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Numerical Analysis of Fe_3O_4 Nanofluid Flow in a Double Pipe U-Bend Heat Exchanger

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Abstract

Numerical analysis (Computational Fluid Dynamics) based on the single phase fluid approach method has been used to estimate the convective heat transfer and friction factor characteristics of Fe_3O_4 nanofluid flow in a two-pass double pipe U-bend heat exchanger in the Reynolds number range from 9000 to 24000 at constant heat flux boundary conditions. The particle volume concentrations used in this study are 0.03, 0.1, 0.2, 0.3 and 0.4 and the nanofluid mass flow rate range from 0.134 to 0.267 kg/sec. The double pipe heat exchanger is modeled in the GAMBIT software by considering stainless steel as inner tube material and galvanized iron as a annulus tube material. The temperatures of nanofluids flow in a heat exchanger are kept at 333 K. The 3D solid modeling and meshing of the heat exchanger was developed using ANSYS 14.0 workbench. The results revealed that as volume fraction and Reynolds number increased Nusselt number increased, and friction factor decreased. The heat transfer enhancement of 34.93% is observed at 0.4% volume concentration of nanofluid at a Reynolds number of 24000 with a friction penalty of 1.34-times when compared to water data.

Keywords: Numerical analysis, heat exchanger, heat transfer, friction factor, enhancement.

1. INTRODUCTION

Cooling is one of the prime important technical challenges facing by numerous industries such as automobiles, electronics, chemical, food and manufacturing etc. The thermal conductivity of heating or cooling fluids is a very important property for the development of energy efficient heat transfer equipments. Meanwhile, in all the processes involving heat transfer, the thermal conductivity of the fluid is one of the basic important properties taken in to account in designing and controlling the processes. Nanofluids are engineered colloids which are made of a base fluid and nanoparticles of (1-100) nm. It has been found by many researchers that the nanofluids provide higher thermal conductivity compared to base fluids. Its value increases with the increase in particle concentration, temperature, particle size, dispersion and stability. Nevertheless, it is expected that other factors like density, viscosity, and specific heat are also responsible for the convective heat transfer enhancement of nanofluids. Nanofluids are having high thermal conductivity and high heat transfer coefficient compared to single phase fluids. One of the most widely-used equipment in various applications, such as petrochemical, oil and gas refinery, thermal power generation etc., are heat exchangers that can be used for either heating or cooling fluids. Among them, shell and tube heat exchangers are one of the most familiar. It is the time to optimize these types of heat exchangers.

Moraveji et al., [1] investigated experimentally and numerically the turbulent heat transfer and friction factor in a tube passage for Fe_3O_4 nanofluid by using CFD method. They have found maximum error of 10%. They have developed following correlation for finding Nusselt number and friction-factor by using their modeling results in a horizontal tube in terms of the Reynolds number (Re), Prandtl number (Pr) and particle volume fraction (ϕ) for both water and Fe_3O_4 magnetic Nanofluid.

$$Nu = 0.00248\, Re^{1.03} Pr^{0.5}\, (1+\phi)^{47.5} \quad (1)$$

Sunder et al., [2] investigated thermal conductivity and viscosity both experimentally and theoretically and they conducted experiments for different volume concentrations range from 0.0% to 2.0% and temperature range from 293K to 333K. Sundar et al., [3] also investigated experimentally with the effect of thermal conductivity of ethylene glycol and water mixture based Fe_3O_4 nanofluid. They conducted experiments in the temperature range of 293K to 333K and at different volume concentrations. Vajjha and Das [4] conducted thermal conductivity experiments for three different nanofluids and proposed correlations. Pak &Cho [5] experimentally examined on the effect of friction-factor and behavior of suspended particles. In their study, they have used two separate metal oxide particles that are of Al_2O_3 and TiO_2. They found that the Nusselt Number increases with the increase in volume concentration and Reynolds number. They developed new correlation for Nusselt number valid for Al_2O_3 and TiO_2 nanofluid under turbulent flow conditions.

$$Nu = 0.021 Re^{0.8} Pr^{0.5} \quad (2)$$

2. EXPERIMENTAL EQUIPMENT AND PROCEDURE

2.1. Description of experimental setup

Fig. 1 represents the photograph of an experimental setup and Fig. 2 represents the test section details. The experimental setup consists of two concentric tube heat exchangers. The inner pipe is made of stainless steel (SS304), inside diameter of inner pipe is 19 mm and outside diameter is 25 mm, the outer tube inner diameter 50 mm and outer diameter is 56 mm made of G.I. pipe. Each section is approximately 2.2 m long. Hot water, which comes from the nearby hot water tank, is passed through the outer pipe and cold nanofluid/water, coming from the supply main is passed through the inner pipe. Valves on both lines are also provided to control the flow rates of the streams. Each section is provided with thermocouples, to measure the temperature of the streams at appropriate points along the heat exchanger. Thermocouples are used to measure the temperature of the fluids at five different locations. Inlet, midpoint and exit of the cold flow stream and inlet and outlet of hot flow stream were recorded. Each thermocouple is an Omega Model T-Type.The thermometer accuracy is of ±0.1°C and in the range 0°C to 70°C. A data acquisition system has been set up to measure the temperatures based on a pre-existing calibration.

Fig. 1 Experimental set up of double pipe U-bend heat exchanger

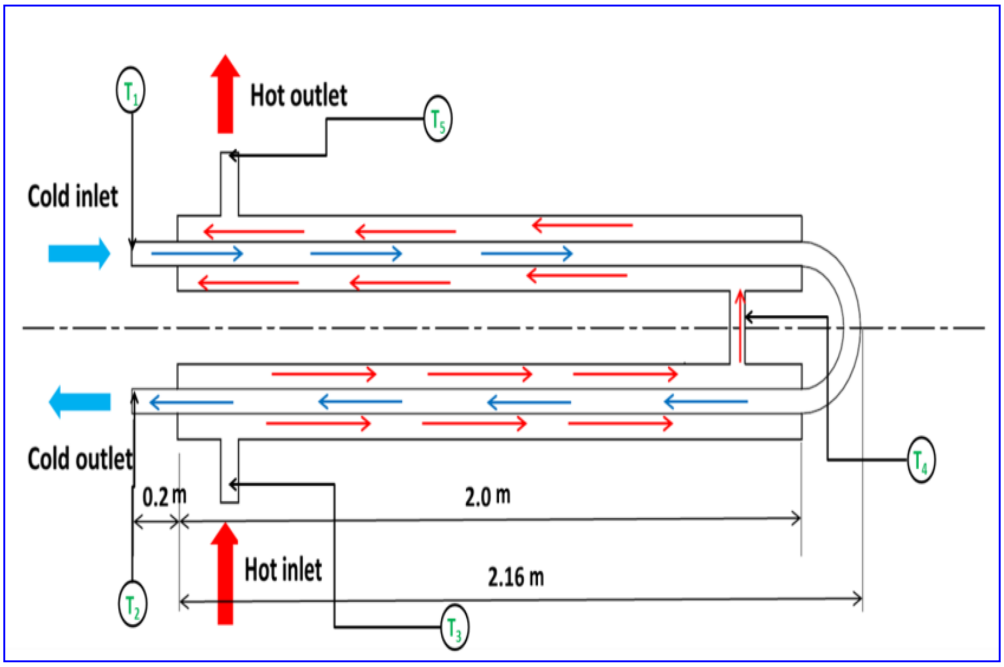

Fig. 2 Schematic representation of double pipe U-bend heat exchanger

2.2. Geometry, modeling and boundary conditions

The analysis is performed on a 2-pass double pipe heat exchanger with the inner diameter of inner pipe is 0.019 m & outer diameter of inner pipe is 0.025 m, similarly for annulus pipe, the inner diameter of outer pipe is 0.05 m & outer diameter of outer pipe is 0.056 m and the total length of each section of the heat exchanger is 2.2 m (2-pass). The mass flow rate of hot water is m_h (kg/s) is kept constant in annulus section, with different temperatures and the mass flow rates of cold water m_c (kg/s). There is insulation for outer wall of annulus pipe with asbestos rope to minimize the heat losses. Geometry model is shown in Fig. 3.

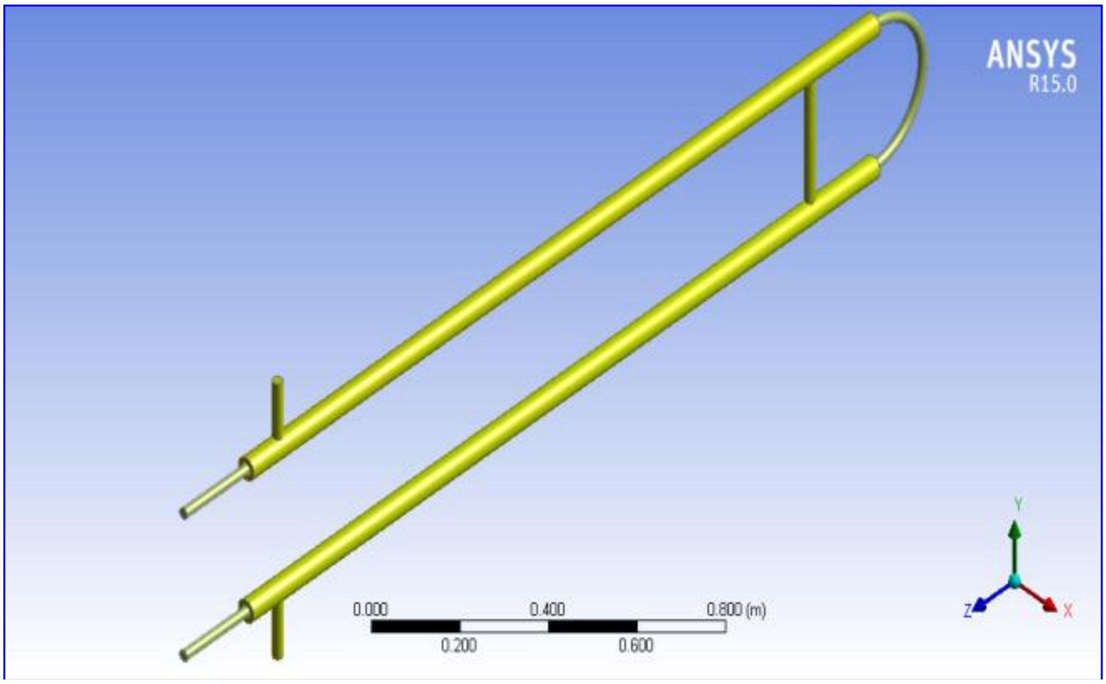

Fig. 3 Geometry modeling of 2-Pass Double Pipe Heat Exchanger in ANSYS work bench

2.3. Meshing of geometry

Structured meshing method in ANSYS Work bench was used for the geometry. The element for meshing considered is hexahedral shape with number of elements of 876874 to 1240000. Naming selections were also done at required places.

2.4. Grid independence test

Grid Independence is the term used to designate the enhancement of results by using successively smaller cell sizes for the calculations. A calculation should reach the correct result so that the mesh becomes smaller; hence the term is known as grid Independence. The ordinary CFD technique is, to start from coarse mesh and gradually improve it until the changes detected in the values are smaller than a pre-defined acceptable error. There are 2 problems with this. Firstly, it can be quite difficult with other CFD software to gain even in a single coarse mesh resulting for some problems. Secondly, refining a mesh by a factor two or above can lead to take more time. This is clearly offensive for software intended to be used as an engineering tool design operating to constricted production limits. In addition to that the other issues have added significantly to the perception of CFD as an extremely difficult, time consuming and hence costly methodology. Finally grid independence test has been conducted at a flow rate of 8 lpm hot water, 10 lpm cold water flow rates in ANSYS-FLUENT, by decreasing and increasing the size of the elements. The gained results are tabulated in Table 1, for outlet temperatures of cold water and hot water of 2-pass double pipe heat exchanger. Final mesh elements of 1124397 have been used for further simulation purpose.

Table 1 Grid test results

No. of Elements	Coldwater Outlet Temperature (°C)	Hot water Outlet Temperature (°C)
876874	31.458	53.970
895812	31.652	53.625
856253	30.256	54.325

2.5. Physical model

The standard k-ε model is used for single phase turbulent flow in circular pipe channel. Based on the Reynolds number, $\rho v d/\mu$ either viscous laminar or standard k-ε model is used for laminar and turbulent flow respectively. The choice of the model is shown in Table 2. Where (d) is the diameter of the pipe, (ρ) is are the density and (μ) is viscosity of the fluid.

Table 2 Physical Model

Reynolds Number (Re)	Flow (Model)
< 2000	Laminar Flow
>2000	Turbulent Flow (k-ε model)

2.6. Material properties

Pure water is used as base fluid, Stainless Steel is used for inner pipes and Fe_3O_4/water used as cold fluid, the properties are shown in Table 3.

Table 3 Properties of Water and Fe_3O_4 nanoparticles

Substance	Mean Diameter	Density (kg/m^3)	Thermal conductivity (W/m-K)	Specific heat (J/kg-k)	Kinematic Viscosity (m^2/s)
Fe_3O_4 nanoparticle	50 nm	5180	80.4	670	--
Water	--	997.5	0.6129	4178	0.000829

The thermo physical properties of Fe_3O_4/water nanofluid such as density (ρ), specific heat (C_p), thermal conductivity (k) and viscosity (μ) are calculated by using following correlations developed. For density, the relation given by Pak and Cho [5], for effective thermal conductivity Hamilton-Crosser [6] relation, for viscosity Brinkman model [7] relation and for Specific heat, Xuan and Li [8] equations were used. The particle size of the Fe_3O_4 nanoparticles is considered as 50 nm to 100 nm. The properties of nanofluid are given in Table 4, at 27°C temperature. The equations for finding the properties are given below.

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p \quad (3)$$

$$C_{p,nf} = \frac{(1-\phi)\rho_{bf}C_{p,bf} + \phi\, \rho_p C_{p,p}}{\rho_{nf}} \quad (4)$$

$$\frac{k_{nf}}{k_{bf}} = \frac{\frac{k_p}{k_{bf}} + (n-1) - (n-1)\left(1-\frac{k_p}{k_{bf}}\right)\phi}{\frac{k_p}{k_{bf}} + (n-1) + \left(1-\frac{k_p}{k_{bf}}\right)\phi} \quad (5)$$

Table 4 Steel, water and nanofluid properties at 27°C

Properties	Steel	Water	0.03%	0.1%	0.2%	0.3%	0.4%
$\rho(\frac{kg}{m3})$	8030	997.5	998.68	1001.6	1005.8	1010.0	1014.23
C_p(J/Kg.K)	502.4	4182	4172.5	4159.8	4141.8	4124.0	4106.33
K(W/mK)	16.27	0.6000	0.6134	0.6147	0.6165	0.6183	0.6201
μ(m^2/S)	---	0.0010	0.0008	0.0008	0.0008	0.0008	0.00083

2.7. Governing equations

Generally steady state simulations were taken out by solving mass, momentum and energy equations for single phase fluid, which are expressed as:

Continuity equation:

$$\partial\rho/\partial t + \partial/\partial X\ (\rho u\_x\ ) + \partial/\partial r\ (\rho u\_r\ ) + (\rho u\_r)/r = 0 \quad (6)$$

Momentum equation:

$$(\partial(\partial u\overline{\ }))/\partial t + (\rho(u.u)\overline{\ }) = \rho g - \nabla P + \nabla.(\tau\overline{\ }) \quad (7)$$

Energy equation:

$$(\partial(\partial E))/\partial t + \nabla(u\overline{\ }(\rho E + P)\ ) = \nabla.(k\_eff.\nabla T) \quad (8)$$

Where (ρ) is the density, (u) is the velocity, (P) is the pressure,(τ) is the viscous stress, (E) is the energy and (k_{eff}) is the effective thermal conductivity. Turbulent flows are characterized by altered/changed velocity fields. These fluctuations mix transported quantities such as momentum, energy, and species concentration, and cause the transported quantities to fluctuate as well. Because of these, changes can be of small scale and high frequency, they are so computationally affluent to simulate directly in practical engineering calculations. Instead, the instantaneous (exact) governing equations can be time averaged, ensemble averaged, resulting in a modified set of equations which contain additional unknown variables, and turbulence models are needed to determine these variables in terms of known quantities.

2.8. Boundary conditions

A Velocity inlet, uniform mass flow inlets and a constant inlet temperature were assigned at the channel inlet. At the exit, pressure was specified.

Table 5 Boundary Conditions

Boundary type	Annulus Pipe	Inside Pipe
Mass flow rate at Inlets	0.134 kg/s	0.134 to 0.267 kg/s
Temperatures	333 K	300 K
Constant heat flux at pipe wall (Insulation)	0 W/m^2	---

2.9. Method of solution

The CFD method follows the use of commercial software ANSYS FLUENT 15.0 for solving the problem. The solver in ANSYS-FLUENT used is a pressure correction based SIMPLE algorithm with 2nd order upwind scheme to discretise the convective transport terms. The criteria for convergence dependent variables are specified as 0.001. In the present analysis, the analytical values of heat transfer coefficients are calculated. The heat transfer coefficients are also obtained using CFD methods and compared with analytical values. After determining the important features of the problem following procedure is adapted for solving the problem, in which first of all we need to specify the solution method, and initialize the solution, then run the calculation. Initially create geometry model in the ANSYS workbench, as per the experimental set up design. Meshing was done on the geometry model by program controlled and sizing was done to get the required element size, nodes and smoothening. After getting the required size of element and meshing, naming selection was done to the domain before getting the results. After meshing is completed, open the setup in the project schematic in fluent, where governing equations are selected like Energy, Viscous- Standard k-ϵ (2 eq), standard wall function to be given to necessary equations to simulate, material creating and boundary conditions to be given and methods to calculate the moment, pressure etc., by using standard finite element method equations. By selecting second order upwind scheme for solving, finally after converging the equations results were obtained.

3. RESULTS AND DISCUSSION

3.1. Validation of numerical results

Numerical results are compared with Analytical results. Nusselt number for nanofluid at different volume concentrations in the turbulent condition are compared with that of Sundar and Sharma [9] correlations. The CFD results are plotted and compared with Analytical results as shown in Fig. 4. The numerical Nusselt number values are in very good agreement when compared with the correlated values.

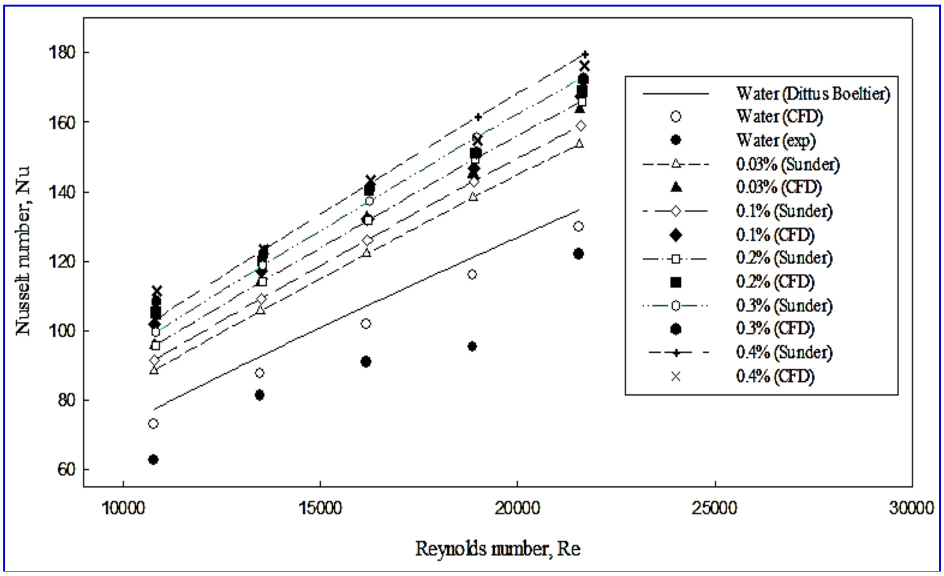

Fig. 4 Comparison of Nu of nanofluid by Sundar and Sharma [9] correlation with Simulation results

3.2. Effect of nanoparticle volume concentration on the Nusselt number

Fig. 5 shows comparison of Nusselt number with corresponding Reynolds number of pure water and Fe_3O_4/water nanofluid. It can be seen that the Nusselt number increases gradually with the increase in Reynolds number. The enhancement of heat transfer coefficient at 0.4% volume concentration of Fe_3O_4 nanofluid is 34.93% for Reynolds number range of 9,000 to 24,000 when compared to water.

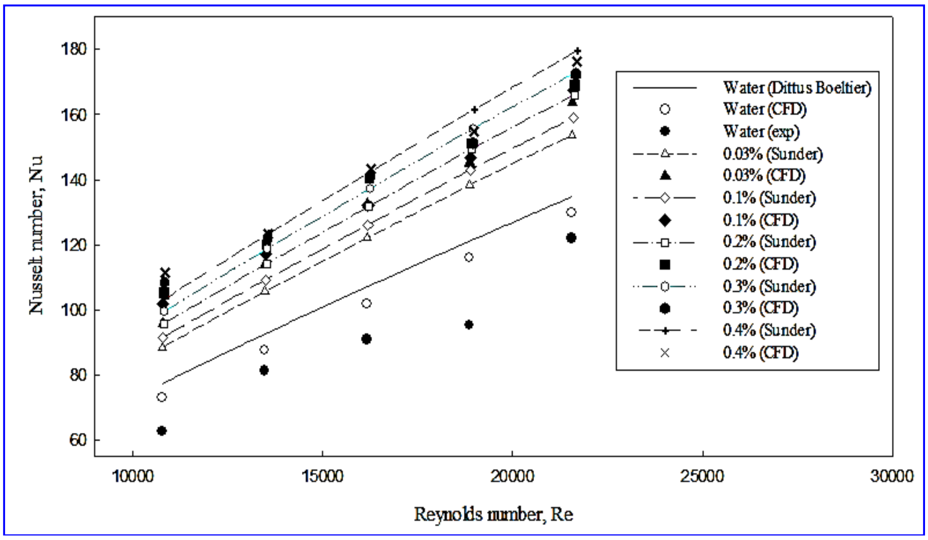

Fig. 5 Comparison of Nu of nanofluid by Sundar and Sharma [9] correlation with Simulation results

Fig. 6 shows the comparison of Numerical Nusselt number values at different volume concentrations of nanofluids. Graph shows that the Nusselt number increases gradually with increasing volume concentration as well as increase in flow rate of Fe_3O_4/water nanofluid.

As shown in Fig. 7, Nusselt number data from the Sundar and Sharma [9] correlation were compared with simulation results, from which the maximum and minimum errors were found to be 10.56% and 5.9% respectively.

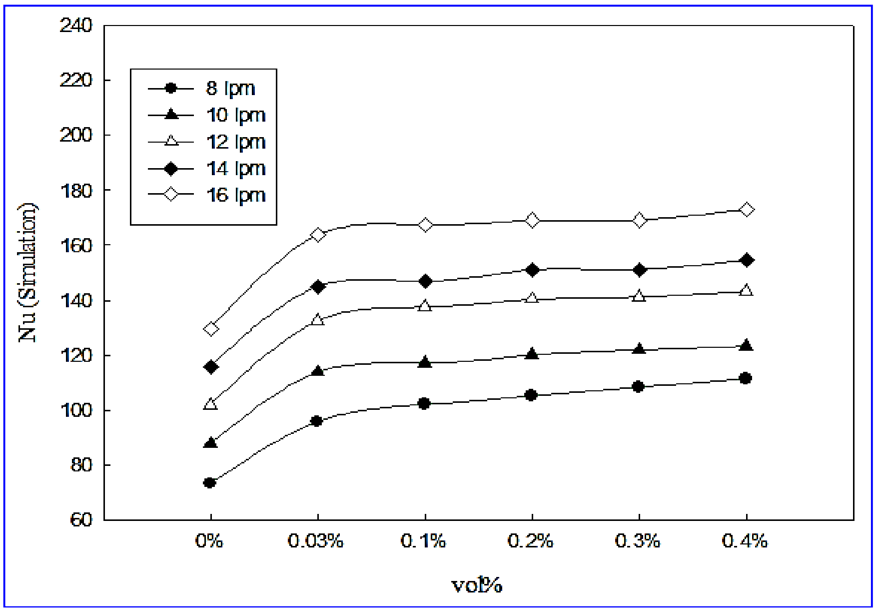

Fig. 6 Comparison of Numerical Nu with Volume concentration at different flow rates

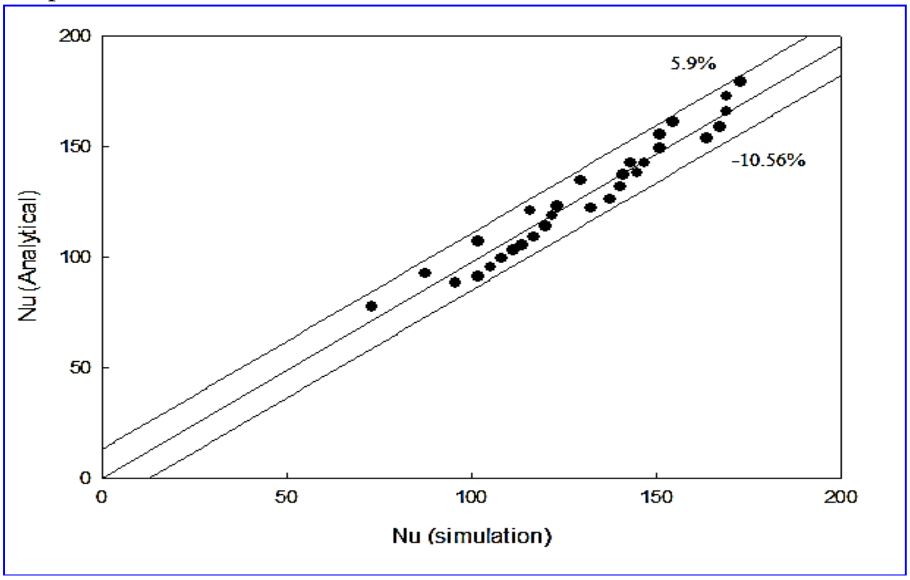

Fig. 7 Comparison of Numerical Nu with Volume concentration at different flow rates

3.3. Effect of nanoparticle volume concentration on friction factor

Fig. 8 shows comparison of friction factor values obtained analytically and Simulation results at corresponding Reynolds numbers. It was observed that the friction factor values are closer to the values obtained by correlation of Sundar and Sharma [9]. There is a decrease in friction factor gradually with increase in Reynolds number. The maximum friction factor of 1.34 times at 0.3% volume concentration of Fe_3O_4 nanofluid at Reynolds number of 10,833 when compared to water.

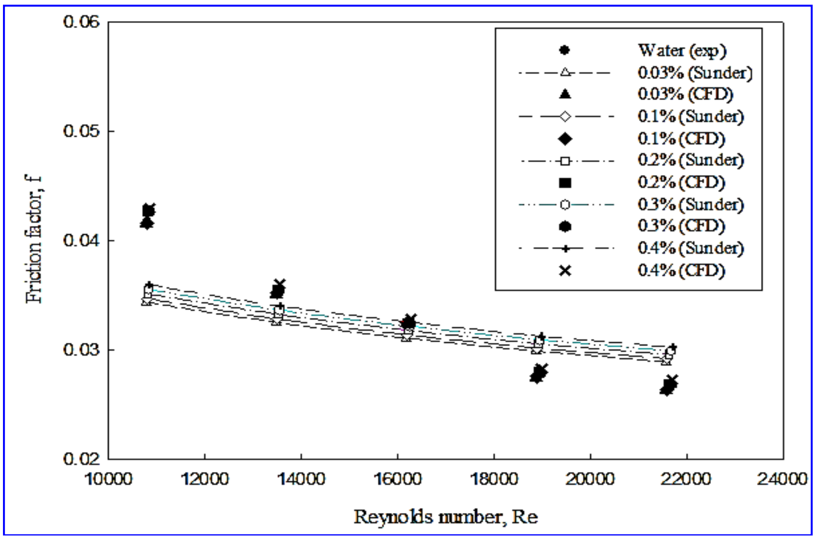

Fig. 8 Comparison of friction factor by Sundar and Sharma [9] correlation with Simulation results at different Reynolds Number.

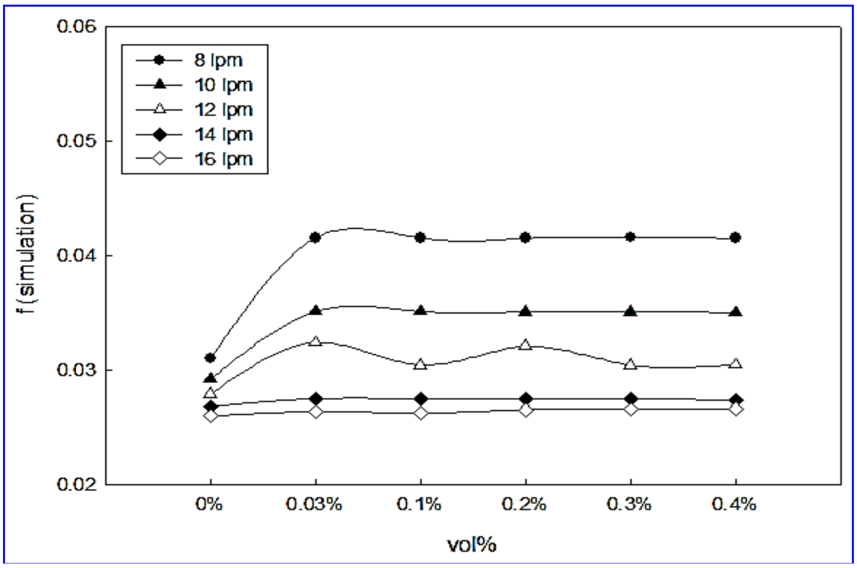

Fig. 9 Comparison of simulation results of friction factor and volume concentrations at different flow rates.

Fig. 9 shows comparison of friction factor obtained from the simulation results at different volume concentrations. It was observed that there is a slight increase in friction factor with the increase in nanofluid volume concentration.

3.3. Temperature contours

Fig. 10 shows the temperature contours of inside pipe of double pipe 2 pass heat exchanger. From the figure it was observed that temperature of inside fluid i.e. Fe_3O_4/water nanofluid gradually increased from inlet to the outlet of pipe.

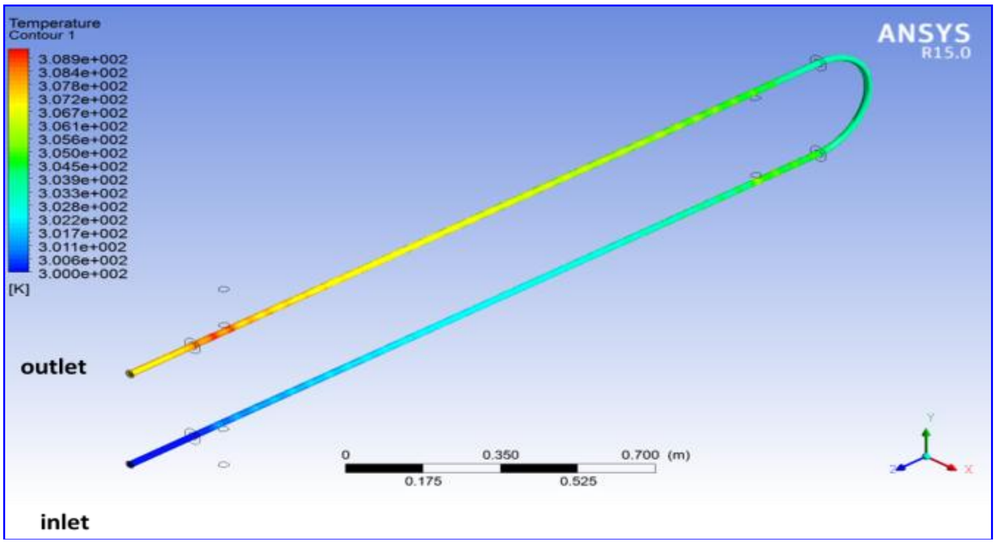

Fig. 10 Temperature contours of inside pipe

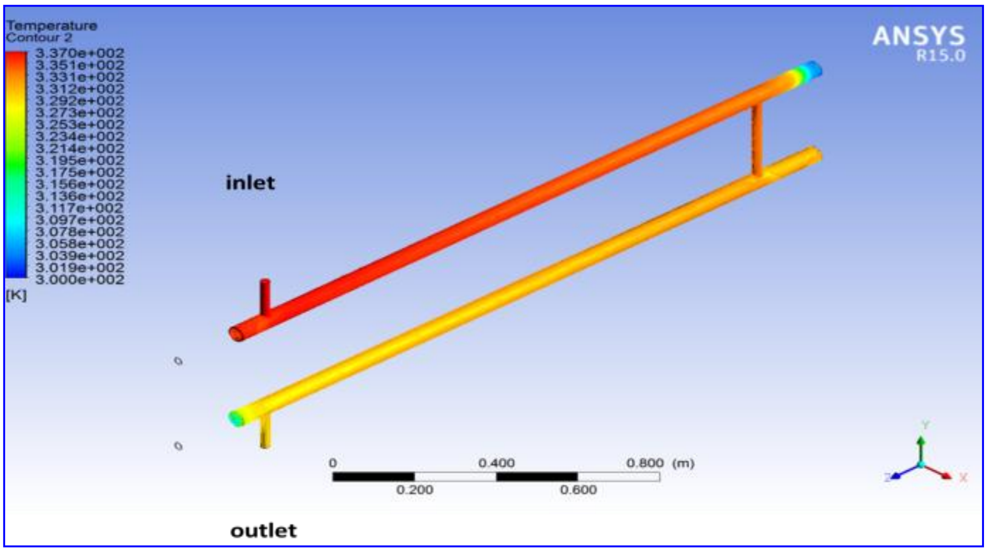

Fig. 11 Temperature contours of annulus pipe

Fig. 11 shows the temperature contours of annulus pipe of double pipe 2 pass heat exchanger. From the figure it was observed that temperature of annulus fluid i.e., pure water gradually decreases from inlet to the outlet of annulus pipe.

4. CONCLUSIONS

A steady state Computational Fluid Dynamics (CFD) models were simulated by ANSYS FLUENT 15.0 and the effect of Reynolds number and Nusselt number on the flow behavior of the nanofluid in the pipe were studied and the variations in the properties are presented. The heat transfer enhancement is observed to be better in the turbulent region for all volume fractions considered in the analysis. There is a good agreement between the results gained from the simulation and analytical data. The maximum error was found that 10.56%. It is observed that according to simulation results there is a 34.93% enhancement in heat transfer coefficient at 0.4% volume-concentration of nanofluid when compared to water at Reynolds number range of 9000 to 24,000. It is observed that there is a maximum friction penalty of 1.34 times at 0.3% volume concentration of Fe_3O_4/water nanofluid at Reynolds number of 10833 when compared to water. The friction-factor is increased with the increase of volume concentration but it is observed that the friction-factor enhancement is less compared to the enhancement to the heat transfer for volume fraction considered in the analysis.

Nomenclature

K	Thermal conductivity of the material ($Wm^{-1}K^{-1}$)
A	Cross sectional area of heat Transfer (m^2)
h	Coefficient of convective heat transfer ($Wm^{-2}K^{-2}$)
Q_c & Q_h	Heat transfer rates of the cold and hot water (W)
m_c & m_h	Mass flow rates of cold and hot water (Kg/Sec)
Cp_c & Cp_h	Specific heat of cold and hot water (J/Kg-K)
$T_{c\text{-}in}$ & $T_{c\text{-}out}$	Inlet and outlet temperatures of cold water (0^oC)
U	Overall heat transfer coefficient (W/m^2K)
V	Velocity of water (m^2/Sec)
ρ	Density of water (Kg/m^3)
h_h & h_c	Heat transfer coefficient on hot and cold water side (W/m^2K).
di & do	Inside and outside tube diameter (m)
D_i & D_o	Inside and outside diameter of annulus Pipe (m)
Q	Rate of heat transfer (W)
Re, Nu &Pr	Reynolds, Nusselt and Prandtl numbers

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