and we are computing the solution from the inlet cold boundary condition and finally we provide the number of iteration for the convergence.

Grid independence Test.

Grid independence or convergence test (shows in fig. 3.9) is an important task for any numerical simulation. In order to find the accuracy of the result obtained one needs to do grid independence test. It is done by gradually refining the mesh and then is simulated using a commercial CFD package. If the results of the simulations do not alter much as compare to previous value, then the results are considered as independent of the mesh created.

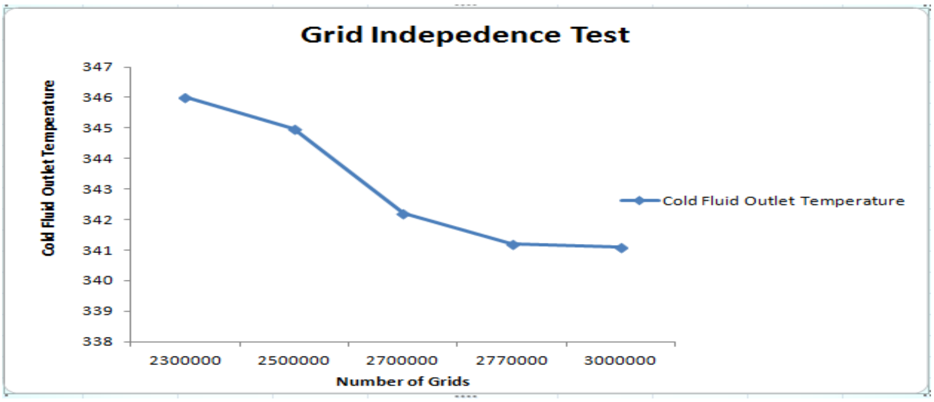

Fig 3.9: Grid Independence Test of Model based on Cold Fluid Outlet Temperature.

3.5Data Reduction:

The area weighted average temperature and static pressure were noted at the inlet and outlet surfaces of the pipe. The friction factor and average heat transfer coefficients were calculated as follows.

Friction Factor: $f = \Delta P/2\big((L/D)\rho V^2\big)$

Nusselt Number

$$Q_1 = m \times C_p \times [T_{ce} - T_{ci}]$$
$$Q_2 = m \times C_p \times [T_{hi} - T_{he}]$$
$$Q_{avg} = \frac{Q_1 + Q_2}{2}$$
$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\log \frac{\Delta T_1}{\Delta T_2}}$$

Where $\Delta T_1 = T_{hi} - T_{ce}$ and $\Delta T_2 = T_{he} - T_{ci}$

$$h = \frac{Q_{avg}}{A \times LMTD}$$
$$A = \pi \times d_i \times L$$

IV. Results And Discussion

4.1 Validation By Analytical Formula:

4.1.1 Twisted Tape in Laminar Flow:

Manglik and Bergles [1] developed the correlation for Nusselt number for laminar flows including the swirl parameter, which defined the interaction between viscous, convective inertia and centrifugal forces. The heat transfer correlation,

$$\boldsymbol{Nu} = .106 S_w^{.767} Pr^{0.3} \left(\frac{\mu}{\mu_w}\right)^{.14}$$
$$S_w = \frac{Re}{Y^{0.5}}$$

4.1.2Twisted tape in Turbulent Flow:

Manglik and Bergles developed the correlation for friction factor and Nusselt number for turbulent flows. Their correlations are as follows. The heat transfer correlation,

$$\mathbf{Nu} = .023 Re^{.8} Pr^{.4} \left(\frac{\pi}{\pi - \frac{4\delta}{d}}\right)^{0.8} \left[\frac{\pi + 2 - 2\delta/d}{\pi - 4\delta/d}\right]^{0.2} \left[\frac{\mu}{\mu_w}\right]^{0.18}$$

For **validation** tube in tube heat exchanger with twisted tape insert has been considered. In particular case P/d ratio is considered as 5 and as the value of d is constant that is 22, therefore the value of P in this case is 110mm.result has been tabulated as given below: and the fig. 4.1 show the Validation of CFD Result with Analytical Result given by Manglik and Bergles.

Reynolds Number		Nusselt Number Using CFD	Nusselt Number From Analytical Result [1]
Laminar Region	800	22.016	24.98
	1200	25.204	28.09
	1600	27.817	29.15
	2000	29.48	31.6
Turbulent Region	4000	34.53	36.88
	6000	41.109	42.75
	8000	45.147	47.98
	10000	47.84	51.56

Table 4.1: Validation of CFD Result with Analytical Result.

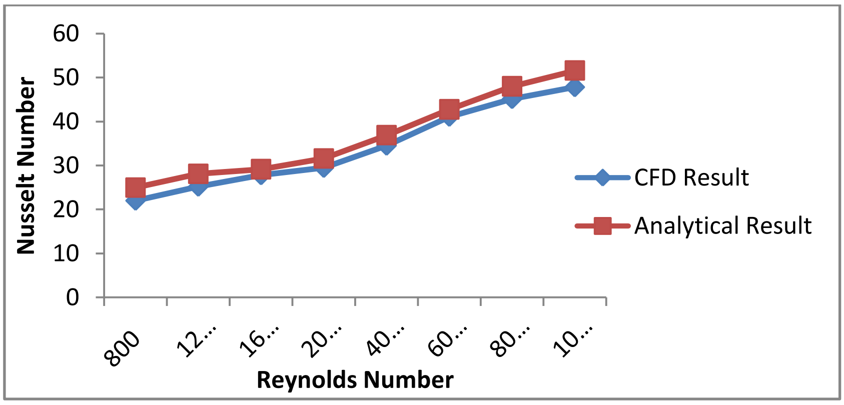

Fig 4.1:Validation of CFD Result with Analytical Result given by Manglik and Bergles

4.2 Validation by Experimental Result

Manglik and Bergles [1] have developed Nusselt number and friction factor correlations based on experiment data for water with tape insert. Depending upon flow rate and type of geometry, the enhancement in heat transfer is due to the tube partitioning and flow blockage, longer flow path and secondary fluid circulation. The onset of swirl flow and its intensity is determined by a swirl parameter.

The fig 4.2 shows the Nusselt number values suggested by Manglik and Bergles experimental data which closely matches with CFD results. And comparison with CFD shows in fig. 4.3.

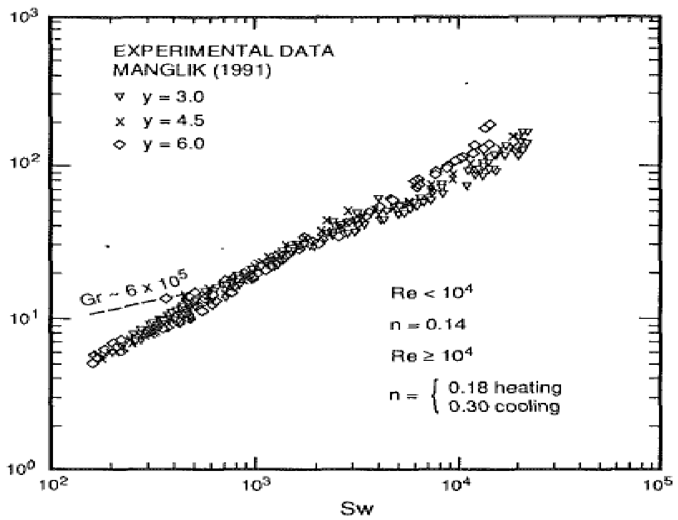

Fig. 4.2 variation of Nu with respect to Reynolds Number predicted by Manglik [1]

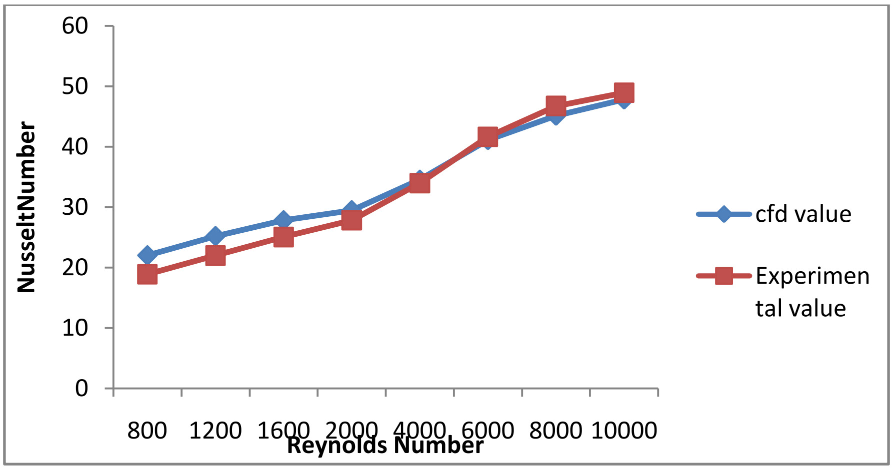

Fig4.3: Validation of CFD Result with Experimental Values.

4.3 Numerical Simulation of Heat transfer, Friction Factor and pumping power Characteristics of Double Tube Heat Exchanger with twisted tape insert.

The model double tube heat exchanger with twisted tape insert is meshed & after setting the boundary conditions with Re ranging from 800 to 10000 is then subjected to steady state analysis in ANSYS Fluent 15 by solving Reynolds Averaged Navier Equation (RANS).

Realizable k-ε turbulence model is used to predict the swirl flow more accurately. For the simulation purpose SIMPLE algorithm is used to solve the pressure-velocity coupling. Second order spatial discretization scheme is used for pressure. Second order upwind scheme for momentum, turbulent kinetic energy and turbulent dissipation rate & energy equation

4.3.1 Effect of different Pitch ratios of the tape on Nusselt Number, Friction factor & Pumping Power

The variation of Nusselt number with Reynolds number for the double tube heat exchanger has been considered with twisted tape insert for different pitch/diameter ratios (P/d= 4, 5, 7). Inner diameter of inner tube has been kept constant to get the following pitch value as 90,110,154 mm respectively. The Nusselt number is found to be increasing with increase in Reynolds number for both tube in tube heat exchanger with and without twisted tape insert. In case of heat exchanger with twisted tape, as the pitch ratios (Pitch/Diameter of inner tube) is decreasing, the Nusselt number is increasing. It is observed from[fig-4.4] that P/d value as 7, 5, 4 of the twisted tape , the rate of heat transfer enhanced upto 37%, 50%, & 59% respectively as compared to tube without twisted tape insert.

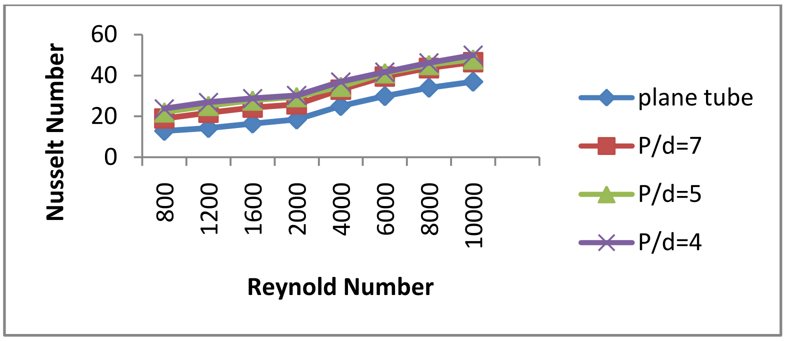

Fig 4.4: Comparison of Nusselt Number for plane tube and tube with twisted insert with different P/d ratio.

4.3.2 The variation of Friction Factor with Reynolds number for the tube in tube heat exchanger with twisted tape of different pitch ratios (P/D= 7,5,4) .Fig [4.5] shows as the Reynolds Number increases, the Friction factor decreases for smooth heat exchanger as well as pipe with twisted tape.In case of heat exchanger with twisted tape inset , as the pitch/diameter ratios (P/ d) increases, the friction factor decreases. The increment in friction factor is around 207%, 302%, and 389% respectively with twisted tape insert in comparison with plain tube.

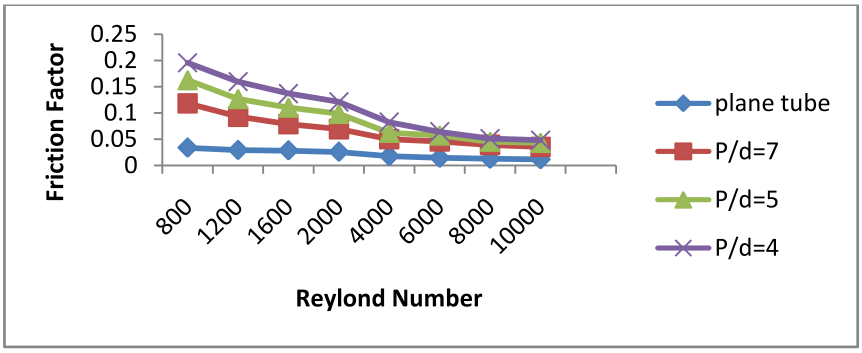

Fig 4.5: Comparison of Friction factor for plane tube and tube with twisted insert with different P/d ratios

4.3.3 The variation of Pumping Power with different Reynolds number for the tube in tube heat exchanger with twisted tape insert for different pitch/diameter ratios (P/D= 4,5,7) has been studied. It is found from fig [4.6] that Pumping power increases with increase in Reynolds number for both with and without twisted tape insert heat exchanger.

In case of heat exchanger with twisted tape insert, as the (Pitch/ inner diameter of inner tube) ratio decreases, the pumping power increases. There is an increment of 207%, 300%and 380% in pumping power with twisted tape in comparison with plain pipe. As we increase the P/d value the pitch increases as d is constant and as pitch increases the obstruction in the flow decreases, therefore power required will be less at the same time Heat transfer rate also decreases.

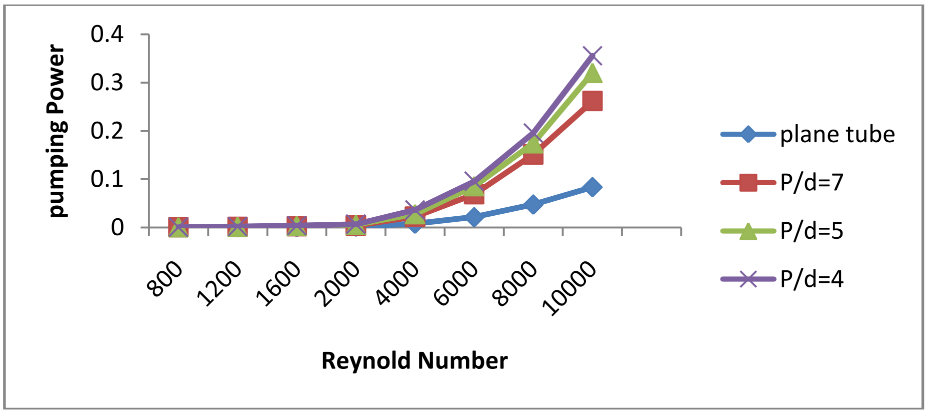

***Fig 4.6:* Comparison of Pumping Power for plane tube and tube with twisted insert with different P/d ratios**

4.4 Optimization of the heat exchanger for different P/d condition.

Improving the heat transfer rate and at the same time reducing the pumping powers are the key factors while designing a heat exchanger. To increase the heat transfer rate the Nusselt number should be high as possible. But with increases in Reynolds number the flow become more turbulent and it increases the pressure drop across the heat exchanger. If more is the pressure loss more will be the pumping power required. With increases in pumping power the cost of the exchanger increase. So we have to make a compromise between increases in heat transfer rate with increase in pumping power requirement.

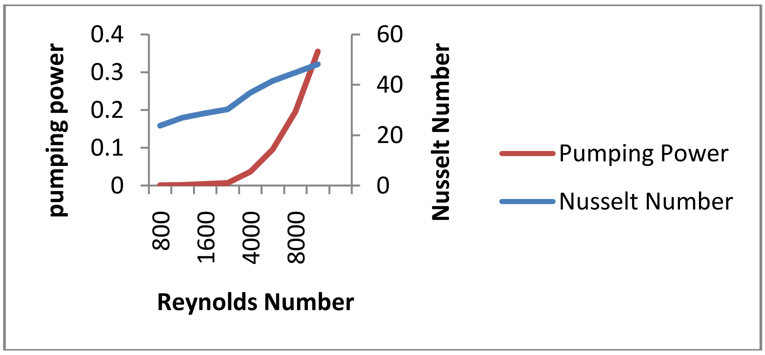

Fig 4.7: Variation of Pumping Power and Nusselt Number with respect to Reynolds Number for P/d = 4

4.4.1 Fig [4.7] shows the variation of Nusselt number and pumping power with Reynolds number for P/d=4. The optimize condition was considered for insulated outer wall condition. The Nusselt number variation is taken in primary Y-axis and pumping power in secondary Y-axis. With increase in Reynolds number Nusselt number increases and pumping power decreases. Both these curve intersect each other at Nusselt number 47.2267, pumping power 0.3158 W corresponding to Reynolds Number as 9516.5756. This intersection point is the optimize condition because with further increase in Re power required increases at a higher rate than that of the Nusselt number. So it is uneconomical to increase the Re beyond this range.

4.4.2 Fig [4.8] shows the variation of Nusselt number and pumping power with Reynolds number for P/d=5. The optimize condition was considered for insulated outer wall condition. The Nusselt number variation is taken in primary Y-axis and friction factor in secondary Y-axis. With increase in Reynolds number Nusselt number increases and pumping power decreases. Both these curve intersect each other at Nusselt number 45.568, pumping power 0.274W corresponding to Reynolds Number as 9387.790.

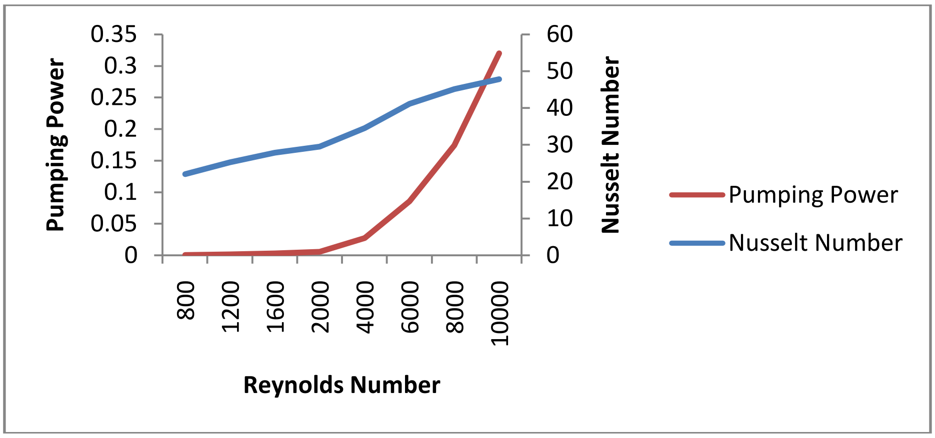

Fig 4.8: Variation of Pumping Power and Nusselt Number with respect to Reynolds Number for P/d = 5

4.4.3 Fig [4.9] shows the variation of Nusselt number and pumping power with Reynolds number for P/d=7. The optimize condition was considered for insulated outer wall condition. With increase in Reynolds number Nusselt number increases and pumping power decreases. Both these curve intersect each other at Nusselt number 42.70, pumping power 0.214W corresponding to Reynolds Number as 9310.07.

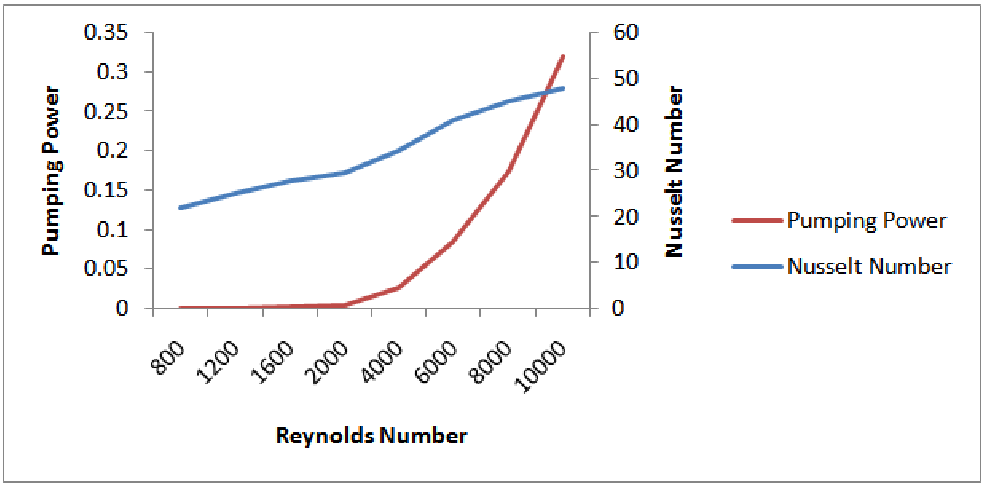

Fig 4.9: Variation of Pumping Power and Nusselt Number with respect to Reynolds Number for P/d = 7

5 Conclusion And Future Scope

Numerical simulation has been carried out for tube in tube heat exchanger with twisted tape insert. Variation of Nusselt number, Darcy friction factor, pumping power required, with respect to Reynolds number for different P/d ratio has been plotted. Heat transfer behaviours for different boundary conditions are predicted and optimize condition for maximum Nusselt number (Nu) and minimum pumping power was plotted against Reynolds number.Following are the outcome of above numerical study:

1. With increase in the Reynolds number, the Nusselt number for tube in tube heat exchanger increases for both with and without twisted tape insert. Increment in Nusselet number for tube with twisted tape is 42% in comparison to tube without insert.

2. With increase in the Reynolds number, the friction factor for tube in tube heat exchanger decreases for both with and without twisted tape insert. Increment in friction factor for tube with twisted tape is 203% in comparison to tube without insert.
3. With increase in the Reynolds number, the pumping power for tube in tube heat exchanger increases for both with and without twisted tape inserts. Increment in pumping power for tube with twisted tape is 214% in comparison to tube without insert
4. The Nusselt number decreases with increase in p/d ratio, for a particular value of Reynolds number. Nusselt number has the maximum value for p/d=4 that is 48.127 corresponding to 10,000 Reynolds number.
5. With increase in p/d ratio the Friction factor decreases, for a particular value of Reynolds number.
6. With increase in p/d ratio the Pumping power decreases, for a particular value of Reynolds number.
7. The optimization point moves towards lower Reynolds Number with increase in p/d ratio.

Future Scope

- In this study single phase flow pattern is considered but in future two or multiphase flow may be considered.

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