CFD ANALYSIS OF HEAT TRANSFER THROUGH A TUBE IN TUBE HELICAL COIL HEAT EXCHANGER

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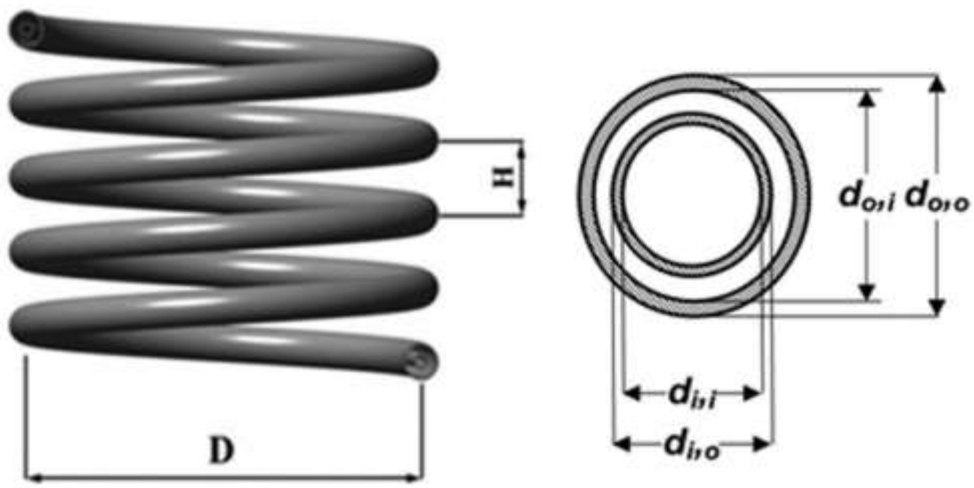
Abstract : Working towards the goal of saving energies and to make compact design for mechanical , chemical devices and plants. The enhancement of heat transfer is one of the key factors in design of heat exchangers. In this process without application of external power can enhance the heat transfer rate by modifying the design by providing the helical tubes, extended surface or swirl flow devices. Helical tube heat exchanger provides certain advantage such as compact structure, larger heat transfer surface area and improved heat transfer capability. This thesis shows the deviation of Nusselt Number and friction factor for different curvature ratio(D/d ratio) and Reynolds Number. CFD analysis has been done for varying inlet conditions keeping the heat flux of the outer wall constant. The turbulent flow model with counter flow heat exchanger is considered for analysis purpose. Copper was used as the base metal for both inner and outer pipe. Water and nano fluid(Al_2O_3-H_2O) was taken as the working fluid for both inner and outer tube. The results show that Nusselt Number and friction factor depend on curvature ratio. From the results complicated behavior of fluid flow is captured for both the fluids flowing inside the tube. With increase in D/d ratio (inverse of curvature ratio) the Darcy friction factor decreases with increase in Reynolds number. The value of Nusselt number was found to increase with increase in mass flow rate.

IndexTerms **- Helical tube heat exchanger, Nusselt Number, Reynolds Number, friction factor, curvature ratio(D/d ratio), nano fluid(Al_2O_3-H_2O)**

1.INTRODUCTION

The objective behind constructing a heat exchanger is to get an effective method of heat exchange starting with one fluid then onto the next, by direct or indirect contact. Heat transfer occurs in three ways: conduction, convection and radiation. In helical coil heat exchangers the heat transfer by radiation is not taken into account as it is much less and negligible as compared to conduction and convection. Conductive heat transfer can be experienced with a minimum thickness if wall of highly conductive material. in the performance of a heat exchanger, convection plays a major role. In natural convection, heat transfer occurs by density difference in a fluid due to temperature gradient, and hence doesn't require any external source like pump, fan or a suction device. Fluid surrounding a heat source receives heat, becomes less dense and rises up. The fluid that is surrounding the hot fluid is cooler and then moves in to replace it. In forced convection, a heat exchanger exchanges the heat from one moving stream to an alternate stream through the pipe wall. The cooler fluid absorbs heat from the hooter one as the flow is counter flow. If the cold fluid moves along the direction of hot fluid it is called parallel flow and if moves opposite to it, we call that counter flow. Heat transfer coefficient: In convective heat transfer heat transfers from one part to another by the movement of fluid particles due to the density difference across a thin film of the surrounding fluid over the hot surface. Through this film heat exchange happens by thermal conduction and as thermal conductivity of most fluids is low, the resistance lies there. The heat transfer by convection is governed by the equation, Q=hA (tw-tatm), Where, h= film/surface coefficient (W/m2-K) A= area of the wall Tw = wall temperature Tatm = surrounding temperature..The value of heat transfer c-efficient depends upon the different properties of fluid within film region.. The overall heat transfer coefficient can be defined as the overall transfer rate of a combination of series and parallel conductive/convective walls. The overall Heat Transfer Coefficient' is expressed in terms of thermal resistances of each fluid stream. The summation of individual resistances is the total thermal resistance and its inverse is the overall heat transfer coefficient.



Schematic diagram of double helical tube heat exchanger

2. Nano fluid

A Nanofluid is a fluid containing nanometer-sized particles, called nanoparticles. These fluids are engineered colloidal suspensions of nanoparticles in a base fluid. The nanoparticles used in nanofluids are typically made of metals, oxides, carbides, or carbon nanotubes. Common base fluids include water, ethylene glycol and oil.

Nanofluids have novel properties that make them potentially useful in many applications in heat transfer, including microelectronics, fuel cells, pharmaceutical processes, and hybrid-powered engines, engine cooling/vehicle thermal management, domestic refrigerator, chiller, heat exchanger, in grinding, machining and in boiler flue gas temperature reduction. They exhibit enhanced thermal conductivity and the convective heat transfer coefficient compared to the base fluid. Knowledge of the rheological behavior of nanofluids is found to be very critical in deciding their suitability for convective heat transfer applications.

In analysis such as computational fluid dynamics (CFD), nanofluids can be assumed to be single phase fluids. However, almost all of new academic paper uses two- phase assumption. Classical theory of single phase fluids can be applied, where physical properties of nanofluid are taken as a function of properties of both constituents and their concentrations.

3.THERMO PHYSICAL PROPERTIES OF NANO FLUID

In the absence of experimental data for nanofluid densities, constant value temperature independent values, based on nanoparticle volume fraction, the following parameters were calculated.

equation for density,

$$\rho_{nf} = 1 - \phi\, \rho_f + \phi \rho_p \, s$$

where

ρ_f=density of base fluid, kg/m^3

ρ_p = density of particle ,kg/m^3

ρ_{nf} = density of nano fluid, kg/m^3

ϕ = volume fraction

Thermal conductivity of nano fluid by Maxwell model:

$$k_{nf} = k_f \frac{k_p + 2k_f - 2\varphi_v (k_f - k_p)}{k_p + 2k_f + \varphi_v (k_f - k_p)}$$

where

k_{nf} = thermal conductivity of Nano fluid, W/m-K
k_{bf} = thermal conductivity of base fluid, W/m-K

k_p = thermal conductivity of particle, W/m-K

Particle	Mean diameter(nm)	Density (kg/m^3)	Thermal Conductivity(W/mK)	ecific Heat (J/kgK)
Al_2O_3	45	3970	46	880

Thermo Physical Properties of Nano fluid

4.DESIGN OF HEAT EXCHANGER

The double tube helical coil heat exchanger or tube in tube helical coil heat exchanger with two numbers of turns. For simplification in numerical analysis consider only two turns but in practical problems it may be large number of turns depending on the requirements. The coil diameter (D) was varying from 80mm to 240mm in an interval of 40mm that is120mm,160mm,200mm respectively. As the coil diameter increases the length of the exchanger (L) also increases. The inner tube diameter (d1) was 8mm.the thickness (t) of the tube was taken 0.5mm.The outer tube diameter (d2) was taken 17mm.In my study I fixed the tube diameter (both inner and outer diameter) of the heat exchanger and vary the coil diameter of the tube to see the effect of curvature ratio (d/D) on heat transfer characteristics of a helical coil heat exchanger. The pitch of the coil was taken 30mm that is the total height of the tube was 60mm. The heat exchanger was made of COPPER. The fluid property was assumed to be constant for analysis. In this study considered the counter flow heat exchanger as it has better heat transfer rate compared to parallel heat exchanger. The cold fluid and the hot fluid flow in opposite directions in their respective tube. In this study for analysis, turbulent fluid flow was considered. Both the hot fluid and cold fluid flow with a velocity, for which Reynolds number is greater than critical Reynolds number as per the correlation calculated by Schmidt,(1967).The flow velocity of cold fluid is remained constant and the hot fluid flow rate varied to find the Nusselt number , friction factor , obtaining the temperature, pressure and the velocity contours and plot the graphs between nusselt number and Reynolds number and friction factor and Reynolds number. The analysis is performed using water and water based nano fluid concentration for different mass flow rates

5.Defining material properties

Water based Nano fluid used as the fluid flowing through (or) piping. Its material properties are calculated by using formulas and copied in the FLUENT Software. The tube in tube properties are defined in FLUENT using its material browser. For the different flow arrangements problem certain properties were defined by the user to prior to computing the model, these properties were thermal conductivity, density, heat capacity at constant pressure and dynamic viscosity.

Material properties	nsity (ρ) kg/m^3	conductivity(K) W/mk	ific heat C_P j/kgK
Copper	8978	387.6	381

Material Properties of tube

Description	value	units
Viscosity	0.001003	kg/m-s
Density	1000	kg/m^3
Specific heat capacity	4186	J/Kg-K
Thermal conductivity	0.7	W/m-K

Properties of water

6. Methodology and Approach

Finite Volume Analysis Modeling: The mass, momentum, and scalar transport equations are integrated over all the fluid elements in a computational domain using CFD. The finite volume method is a particular finite differencing numerical technique, and is the most common method for calculating flow in CFD codes. This section describes the basic procedures involved in finite volume calculations.

The finite volume method involves first creating a system of algebraic equations through the process of discretising the governing equations for mass, momentum, and scalar transport. To account for flow fluctuations due to turbulence in this project, the RANS equations are discretised instead when the cases are run using the k-epsilon turbulence model. When the equations have been discretised using the appropriate differencing scheme.

6.1Defining Variable Temperature and Velocity

In the FLUENT computer application, temperature, velocity, and various fluid parameters are easily defined and changed by the left-hand tab. non-isothermal flow was used to define the fluid flow parameters and temperature distribution, but in the later models, heat transfer equations were added. This allowed for laminar flow parameters as well as heat transfer equations to be added. For the fluid flowing both an inlet and outlet point was chosen. Under these the velocity field incoming is defined as well as if there is any viscous stress at the outlet. Now that the velocity is defined, the heat transfer in solids is added when heat transfer is used for models with pipe walls, or heat transfer in non- isothermal flow is used. Under this tab (right clicking on the flow tab) these are many applications that can be defined from heat flux, heat conduction, cooling, insulation, to temperature definition and outflow. For the purposes of the models in this paper, temperature is defined in this method both for incoming fluid as well as the constant wall temperature as defined in the beginning models. The point at which the fluid outflows is also defined for the heat transfer. Now that temperature and velocity of the fluid and/ or tubing or pipe wall is defined, the models can be meshed and solved. The parameters are easily changed and much iteration with various values can be performed.

After creating the geometry on CREO PARAMETRIC 4.0 and doing the meshing in ANSYS 18 the problem was analyzed in ANSYS 18 (FLUENT) for different boundary conditions as specified later. For analysis of the problem turbulent fluid flow condition was considered.

Inlet condition	inlet velocity varies (1 to 1.8m/sec)
Outlet condition	zero pascal

Inlet temperature	283K

Boundary conditions for inner fluid

The outer wall of heat exchanger has taken constant heat flux of 5000 W/m ,
that is at d_4=18mm to computing the model, these properties were thermal conductivity, density, heat capacity at constant pressure and dynamic viscosity.

Material properties	**sity (ρ) kg/m^3**	**nductivity(K) W/mk**	**ic heat C_P j/kgK**
Copper	8978	387.6	381

Material Properties of tube

Description	**value**	**units**
Viscosity	0.001003	kg/m-s
Density	1000	kg/m^3
Specific heat capacity	4186	J/Kg-K
Thermal conductivity	0.7	W/m-K

Properties of water

7. MODELLING & CFD AND SETUP

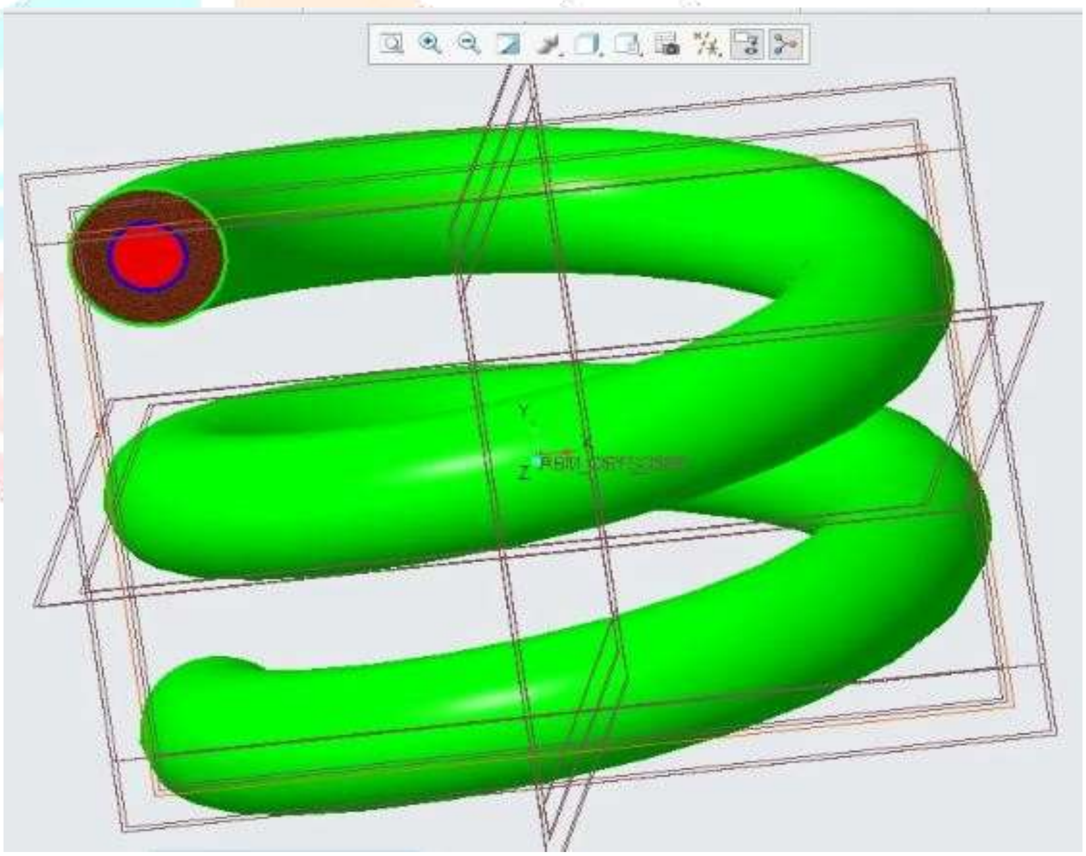
The feature-based parametric modeling technique enables the designer to incorporate the original design intent into the construction of the model. The word parametric means the geometric definitions of the design, such as dimensions, can be varied at any time in the design process. Parametric modeling is accomplished by identifying and creating the key features of the design with the aid of computer software. The design variables, described in the sketches and features, can be used to quickly modify/update the design.

The approach of creating three-dimensional features using two-dimensional sketches is an effective way to construct solid models. Many designs are in fact the same shape in one direction. Computer input and output devices we use today are largely two-dimensional in nature, which makes this modeling technique quite practical. This method also conforms to the design process that helps the designer with conceptual design along with the capability to capture the design intent. Most engineers and designers can relate to the experience of making rough sketches on restaurant napkins to convey conceptual design ideas. Note that Creo Parametric provides many powerful modeling and design tools, and there are many different approaches to accomplish modeling tasks. The basic principle of feature-based modeling is to build models by adding simple features one at a time. In this chapter, a very simple solid model with extruded features is used to introduce the general feature-based parametric modeling procedure.

Parameters	**Dimensions**
Diameter of outer tube	17 mm
Diameter of inner tube	8 mm

Pitch of the coil	30 mm
Thickness of the tube	0.5 mm
Number of coil turns	2
Coil diameter	80-240
Heat exchanger wall material	Copper
Fluid	$_2O_3$-H_20nano fluid

Modelling Dimensions



final assembly with different appearance

8.RESULTS AND DISCUSSIONS

Outer tube inner fluid (Al_2O_3-H_2O nanofluid) properties :

Volume fraction	Thermal conductivity (W/m-K)	Density (Kg/m^3)	Dynamic viscosity (N-s/m^2)	Specific heat (J/Kg-K)
0.4%	0.625	2125	0.00083	1663
0.8%	0.635	3006	0.00085	1014

Nano fluid properties are as follows

Boundary conditions

Inner tube fluid (hot fluid): water

Outer tube fluid (cold fluid):

1.water

2.Al_2O_3-water nanofluid

Inlet temperature of inner tube=348 k

Inlet temperature of outer tube=283k

CFD calculations for Nusselt number:

$$Q_{nf} = m_{nf}\, c_{pf}\, (T_{out} - T_{in})_f$$

$$Q_w = m_w\, c_{p,w}\, (T_{in} - T_{outn})_w$$

$$h_{nf} = \frac{Q}{A_i\,(T_{wall} - T_{bulk})}$$

$$Nu_{i,nf} = \frac{h_{nf}\ d_i}{k_{nf}}$$

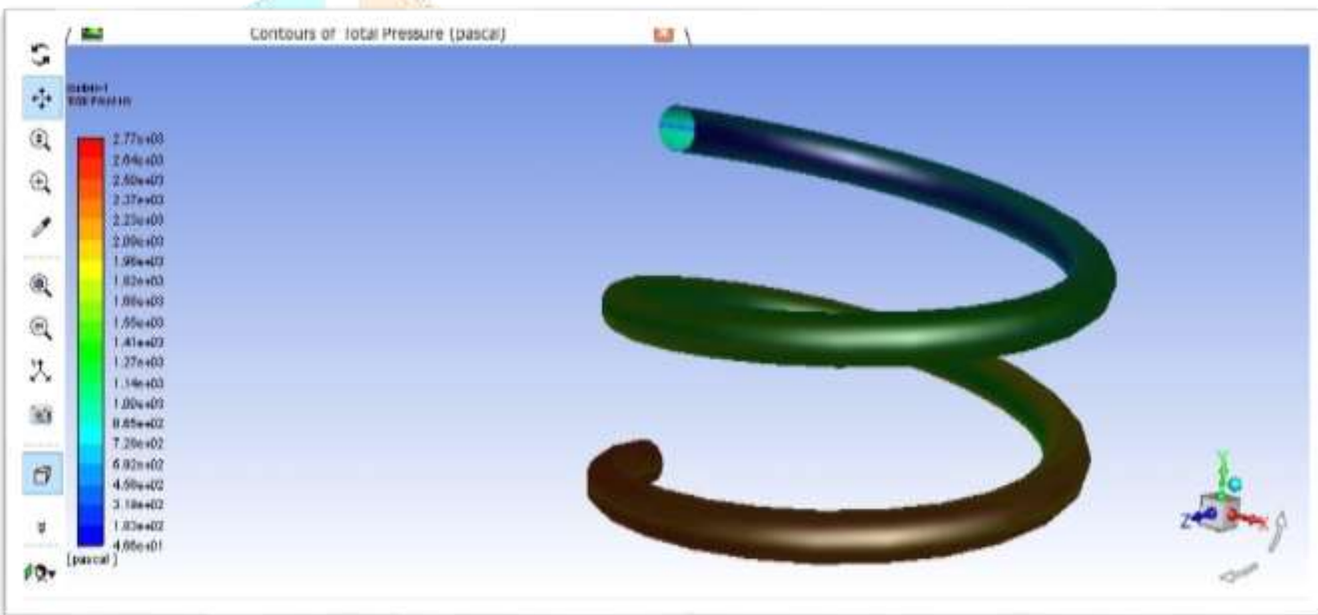
Theoretical calculations for Nusselt number

$$Nu_i = 0.085 Re^{0.74} Pr^{0.4} \delta^{0.1}$$

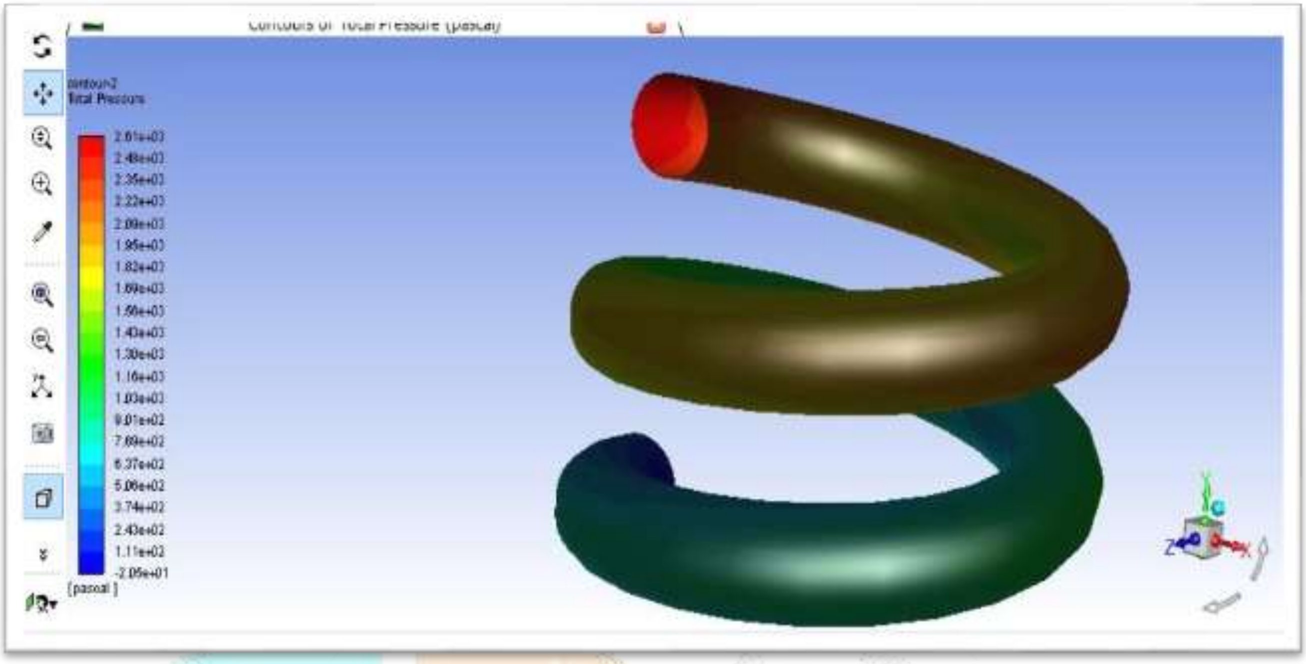
$$De = Re \left(\frac{d_i}{2R_o}\right)^{0.5}$$

$$Nu = \frac{h_{nf}\, d_i}{k_{nf}}$$

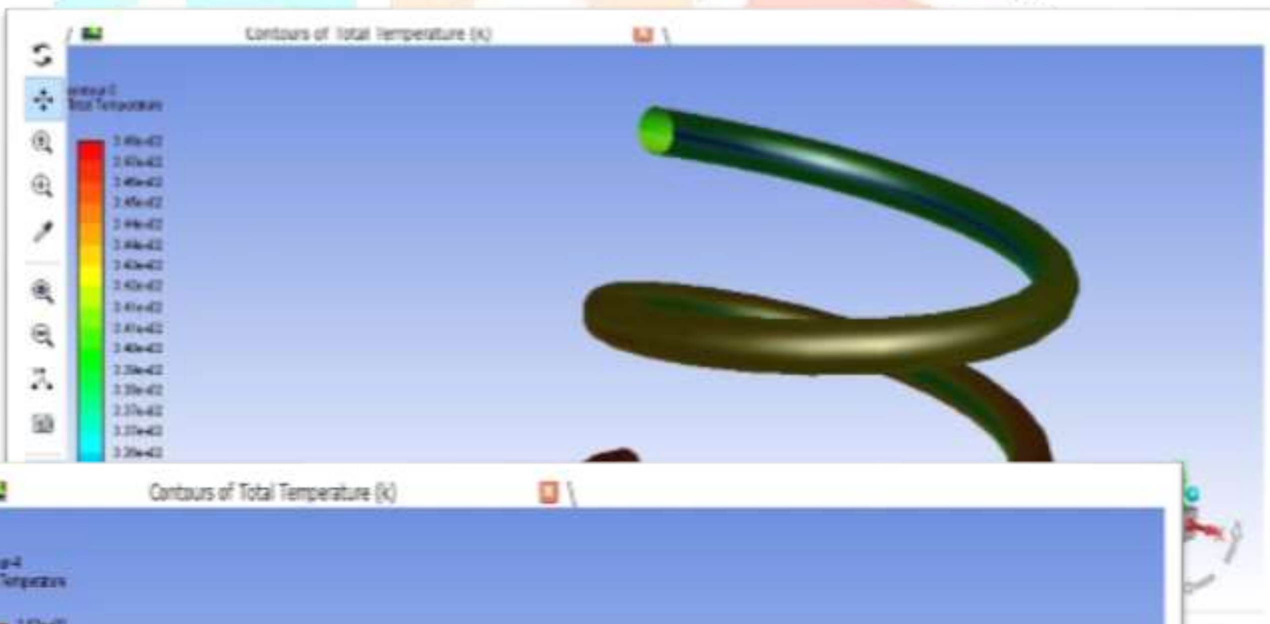
Results of simulation

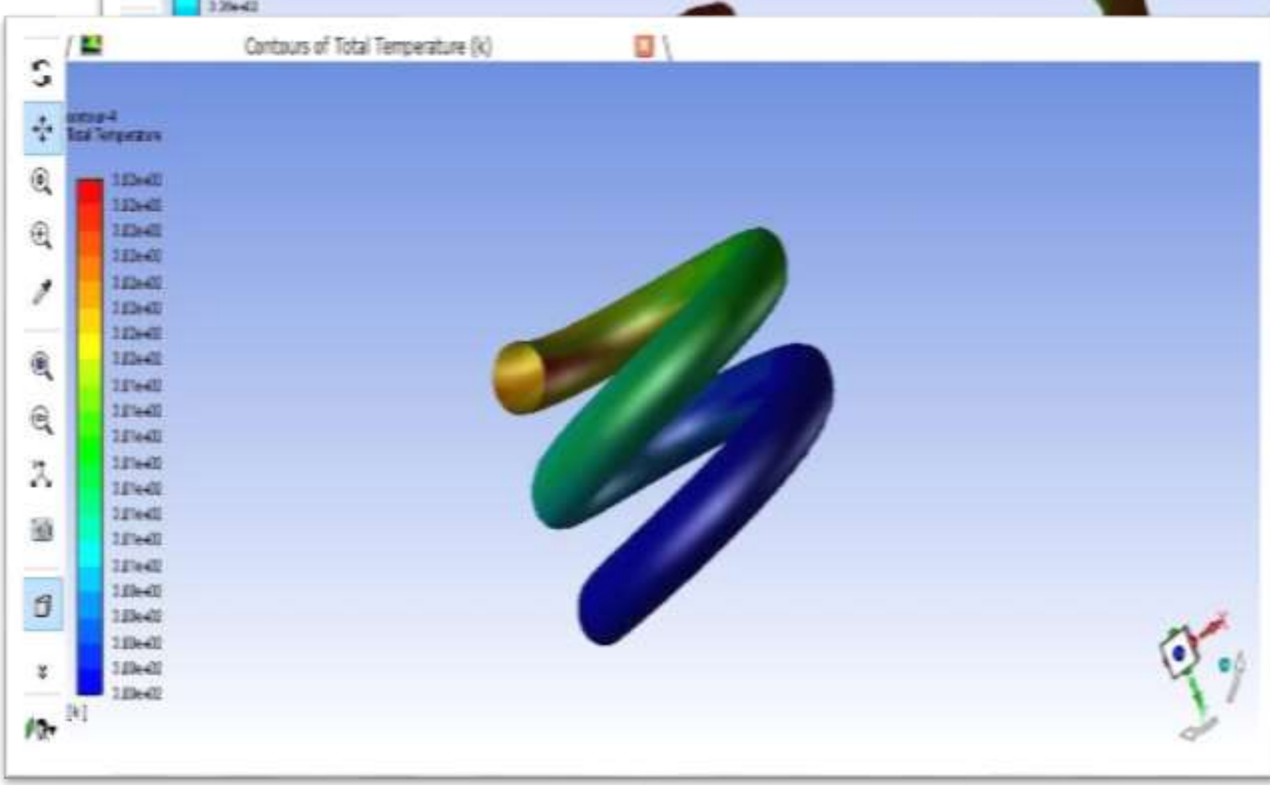


pressure variation in inner tube



pressure variation in outer tube





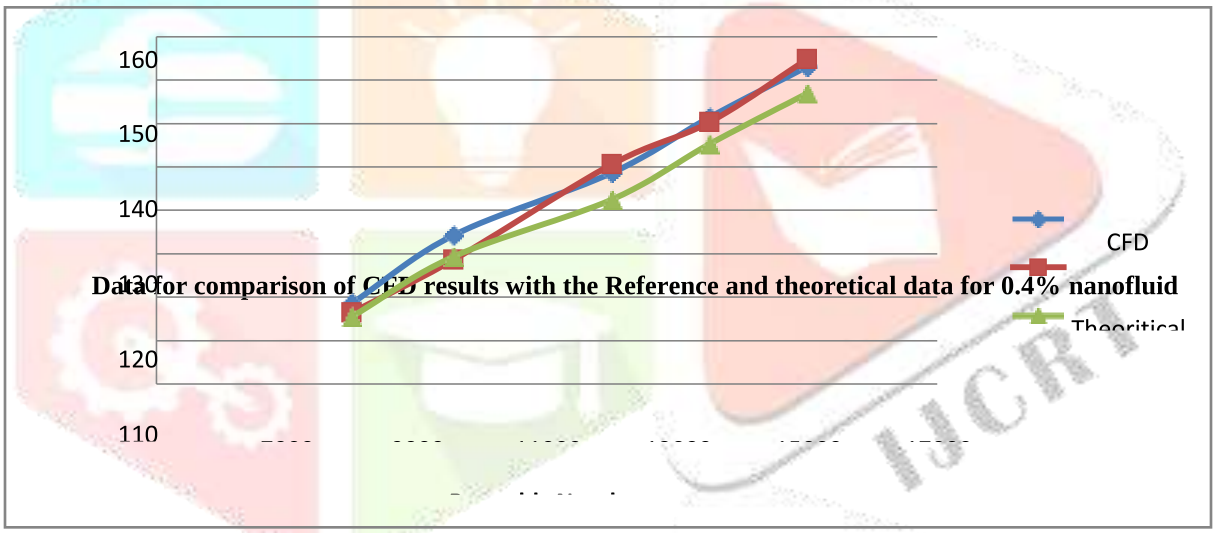
temperature variation in inner tube

temperature variation in outer tube

D/d ratio=10

Reynolds Number	CFD	Theoretical	Reference
8000	97.46	96.42	95.5
9570	114.31	108.69	109.36
12000	128.64	130.65	122.57
13500	141.5	140.3	135.3
15000	153.07	154.79	147

omparison of CFD results with the reference and theoretical data for water for different Reynolds number



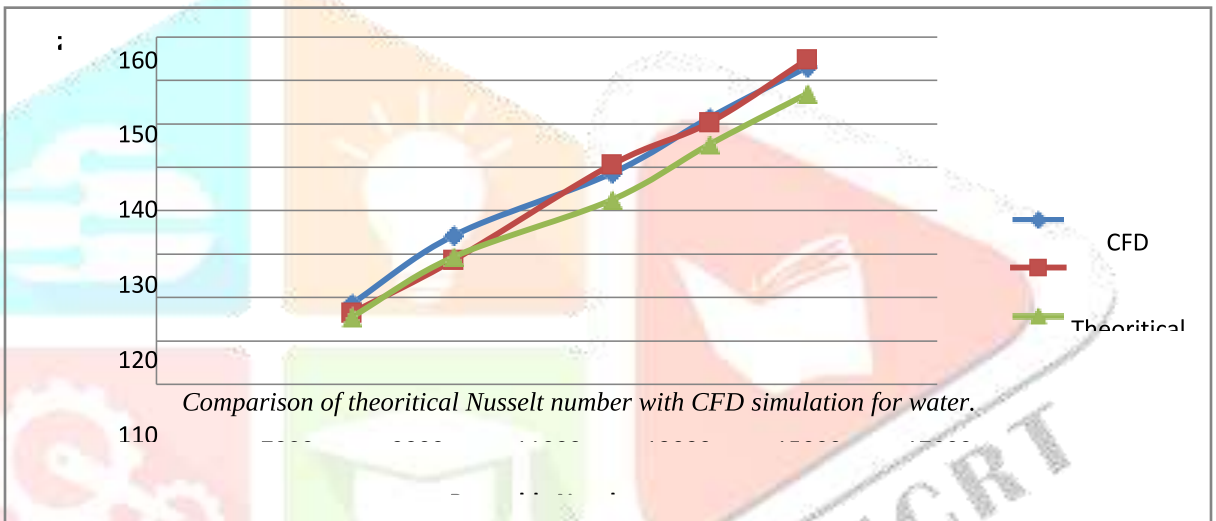
Data for comparison of CFD results with the Reference and theoretical data for 0.4% nanofluid

Comparison of theoritical Nusselt number with CFD simulation for water

Reynolds number	Theoritical (Nu)	CFD (Nu)
12250	126	121.2
14700	144	140
17150	161.7	153.7
19600	178.5	168.2
22000	194.4	187.6

Data for comparison of CFD results with the reference and theoretical data for water for different Reynolds number

Reynolds Number	CFD	Theoretical	Reference
8000	97.46	96.42	95.5
9570	114.31	108.69	109.36
12000	128.64	130.65	122.57
13500	141.5	140.3	135.3
15000	153.07	154.79	147

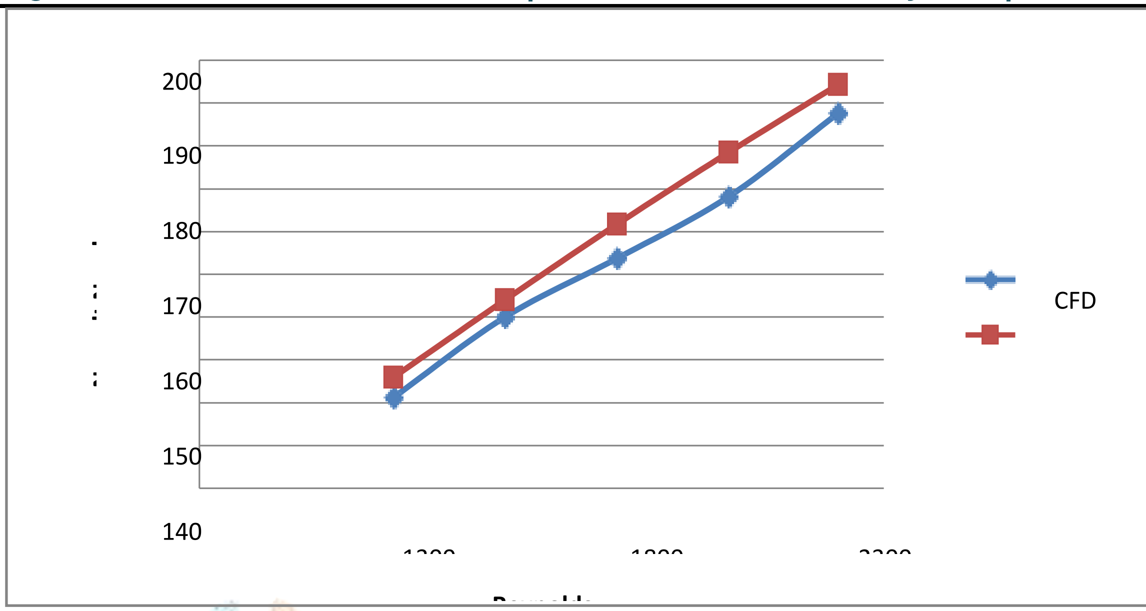


Comparison of theoritical Nusselt number with CFD simulation for water.

The above graph shows comparison of reference and theoretical results with CFD for water under turbulent flow condition. It is observed that the Nu increases with increase in Re number. The Nu is minimum at the Re of 8000 and maximum at 15000. CFD results have been compared with the theoretical results reported by Jayakumar et al. The average relative error between the theoritical and CFD result is about 5.5%. The average relative error between the theoretical results and CFD results is 2.5%.

Reynolds number	Theoritical (Nu)	CFD (Nu)
12250	126	121.2
14700	144	140
17150	161.7	153.7
19600	178.5	168.2
22000	194.4	187.6

Data for comparison of CFD results with the Reference and theoretical data for 0.4% nanofluid



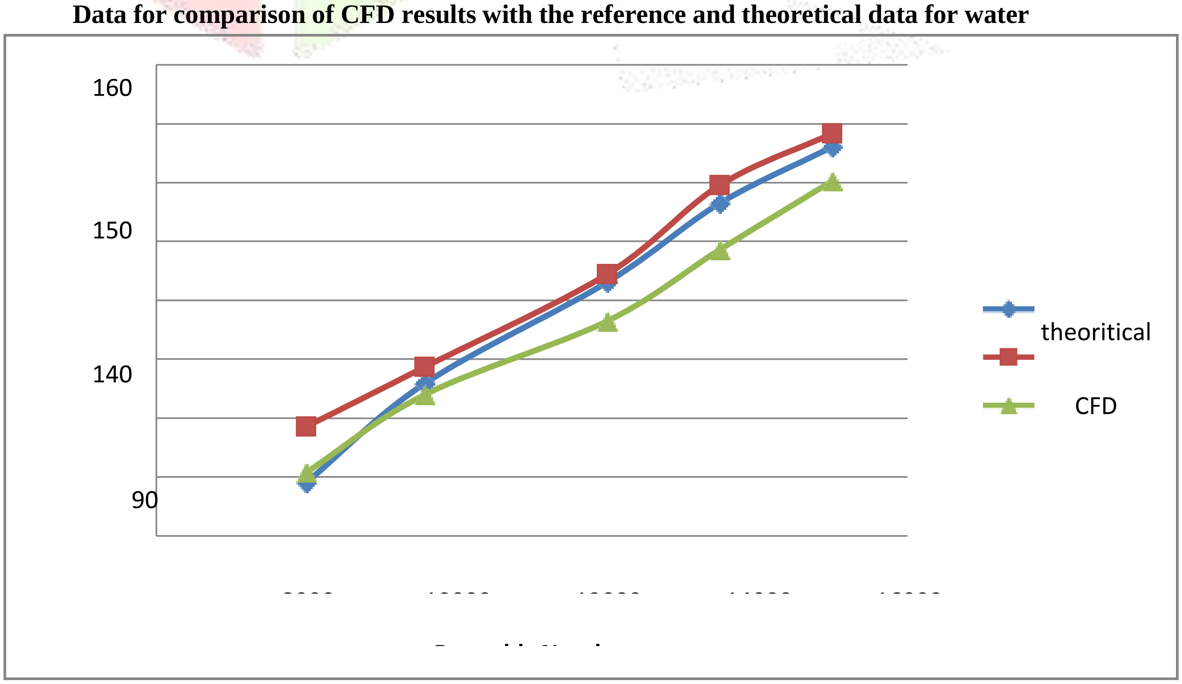
Comparison of theoritical Nusselt number with CFD simulation for 0.4% Al_2O_3-H_2O nanofluid

the average relative error between the theoritical and CFD results is about 4.5%. It is observed that Nu is minimum at the Re of 8000 and the maximum at 15000. The Nusselt number is increased with the increase of Re.

D/d ratio=15

Reynolds number	**CFD**	**theoritical**	**Refernce**
8000	98.53	89	90.7
9570	108.62	105.82	103.91
12000	124.45	123.1	116.47
13500	139.5	136.5	128.56
15000	148.3	146.1	140.27

Data for comparison of CFD results with the reference and theoretical data for water

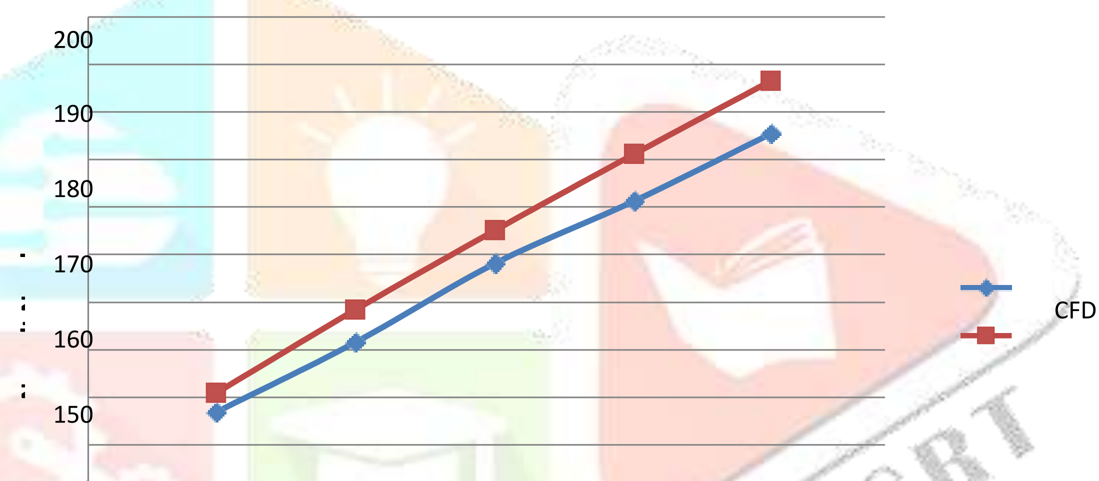


Comparison of theoritical Nusselt number with CFD simulation for water

The Nu is minimum at the Re of 8000 and maximum at 15000. CFD results have been compared with the theoretical results reported by Jayakumar et al. The average relative error between the reference and CFD result is about 7.5%. The average relative error between the theoretical results and CFD results is 6.5%

Reynolds number	Theoritical (Nu)	CFD (Nu)
12250	120.9	116.8
14700	138.4	131.5
17150	155.1	148.2
19600	171.2	161.3
22000	186.5	175.4

Data for comparison of CFD results with the Reference and theoretical data for 0.4% nanofluid



Comparison of theoritical Nusselt number with CFD simulation for 0.4% Al_2O_3-H_2O nanofluid

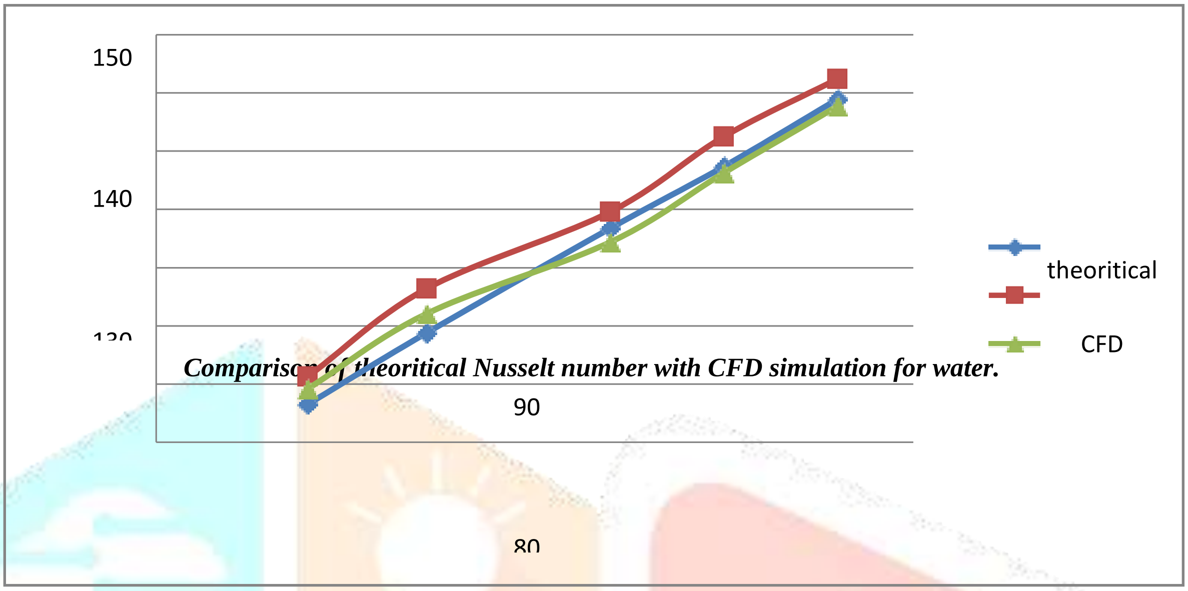
the average relative error between the theoritical and CFD results is about 8%. It is observed that Nu is minimum at the Re of 12000 and the maximum at 22000. The Nusselt number is increased with the increase of Re

D/d ratio=20

Reynolds number	CFD	Theoritical	Reference
8000	91.23	86.47	89.15
9570	106.38	98.73	102.037
12000	119.52	116.73	114.33
13500	132.5	127.36	126.24

15000	142.35	138.9	137.74

Data for comparison of CFD results with the reference and theoretical data for water



Comparison of theoritical Nusselt number with CFD simulation for water.

Reynolds Number	**Theoritical (Nu)**	**CFD (Nu)**
12250	117.6	111.3
14700	134.6	121.5
17150	150.8	141.2
19600	166.5	157.3
22000	181.4	169.4

Data for comparison of CFD results with the Reference and theoretical data for 0.4% nanofluid

Variation of friction factor with Reynolds number for different D/d ratio with inner tube as hot water-

Table 5.12 Data for CFD evaluated friction factor with Reynolds number for hot water

Reynolds number	**D/d=10**	**D/d=15**	**D/d=20**	**D/d=25**	**D/d=30**
8000	0.212	0.166	0.1416	0.138	0.121
9570	0.209	0.162	0.1378	0.131	0.115

12000	0.196	0.156	0.135	0.125	0.11
13500	0.19	0.153	0.1314	0.121	0.108
15000	0.188	0.15	0.1298	0.117	0.104

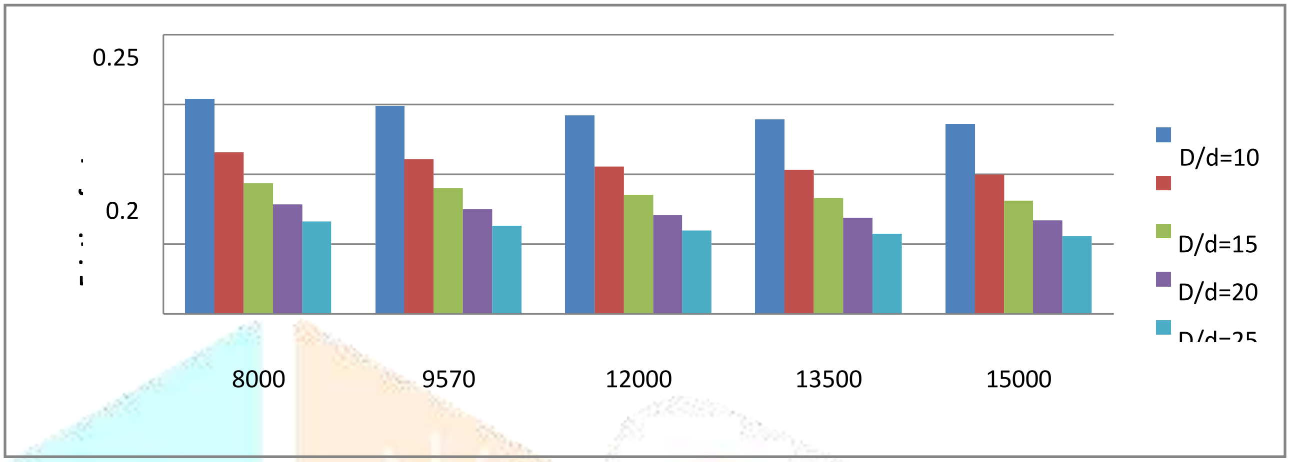


Fig 5.15 comparison CFD evaluated friction factor with Reynolds number for different D/d ratio

From the above graph we observe that with increase in D/d ratio friction factor decreases.Friction factor has maximum value for D/d ratio=10 which is 0.212 and minimum for D/d ratio=30 which is 0.104.

Reynolds number	**D/d=10**	**D/d=15**	**D/d=20**	**D/d=25**	**D/d=30**
8000	0.204	0.165	0.143	0.128	0.116
9570	0.199	0.160	0.14	0.124	0.113
12000	0.192	0.155	0.135	0.120	0.109
13500	0.189	0.153	0.132	0.118	0.107
15000	0.186	0.149	0.130	0.116	0.105

Table 5.13 Data for theoretically evaluated friction factor with Reynolds number using hot water.

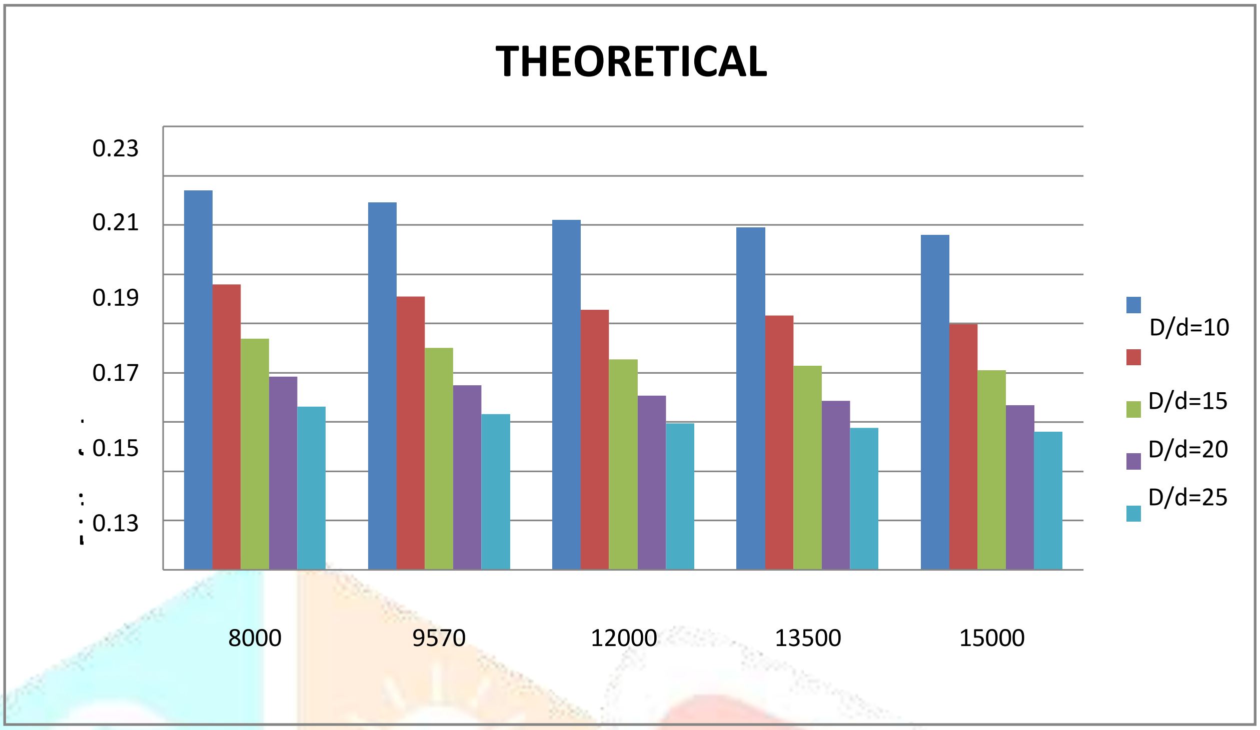


Fig 5.16 comparison theoretically evaluated friction factor with Reynolds number for different D/d ratio using water.

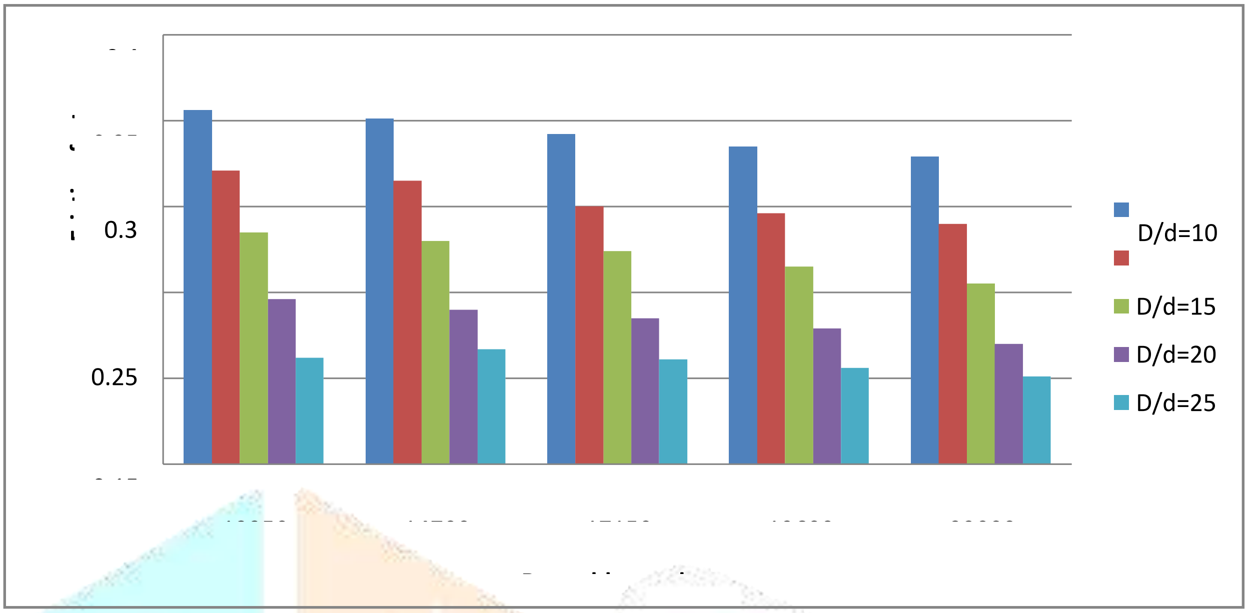
From the above graph we observe that with increase in D/d ratio friction factor decreases.Friction factor has maximum value for D/d ratio=10 which is 0.204 and minimum for

D/d ratio=30 which is 0.105.

Table 5.15 Data for CFD evaluated friction factor with Reynolds number using

0.4% Al_2O_3-H_2O nanofluid.

Reynolds number	**D/d=10**	**D/d=15**	**D/d=20**	**D/d=25**	**Dd=30**
12250	0.356	0.321	0.285	0.246	0.212
14700	0.351	0.315	0.28	0.24	0.217
17150	0.342	0.3	0.274	0.235	0.211
19600	0.335	0.296	0.265	0.229	0.206
22000	0.329	0.29	0.255	0.22	0.201



comparison of CFD evaluated friction factor with Reynolds number for different D/d ratio using 0.4% Al_2O_3-H_2O nanofluid.

The above graphs show the variation of friction factor with respect to Reynolds number for different D/d ratio.The friction factor will decrease with increase in Reynolds number. The variation of friction factor for a particular Reynolds number is maximum between D/d=10 to D/d=15.As we increase the curvature ratio,the variation will decrease.

Data for CFD evaluated Nusselt number with Dean number using hot water in inner tube dor different D/d ratio

Dean number	**D/d=10**	**D/d=15**	**D/d=20**	**D/d=25**	**D/d=30**
2530	98.6	98.03	91.23	90.2	87.12
3025	114.31	108.62	106.38	105.32	102.56
3800	128.64	128.7	119.52	118.59	113.62
4270	141.5	139.5	132.5	126.63	123.45
4740	153.07	148.3	142.35	137.89	137.58

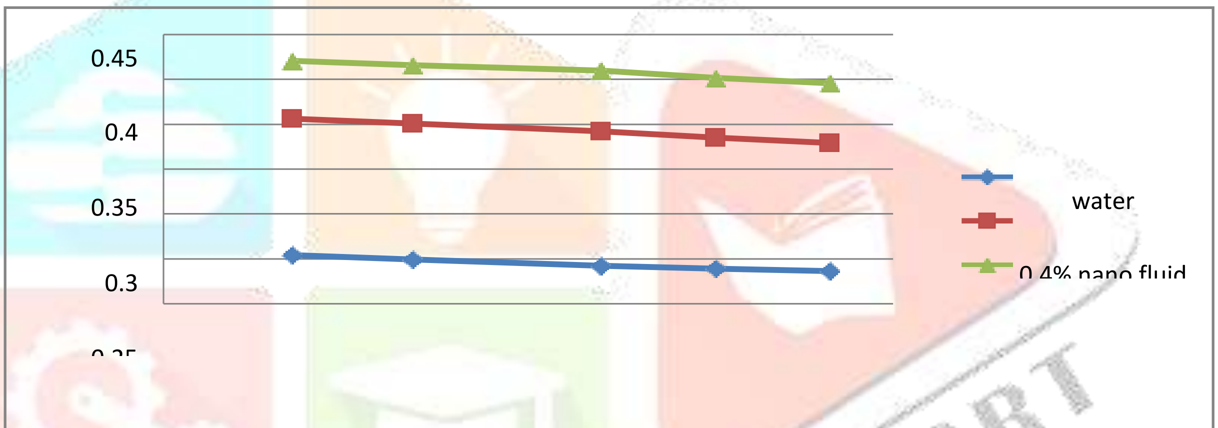
The above data shows comparison of nusselt number for different D/d ratio with respective to dean number for water under turbulent flow condition. It is observed that the Nu increases with increase in dean number. The Nu is minimum at de of 2530 and maximum at 4740. With increases in D/d ratio (inverse of curvature ratio) the Nusselt number decreases.

Data for CFD evaluated Nusselt number with Dean number using 0.4% Al_2O_3-H_2O nanofluid in inner tube For different D/d ratio.

Dean number	**D/d=10**	**D/d=15**	**D/d=20**	**D/d=25**	**D/d=30**
2530	121.2	116.8	111.3	110.1	109.3
3025	140	131.5	121.5	125.6	120.6
3800	153.7	148.2	141.2	139.5	135.2
4270	168.2	161.3	157.3	152.4	149.5
4740	187.6	175.4	169.4	168.7	165

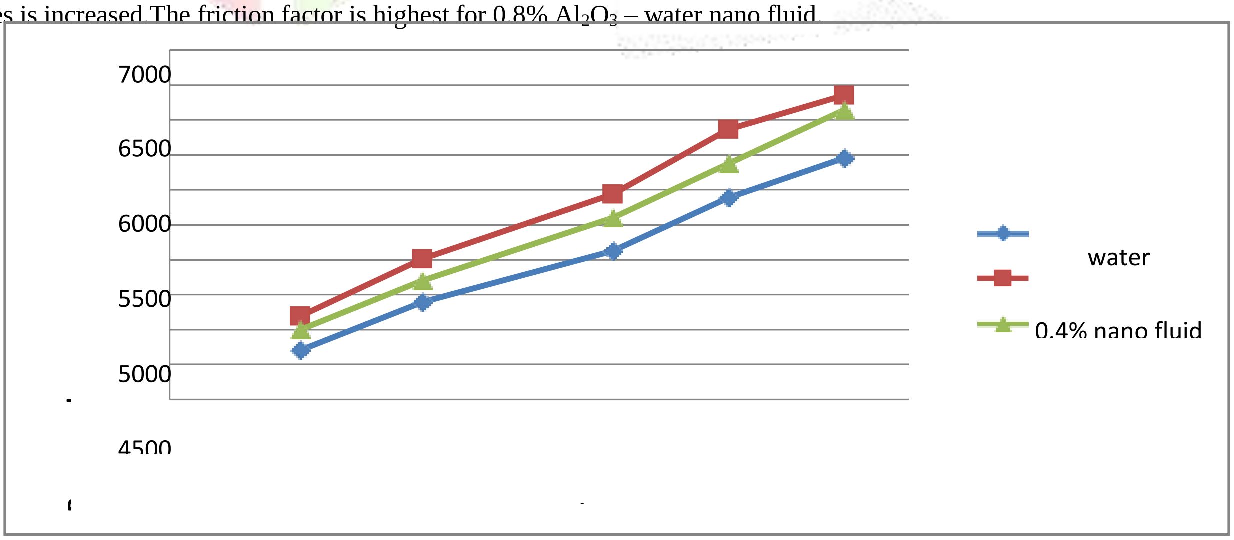
comparison of CFD evaluated Nusselt number with Dean number for different D/d ratio using 0.4% Al_2O_3-H_2O nanofluid.

The above data shows comparison of nusselt number for different D/d ratio with respective to dean number for 0.4% Al_2O_3-H_2O nanofluid under turbulent flow condition. It is observed that the Nu increases with increase in dean number. The Nu is minimum at de of 2530 and maximum at 4740. With increases in D/d ratio (inverse of curvature ratio) the Nusselt number decreases.The graph illustrates the effect of nano particles for different D/d ratio. it is clear that friction factor increases with increase in D/d ratio.



comparision of friction factor with different nano fluid concentrations.

It is observed that water has less friction factor compared to the nano fluid and also observed that as the particle concentration increases ,friction factor increases.This is because of as the particle concentration increases ,the Brownian motion and colloidal motion of the particles is increased.The friction factor is highest for 0.8% Al_2O_3 – water nano fluid.



comparision of pressure drop with Dean number.

From the above graph it is found that the pressure drop increases with increasing particle volume concentration and De. The pressure drop of 0.4% and 0.8% are 5% and 9% respectively higher than water at De=2650.This is because of increased viscosity when particle volume concentration is increased. It is observed that the rate of pressure drop is increased when Dean Number is increased. The rate of increase of pressure drop at De=2530 is smaller than at De=4740. The rate of pressure drop is higher at higher De and lower at lower De. The maximum CFD pressure 5457Pa for water, 5800 Pa for 0.4% nanofluid and 6144 Pa for 0.8% nanofluid at the De of 4740

CONCLUSION

. Characteristics of the fluid flow were studied for the constant wall heat flux conditions. Several important conclusions could be drawn from the present simulations and would be presented as follows:

- The fluid particles were undergoing an oscillatory motion inside both the pipes.It is visible from the results that Nusselt Number depends on curvature ratio. It is increasing with increase in curvature ratio.).With decrease in D/d ratio (inverse of curvature ratio) the Nusselt number increases and frictional factor will decreases; for a particular value of Reynolds number. Nusselt number and frictional factor has maximum value for D/d=10 and minimum value for D/d=30.
- Along the outer side of the pipes the velocity and pressure values were higher in comparison to the inner values.
- In addition, the value of Nusselt number was found to increase with increase in mass flow rate.
- With tube-in-tube helical coil heat exchangers it is possible to accommodate larger heat transfer surface area within small space when compared with straight tube arrangement.
- Increase in coil pitch weakens the secondary flows with the same Reynolds number and ultimately approaches to straight tube characteristics. Therefore, when coil pitch is increased, heat transfer coefficient of the fluid decreases.
- Friction factor decreases with increase in Reynolds number due to relative roughness of surface and velocity of flowing fluid.
- As long as the heat transfer is concerned from the hot fluid any boundary condition can be assumed for outer wall of external tube because it does not affect significantly the heat transfer rate.
- The optimum Reynolds number decreases with increase in D/d ratio

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