Heat Transfer Characteristic Analysis of Double Pipe Heat Exchanger With A Continuous Helical Baffle By Numerical Analysis

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ABSTRACT

In this paper, Three-dimensional computational fluid dynamics (CFD) simulations have been performed to study the annulus side laminar flow distribution, heat transfer characteristic, and pressure drop. This paper shows the effect of Reynolds number (*Re*), helical pitch and baffle height on Nusselt number (*Nu*), heat transfer coefficient, heat transfer rate, effectiveness & pressure drop of a double pipe heat exchanger with or without a continuous helical baffle. There are eight geometry was drawn by the design modeler such as pitch 102 mm, pitch 128mm, pitch 170mm, pitch 210mm, pitch 250mm with the pitch height 5mm & again for the pitch 102mm, spiral height 7mm, height 9mm, height 11mm. The outer diameter of the inner tube & inner diameter of the outer tube is fixed in all cases. The spiral was drawn 250 mm far from the inlet because the flow is fully developed after 250 mm. The inner pipe's outer surface is kept at a constant temperature of 340k. The temperature of the cold fluid is 293K. The outer tube's the outside wall is kept adiabatic. The fully developed laminar flow passes through the annulus side with Reynolds numbers vary 700 to 1500.When the increases the pitch of the helical the Nusselt decreases. On the other hand, the Nusselt number increase while increasing helical height. If the comparison will take place between plain tube and helical tube the 78.53 % Nusselt number increases while using helical baffle for the pitch 102 mm & Reynolds number 1500. While using pitch 102 mm with a height of 11 mm instead of a plain tube the Nusselt number increases by 133.89 % for the Reynolds number 1500.When the increases the pitch of the helical the effectiveness decreases. On the other hand, the effectiveness increase while increasing helical height. When the increases the pitch of the helical the pressure drop decreases. On the other hand, the pressure drop increase while increasing helical height.

Keywords: CFD analysis, Laminar flow, Baffle pitch & Height, Nusselt number, Pressure drop.

1. Introduction

The heat exchanger is a device that transfers heat from a hot fluid to a cold fluid at the fastest possible rate with the least amount of expenditure. A double pipe heat exchanger is a heat exchanger that is made up of two concentric pipes separated by a mechanical closure. This heat exchanger, as its name implies, employs two pipes to transfer heat between two fluids. The heated fluid is in one pipe, while the cool fluid is in the other. This heat exchanger is also known as a jacketed u-tube heat exchanger, jacketed tube heat exchanger, hairpin heat exchanger, and pipe-in-pipe heat exchanger become more popular day by day. But this technology has added more complexity [1]. The helical baffle on the inner tube of a double pipe heat exchanger increases turbulence while increasing surface area available for heat transmission. When the entire length of the heat exchanger is unchanged, the efficiency of a helical baffle with a larger pitch for heat transfer improvement is fairly weak. If the Reynolds number is high, the pressure drop will be significant if the helical pitch is reduced. As a result, at low Reynolds numbers, helical baffles are more ideal for heat transfer augmentation. Researchers from all across the world are collaborating to find a suitable answer [2] .Shewale omkar M et al. [3] investigated a double pipe heat exchanger with helical fins on the inner rotating tube experimentally. The Nusselt Number derived from the experimental results is higher than the theoretical numbers derived from the Dittus-Boelter equation in this study. The addition of helical fins to the inner

tube increases the heat transfer surface while reducing the hydraulic diameter of the flow channel. For stationary conditions, the Nusselt number for the inner tube with helical fins is 4 times higher than that of the plain inner tube. The Nusselt number is 36 percent higher at 50 rpm and 64 percent higher at 100 rpm than a motionless inner tube.The heat transmission and pressure drop of tubes with internal wave-like longitudinal fins were computed by Yu et al [4]. For this study, they employed air as a working fluid and conducted two cases. Because the inner channel of the insertion is not blocked, the flow cross-section area of the tube of type A differs only slightly from that of the tube without the insertion. The inner tube of type B, on the other hand, has a completely blocked cross-section. The annulus has wave-like fins that span its whole diameter. There are a total of 20 waves in this game. The outside tube was heated by electricity. In 13 cross-sections along the test tube axis, pressure taps were not evenly distributed.M Sheikholeslami et al [5], conducted both experimental and numerical evaluations of a conical ring in the annulus side of an exchanger. They investigated how various design features such as conical angle, pitch ratios, and open area affect performance characteristics. Experimentally, the heat transfer characteristics of helical fins on the annulus part of an exchanger were examined [6]. It was discovered that the addition of helical fins had no effect on the performance of the exchanger, and that the increased rate of heat transfer is precisely proportional to their fluid flow rates. Their

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performance in an exchanger was investigated experimentally using a variety of inner pipe materials [7] as well as varied fin pitches. The use of copper as an inner tube improves the heat exchanger's performance because of its high thermal conductivity; an exchanger with fins has a higher heat transfer rate than one without fins, and flow rate increases as the heat transfer coefficient rises.

2. Numerical Model (Finite volume method)

2.1 Physical Model

The geometrical model of the computational fluid domain is developed using a design modeler in ANSYS FLUENT. There are eight geometry was drawn by the design modeler such as pitch 102 mm, pitch 128mm, pitch 170mm, pitch 210mm, pitch 250mm, the pitch height 5mm & again for the pitch 102mm, spiral height 7mm, height 9mm, height 11mm.

Table 1 Dimensions of geometry

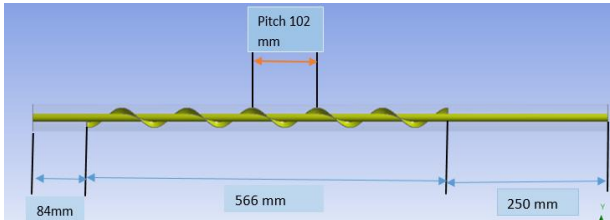| Length of pipe, L	900 mm
Inner diameter of inner tube,	10 mm
Outer diameter of inner tube,	12 mm
Inner diameter of outer tube,	40 mm
Outer diameter of outer tube,	42 mm
Spiral length	566 mm
Spiral thickness	1 mm
Spiral height	5,7,9,11 mm
Spiral pitch	102,128,170,210,250 mm
Material of pipe	steel

Fig 1 Computational domain and Boundaries

2.2 Boundary Conditions

Cold water inlet temperature is kept 293k. Cold water outlet condition is assigned at pressure outlet. The outer wall of inner tube's temperature (340k) is kept constant. No slip condition is assigned to all surfaces such as spiral, all tube wall. The outer tube wall is perfectly insulated. So, heat flux is set zero. The flow is Steady state. Uniform density. So, ρ =constant.

2.2 Grid Generation and near wall treatment

For this heat exchanger simulation, unstructured tetrahedral meshing is performed and then an inflation layer is added within the inner tube wall and spiral. In the inflation layer, the maximum layer is fixed in 7 layers with a growth rate of 1.5 & first-layer thickness of .00001mm. A mesh had been generated with .008 mm global element size & found 1690361 nodes & number of element 7183209.
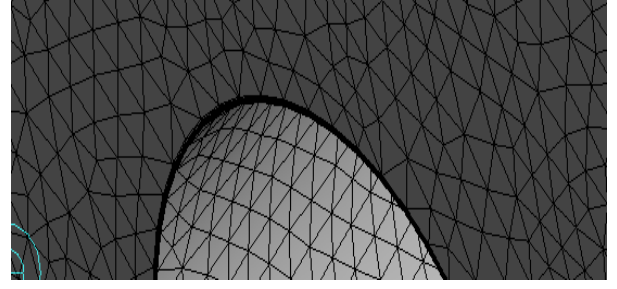

Fig.2.2 Unstructured computational mesh

2.3 Governing Equation:

The fluid flow and heat transfer is considered steady and laminar. The governing equations for mass, momentum and energy conservation are used. These equation are given below [8]:

2.3.1 Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial(r\rho u_r)}{\partial r} + \frac{1}{r}\frac{\partial(\rho u_\theta)}{\partial \theta} + \frac{\partial(\rho u_z)}{\partial z} = 0$$

Since the problem is assumed to be steady state & incompressible, so time dependent parameter ($\frac{\partial \rho}{\partial t}$) & ρ dropped from the equation & get,

$$\frac{\partial(r u_r)}{\partial r} + \frac{1}{r}\frac{\partial(u_\theta)}{\partial \theta} + \frac{\partial(u_z)}{\partial z} = 0$$

2.3.2 Momentum equation (Navier–stokes equation):-

In cylindrical co-ordinates, (r, θ,z) the Navier-stokes equation of motion for an incompressible fluid of constant dynamic viscosity μ, and density ρ are :

$$\rho\left[\frac{Du_r}{Dt} - \frac{u_\theta^2}{r}\right] = -\frac{\partial p}{\partial r} + f_r + \mu\left[\nabla^2 u_r - \frac{u_r}{r^2} - \frac{2}{r^2}\frac{\partial u_\theta}{\partial \theta}\right]$$

$$\rho\left[\frac{Du_\theta}{Dt} + \frac{u_\theta u_r}{r}\right] = -\frac{1}{r}\frac{\partial p}{\partial \theta} + f_\theta + \mu\left[\nabla^2 u_\theta - \frac{u_\theta}{r^2} + \frac{2}{r^2}\frac{\partial u_r}{\partial \theta}\right]$$

$$\rho\frac{Du_z}{Dt} = -\frac{\partial p}{\partial z} + f_z + \mu \nabla^2 u_z$$

Where u_r, u_θ, u_z are the velocities in the r, θ , z cylindrical co-ordinate directions, p is the pressure, f_r, f_θ , f_z are the body force components in the r , θ , z directions and the operators $\frac{D}{Dt}$ & ∇^2 are

$$\frac{D}{Dt} = \frac{\partial}{\partial t} + u_r\frac{\partial}{\partial r} + \frac{u_\theta}{r}\frac{\partial}{\partial \theta} + u_z\frac{\partial}{\partial z}$$
$$\nabla^2 = \frac{\partial^2}{\partial r^2} + \frac{1}{r}\frac{\partial}{\partial r} + \frac{1}{r^2}\frac{\partial^2}{\partial \theta^2} + \frac{\partial^2}{\partial z^2}$$

2.3.3 Energy equation:-

$$\rho c_p \left(\frac{\partial T}{\partial x} + v_r\frac{\partial T}{\partial r} + v_\theta\frac{\partial T}{\partial \theta} + v_z\frac{\partial T}{\partial z}\right) = k\left(\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right) + \frac{1}{r^2}\frac{\partial^2 t}{\partial \theta^2} + \frac{\partial^2 t}{\partial z^2}$$

2.4 Data Reduction:

The required equations are given below [1]-
Heat transfer rate of the annulus side fluid (cold water):
$$Q = m_c \, cp_c \, (t_{c,outlet} - t_{c,inlet})$$

Where, m_c is the mass flow rate of cold water, cp_c is the specific heat of the cold water, $t_{c,inlet}$ is the cold fluid inlet temperature, $t_{c,outlet}$ is the cold fluid outlet temperature.

$$Q = U_o A_o \; LMTD$$

Where U_o is the overall heat transfer co-efficient, A_o is the outside surface area of inner pipe, LMTD is the log mean temperature difference

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

Where, $\Delta T_1 = t_{h,inlet} - t_{c,inlet}$ $\Delta T_2 = t_{h,outlet} - t_{c,outlet}$

Hydraulic diameter:

$$D_h = \frac{4A_c}{p} = \frac{4\pi(D_i^2 - D_o^2/4)}{\pi(D_i + D_o)} = D_i - D_o$$

Where, D_i is the inner diameter of outer tube & D_o is the outer diameter of inner tube.

For calculating Nusselt Number, following equation is used:

$$Nu = \frac{hl}{k}$$

Where, l=characteristic length, which is varied for different inlet side velocity and h & k are heat transfer co-efficient and thermal conductivity of the fluid respectively.

Reynolds number is calculated from:

$$Re = \frac{4m}{\pi(D_i + D_o)\mu}$$

Where, m is mass flow rate, μ is the dynamic viscosity which is fixed for a fluid at specific temperature & velocity, D_i is the inner diameter of outer tube & D_o is the outer diameter of inner tube.

2.5 Model validation:

Obtained simulation results are compared with the numerical results of N Sreenivasalu Reddy, K Rajagopal & P H Veena [6] to validate the results .They performed a numerical simulation of double pipe heat exchanger for parallel flow.
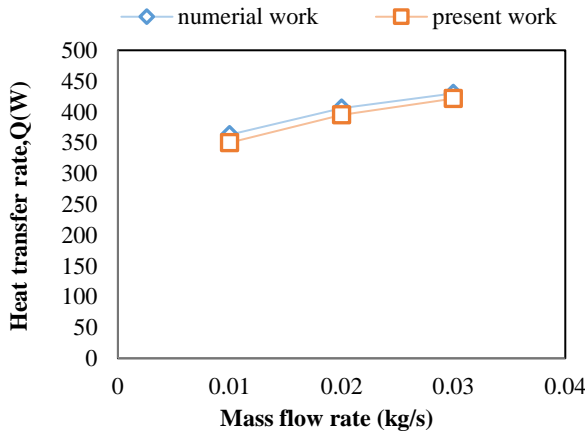

Fig 2 Heat transfer rate variation for different mass flow rate.

Again, the Nusselt number for a simple circular tube in a fully developed laminar flow (thermally and hydrodynamically), constant heat flux condition is 4.36 for any Reynolds number [1]. To validate the results provided below, the obtained simulation results are compared to the theoretical results:

Table 2 Nusselt number Validation comparison.

Case	Reynolds number	Theoretical value, Nu	Numerical value, Nu	Error
1	100	4.36	4.38	0.46%

2.6 Mesh dependency Test:

Mesh dependency for the CFD analysis was done for the different element sizes. For the pitch 170mm & height 5mm, for the different element size such as (.0012 ,0.0008mm , 0.0007mm , 0.0006mm) at 1500 Reynolds number .If we compare meshing 2 & meshing 3, the error is found 0.21%. So I working on meshing 2 element size .008 mm.

Table 3 Mesh dependency

Element size, mm	Nodes	Mass flow rate Kg/m	Cold fluid outlet temperature ($^\circ k$)	Heat transfer rate ,Q (W)	Error %
.0012	7,04895	.0478	297.81	960.83	
.0008	1,69,036	.0478	297.79	956.83	0.42
.0007	22,84205	.0478	297.78	954.83	0.21
.0006	32,27316	.0478	297.77	952.84	0.208

Table 4 Properties of water

Density, ρ (kg/m³)	996
Specific heat cp_c, (J/kg K)	4179
Thermal conductivity, k (W/m ·K)	0.61
Dynamic viscosity, μ (kg m⁻¹ s⁻¹)	0.00078

3.0 Result & Discusion:

There is five plane view such as the inlet, at 240 mm, at 500 mm, at 816 mm, at the outlet, and two sectional views for the pitch of 102 mm with height of 11 mm & Reynolds number 700 are shown in figure 4. When the position is changed horizontally, the colors in the legend change to indicate the temperature, regardless of whether it is the highest or lowest. The figure shows that the minimum temperature & maximum temperature range found 293 k to 349 k respectively.
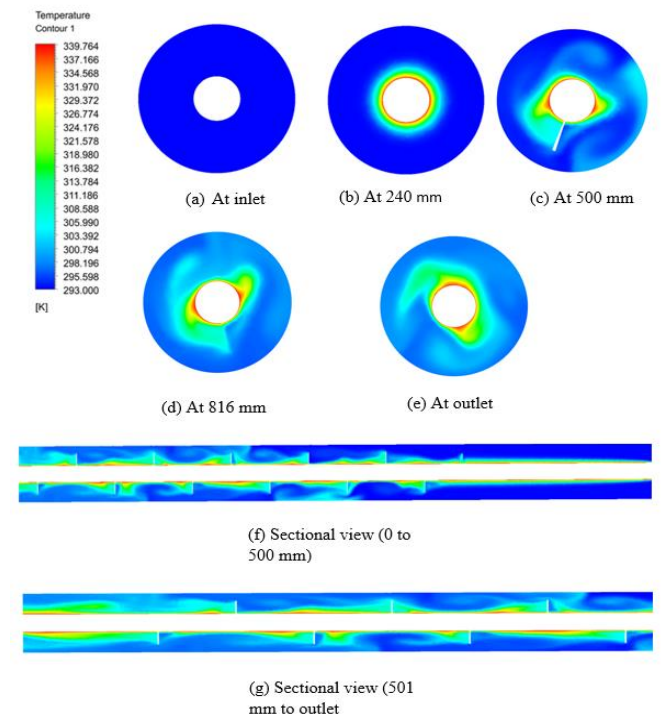

Fig 3 Temperature contours for different positions in annulus side for pitch 102 mm with height 9 mm.
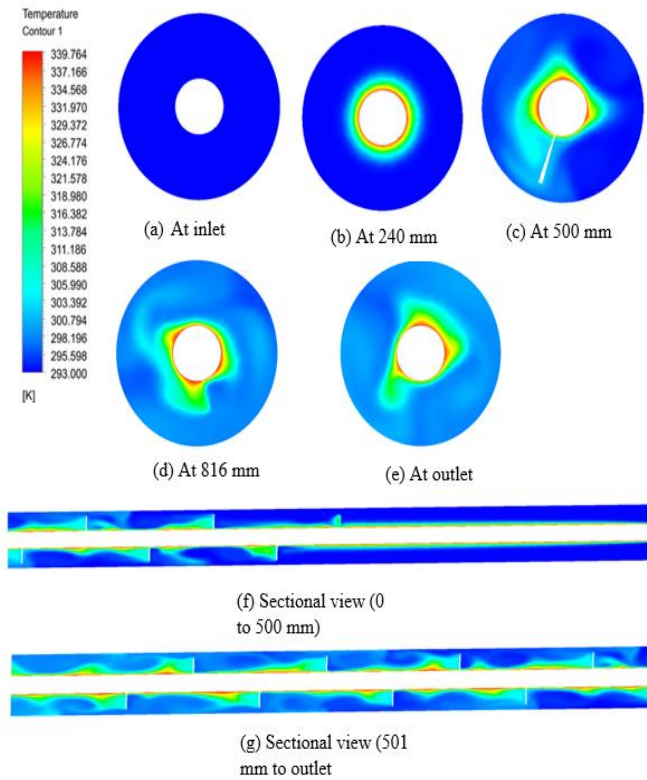
Fig 4 Temperature contours for different positions in annulus side for pitch 102 mm with height 11 mm

The minimum velocity was found at the inlet. When the flow is moving forward the flow velocity decreases on the hand temperature increases the cold water because of absorbing heat from the heating surface. While the fluid enters the baffle, the baffle creates an obstruction that occurs proper mixing of fluid & gets more contact area as a result heat transfer rate enhanced. As can be seen from fig 4 (c) around the baffle the legend color changes because of absorbing heat from heating surface.

While the fluid leaves the baffle the average temperature increases & legend color changes are going to be noticed. When the fluid reaches the outlet the area-weighted average temperature is found 302.8134 k. As can be seen from the sectional view fig 4 (f & g) while passing through the free space between the baffle & tube wall the temperature increases around the baffle wall by absorbing heat.
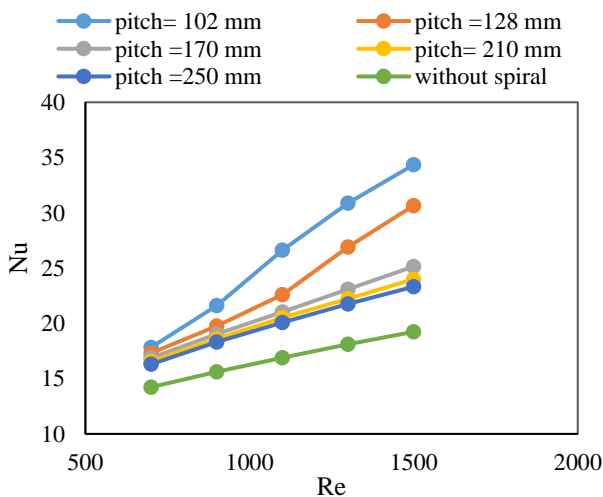

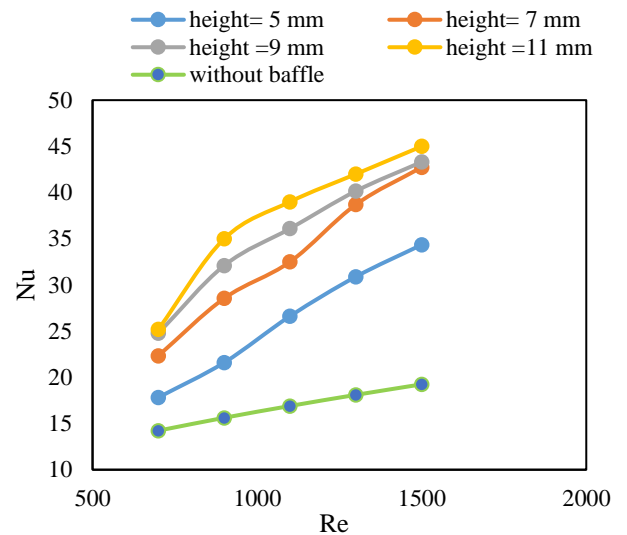**Fig 5** Nusselt number variation for different Re for the baffle height 5mm & different pitch.

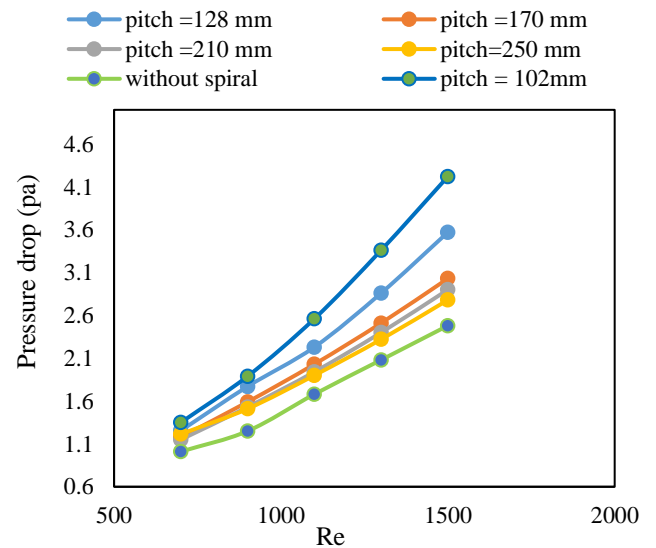**Fig 6** Nusselt number variation for different Re for pitch 102 mm & different height.

Fig 7 Pressure drop variation for different Re for height 5 mm & different pitches.
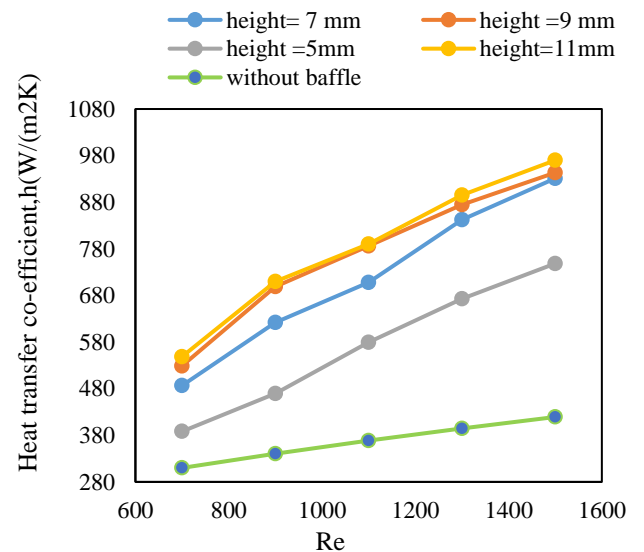

Fig 8 Heat transfer co-efficient variation for different Re for pitch 102 mm & different height.
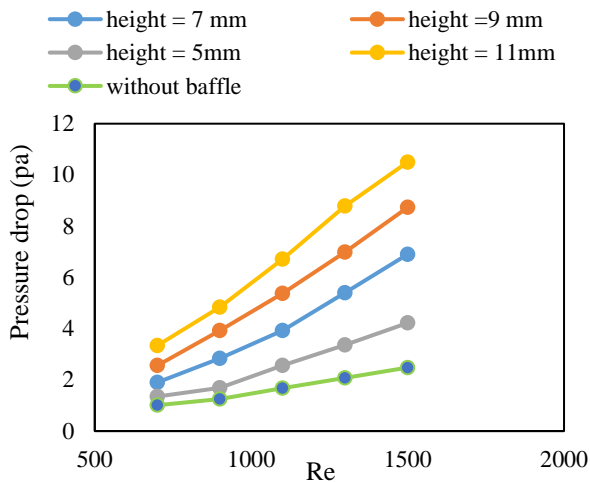
Fig 9 Pressure drop variation for different Re for height 5 mm & different pitches.
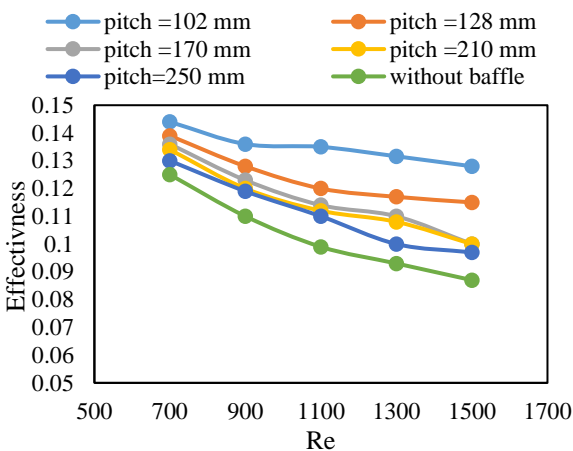

Fig 10 Effectiveness variation for different Re for pitch 102 mm & different height.
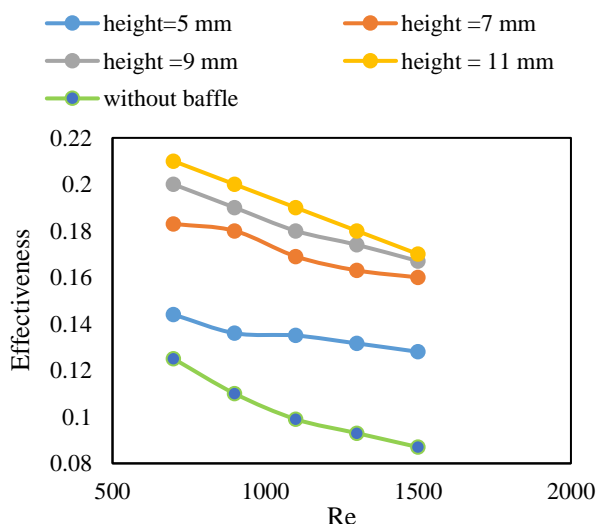

Fig 11 Effectiveness variation for different Re for pitch 102 mm & different height

4. Conclusion:

In the present study, computational fluid dynamics method is employed to systematically study the heat transfer characteristic in annulus side of double pipe heat exchanger with helical baffle. The major finding are summarized as follow –

When the increases the pitch of the helical baffle the Nusselt decreases. On the other hand, the Nusselt number increase while increasing helical height of baffle. If the comparison will take place between plain tube and helical tube the 78.53 % Nusselt number increases while using helical baffle for the pitch 102 mm & Reynolds number 1500. While using pitch 102 mm with a height of 11 mm instead of a plain tube the Nusselt number increases by 133.89 % for the Reynolds number 1500.When the increases the pitch of the helical baffle the effectiveness decreases. On the other hand, the effectiveness increase while increasing helical height of the baffle.

The pressure drop increase while increasing helical height of the baffle. The shape of the helical baffle plays an important role in the annulus side heat transfer and fluid flow performance. To enhance heat transfer performance of the heat exchanger by selecting a better helical pitch & height. So, while using a pitch of 102 mm with a height of 11 mm instead of a height of 9 mm the heat transfer rate, heat transfer coefficient, Nusselt number, effectiveness slightly increases by 2.55 %, 2.8%, 3.9 %, 1.5% respectively. On the other hand, pressure drop rapidly increases by 20 %.

Pressure drop is an important parameter in the design of the heat exchanger because the pressure drop increases the pumping cost will also increase as a result operating costs, the maintenance cost will increases. As a result, the overall performance of the helical baffle with a pitch of 102 mm with a height of 9 mm is the best.

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NOMENCLATURE
Nu : Nusselt Number
Re : Reynolds Number