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Computational Analysis of Convective Heat Transfer Enhancement in Double Pipe Helical Coil Heat Exchanger using SiO_2 –Water Nanofluid with Mixture Model

Neha Maheshwari[1], Madhusmita Sahu[2], Saurabh Kumar[3]

[1]Mtech, ME, NIU, U.P. India
[2]Assistant Professor, ME, NIU, UP, India
[3]Ph.D, ME, UPTU, U.P. India

Abstract

The heat transfer enhancement and flow characteristics in helical coil heat exchanger has been studied and researched by many researchers. A numerical study is carried out in double pipe helical coil heat exchanger using nanofluid. SiO_2–water is used as a nanofluid with different volume fractions in the range of 1–4%. For nanofluids, MIXTURE model was used. The graphs are plotted for Nusselt number, Heat transfer coefficient, and Total surface heat flux using ANSYS 15 where the finite volume method (FVM) and the k–ε standard turbulent model are used to solve the main governing equations in the domain. Results are indicated that the Nusselt number for SiO_2-Water nanofluid is greater than that of the base fluid and heat transfer enhancement increases with increasing the Dean Number (De) and volumetric concentrations. The present study shows that these double pipe helical coil heat exchanger have advantages by using nanofluids.

Keywords: *Nanofluid, CFD, Dean Number, Nusselt number, Heat transfer coefficient*

1. Introduction

Due to limitations of energy resources and rising demand on energy, proficient use of energy has become an issue of vast need for humanity. Also, with the help of new technologies, we can effectively increase the efficiency of an industry. In an industry heat exchangers are widely used and performance of heat exchangers can be improved by different methods. Due to the compact structure and high heat transfer coefficient, helical coil heat exchangers find wide use in industrial applications such as power generation, nuclear industry, process plants, heat recovery systems, food industry, refrigeration, etc.

Abdulla et al. [1]; studied that there is a need for miniaturization of the heat exchanger in industries where space is not enough. Nanofluid technology uses the new grade of heat transfer fluids with 1-100nm sized suspended nanoparticles in base Fluid is known as nanofluid. The conventional HTF such as water, engine oil and ethylene glycol have poor heat transfer capability, so we need a different HTF, nanofluid has fast heating and cooling capability. Therefore, an enhancement in heat exchanger efficiency through this technique may result in a considerable saving of energy, material and total cost by modifying fluids property. Wide range of literature has been found to improve the heat transfer rate by using helical coil heat Exchanger [2, 3]. Seban and McLaughlin et al. [4] studied the heat transfer in coiled tubes for both laminar and turbulent flows experimentally. It was analyzed that the coefficients for turbulent flow depend on the coil diameter. Prabhanjan et al. [5] compared the heat transfer coefficients of a straight tube heat exchanger to that of a helically coiled heat exchanger for heating liquids and found that it was affected by the geometry of heat exchanger.

Abbreviation

Ac= flow area, m2	q′ =total surface heat flux
Al_2O_3 =aluminium oxide	Q =heat transfer rate (kW)
As =surface area, m2	*Re* =Reynolds Number
Cu =copper	*Recr*=Critical Reynolds number
CuO =copper oxide	*St*= Stanton Number
Cp =specific heat, J/kg-K	*Thi*=temperature of hot fluid at inlet, K
CFD =computational fluid dynamics	*Tho*=temperature of hot fluid at outlet, K
dii=inner diameter of inner pipe, inch	*Tci*= temperature of cold fluid at inlet, K
dio=outer diameter of inner pipe inch	*Tco*= temperature of cold fluid at outlet, K
doi =inner diameter of outer pipe, inch	Tw= wall temperature, K
doo =outer diameter of outer pipe, inch	Tf= fluid mean temperature, K
Dh = hydraulic diameter, mm	
Dn =diameter of nanofluid particles	*ux*, *uy* and *uz*=velocity in x ,y, z directions
D = coil diameter, mm	*x*, *y*, *z* coordinates
De =Dean Number	*X*, *Y*, *Z* body force in x, y, z directions
HTF =Heat transfer fluid	*V= Average velocity of flow*
h =heat transfer coefficient, W/m2k	*Vs* = wetted volume, m3
H =pitch of coil, mm	
k =thermal conductivity, W/m-K	**Greek symbol**
K =turbulent kinetic energy	β=surface area density, m2/m3
L =length of pipe, m	ρ =density, Kg/m3
m =mass flow rate, kg/s	μ=dynamic viscosity, Kg/ms
n= number of turns	Φ=Rayleigh dissipation factor
Nux =Local Nusselt Number	ε =dissipation kinetic energy (m^2/s^3)
Nu=Average Nusselt number	φ =volume fraction (%)
P= Pressure, N/m2	δ= Curvature Ratio
Δp= Pressure drop	
Pr= Prandtl Number	
q= heat flux, W/m2	

Kumar et al. [6] studied numerically and experimentally the friction factor and heat transfer coefficient for fluid in a tube-in-tube helically coiled heat exchanger. Heat transfer in helical coiled heat exchangers was experimentally and numerically investigated by Jaya kumar et al. [7]. It was found for calculation of heat transfer in a situation of fluid-to-fluid heat transfer, random boundary conditions for example constant wall temperature and constant heat flux, are not appropriate. Naphon [8] compared the thermal efficiency and pressure drop of the helical-coil heat exchanger with and without helical undulating fins. The results showed that with increasing Reynolds number, the friction factor decreases.Wongwises et al. [9] experimentally investigated the condensation heat transfer in a tube-in-tube helical heat exchanger. Also there is an extensive research had been done on the performance of other types of heat exchangers.[10-12]

In this work, the heat transfer performance of a double helical coiled heat exchanger was investigated numerically and compared with Soumya Ranjan Mohanty [13]. The end of the tubes act as the entry and exit of hot as well as cold fluid where the cold fluid flows in the outer pipe and the hot fluid flows in the inner pipe of the double pipe helical coil heat exchanger.

In double tube helical heat exchangers, due to the curvature of tubes and application of centrifugal force on fluid flow, the secondary flow motion is generated which improves heat transfer coefficient substantially [14]. Dravid et al. [15]

proposed the correlation for inner Nu at the Dean Number (De) range of 50 to 2000. He investigated the secondary flow motion induced by the curvature effects. They analyzed the resultant centrifugal force makes heat transfer coefficient greater than that of straight tube.

The objective of this work was to obtain the effects of several geometrical parameters on heat transfer characteristics for water and the effect of SiO2 –water nanofluid with different volume fractions (1-4%) on heat transfer characteristics.

2. Methodology:

2.1 Characteristics of helical coils:

One of the key points of helical heat exchanger is the stretch of the flow through the coils, which produces the centrifugal forces that leads to secondary flow.

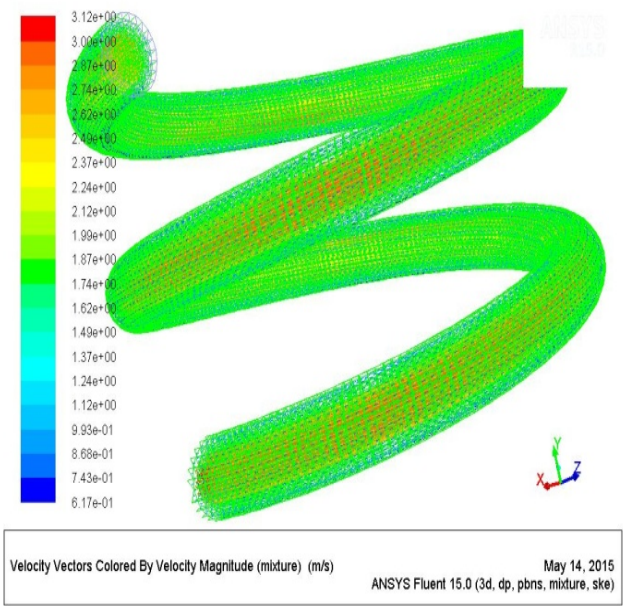

Fig. (1) Velocity vector of double pipe helical coil heat exchanger.

The figure (1) shows that maximum velocity occurred at the centre of inner pipe because this section has maximum distance from boundary layer. For the analysis of flow in helical pipe, Dean (De) number is used which is defined as follows,

$$De_{in} = \left(\frac{\rho V d_{in}}{\mu}\right)\left(\frac{d_{oi}}{D}\right)^{1/2}$$

As double pipe helical heat exchangers are used in turbulent flow, thus in this work, turbulent flow ranges were chosen for flow rates. Therefore, to know about the type of flow, the obtained Reynolds number can be compared with critical number:

$$\mathrm{Re}_{cr} = 2300[1 + 8.6(d/D)^{0.45}]$$

Where δ is curvature ratio and defined as:

$$\delta = \left(\frac{d_{oi}}{D}\right)$$

2.2 Problem description:

The double pipe helical coil heat exchanger model used in this analysis is the experimental model presented by Soumya Ranjan Mohanty [13] (figure 2). It consists of two pipes in which 348^{0}C water in first case and SiO_2-water nano fluid in second case taken as hot fluid, flows through inner pipe and 283^{0}C water taken as cold fluid, flows in outer pipe in the same direction with hot fluid. The end of pipes acts as the entry and exit of hot as well as cold fluid. The pipes are made of copper to maximize the heat transfer. Tables 1, 2, and 3 shows the dimensional parameters, properties of base fluid, nanoparticle and copper respectively.

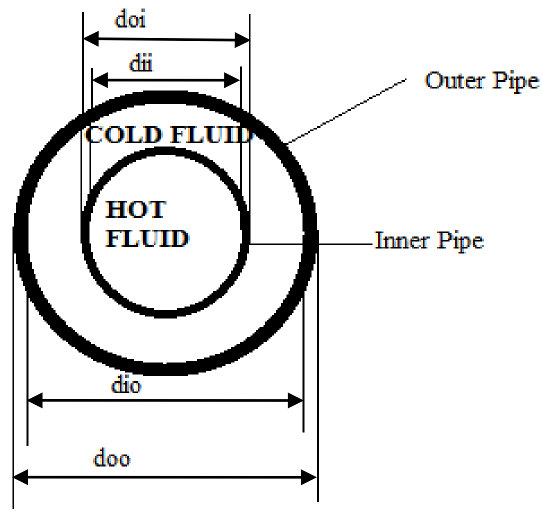

Fig. (2) Schematic of cross section of used model

Table 1: The dimensional parameters of inner and outer pipe

Diameter	d_{ii}	d_{oi}	d_{io}	d_{oo}
Value (inch)	0.545	0.625	0.785	0.875

Table 2: The thermo - physical properties of base fluid and nanoparticle at T = 300k.

Thermo-physical Properties	Water	SiO_2
Density, ρ (Kg/m^3)	996.5	2220
Specific Heat, C_p (J/Kg K)	4181	495.2
Thermal Conductivity, K (W/m K)	0.0103	13
Viscosity, μ (mpa s)	0.001003	-

Table 3: Properties of copper

Description	*Value*	*Unit*
Density	***8978***	***Kg/m^3***
Specific Heat Capacity	***381***	***J/Kg-K***
Thermal Conductivity	***387.6***	***W/m-K***

2.3 CFD Approach

In this study, the geometry of double pipe helical coil was created using commercial software (ANSYS 15). The meshing was done using MAPPED FACE MESH for fine mesh with Tetra and Hexahedral cells. A numerical analysis was done to understand the flow characteristics. Standard (k-ε) k-epsilon model was used as a turbulent model. For nanofluid, a mixture model was used. Velocity and pressure are given at inlet and outlet respectively for insulated outer wall condition. The input parameters were indirectly taken from the Dean number. The second order upwind scheme was used to solve governing equations. The coupled scheme was used to deal with pressure-velocity coupling problem. The relaxation factor has been set to default values. The normalized residual values are set to 10^{-6} for all the variables to converge the solution.

2.3.1 Governing Equations:

The governing differential equations for the fluid flow are given by Continuity equation, Navier Stokes equation and energy conservation equation. These can be written in the following form [16]:

2.3.1.1 Continuity Equation:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \qquad \textbf{(2.1)}$$

2.3.1.2 Navier Stokes equation:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=\rho X-\frac{\partial p}{\partial x}+\frac{1}{3}\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 u$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=\rho Y-\frac{\partial p}{\partial y}+\frac{1}{3}\mu\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 v$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=\rho Z-\frac{\partial p}{\partial z}+\frac{1}{3}\mu\frac{\partial}{\partial z}\left(\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right)+\mu\nabla^2 w \qquad \textbf{(2.2)}$$

2.3.1.3 Energy Equation:

$$\rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right)=\left(u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y}+w\frac{\partial p}{\partial z}\right)+k\nabla^2 T+\mu\phi \qquad \textbf{(2.3)}$$

Where,

$$\phi=2\left[\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right]+\left[\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2\right]$$
$$-\frac{2}{3}\left[\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right]^2$$

2.3.1.4 Turbulent kinetic energy (k) equation:

$$\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]+G_k-\rho\varepsilon \qquad \textbf{(2.4)}$$

2.3.1.5 Turbulent kinetic energy dissipation (ε) equation:

$$\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left[\left(\mu+\frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial\varepsilon}{\partial x_j}\right]+C_{1\varepsilon}\left(\varepsilon/k\right)G_k+C_{2\varepsilon}\rho\left(\varepsilon^2/k\right) \qquad \textbf{(2.5)}$$

In the above equations, G_k represents the generation of turbulent kinetic energy due to mean velocity gradients, σ_k and σ_ε are effective Prandtl numbers for turbulent kinetic energy and rate of dissipation, respectively; $C1_\varepsilon$ and $C2_\varepsilon$ are constants and μ_t is the eddy viscosity and is modeled as

$$\mu_t=(\rho C_\mu k^2)/\varepsilon \qquad \textbf{(2.6)}$$

The empirical constants for the turbulence model are arrived by comprehensive data fitting for a wide range of turbulent flow

$$C_\mu=0.09,\ C_{\varepsilon 1}=1.47, C_{\varepsilon 2}=1.92,\ \ \sigma_k=1.0, \sigma_\varepsilon=1.3 \qquad \textbf{(2.7)}$$

The governing differential equation for solid domain is only the Energy equation which is given by;

$$\nabla^2 T=0 \qquad \textbf{(2.8)}$$

Heat transfer coefficient is obtained by equating the conduction heat transfer to the convection heat transfer;

$$q_{cond} = q_{conv}$$

$$h = \frac{-k\dfrac{\partial T}{\partial x}}{T_w - T_f} \tag{2.9}$$

Local Nusselt number is given by;

$$Nu_X = hD/k \tag{2.10}$$

2.3.2 Boundary Conditions:

In the present study, the inlet boundary conditions imposed at inner and outer pipe are defined as velocity inlet as the entrance velocities of flow are known. Outlet boundary condition is defined as pressure outlet. The flow is assumed Newtonian, turbulent and incompressible. In double pipe helical heat exchanger, there are two walls. Insulated outer wall condition is used for outer tube wall and bilateral wall is used for inner pipe for the heat transfer to be occurred on both sides.

3. Result and discussion:

Secondary flow plays an important role for the heat transfer to be occurred in helical heat exchanger. In this work, the effect of several geometrical parameters, different nanoparticles, and different volume fraction on heat transfer characteristics were studied.

3.1 The effects of several geometrical parameters on heat transfer characteristics:

3.1.1 The effect of coil diameter:

The figure (3) illustrates the effect of coil diameter on Nusselt number. It is clear that Nusselt number decreases with increasing coil diameter. Because by increasing the coil diameter, the effect of secondary flow diminishes. And fluid behaves like a flow in a straight pipe.

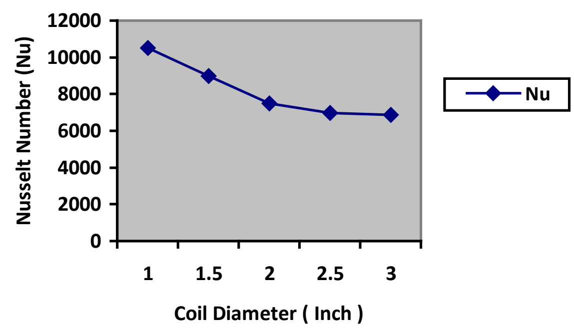

Fig. (3) Variation of Nu with respect to coil diameter.

3.1.2 Influence of pitch of coil:

The figure (4) illustrates the variation of pitch of coil on heat transfer coefficient. By increasing the pitch of coil as 1, 1.5 and 2 inch, the fluid's torsion behavior diminishes and this results in decreasing heat transfer coefficient.

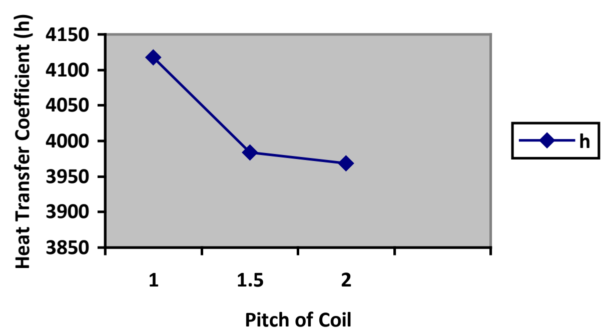

Fig. (4) Variation in heat transfer coefficient with respect to pitch of coil.

3.2 The effect of nanoparticle with different volume fractions on heat transfer characteristics:

3.2.1 Variation of Dean Number with Nusselt Number for water and SiO_2 –water nanofluid:

In this section, SiO_2 nano particle with water as a base fluid is used. At 4 % nano particle volume fraction with 20 nm particle size, the graph is plotted between Nusselt number and Dean Number. The figure (5) indicates that increasing of the Dean Number results in increasing Nusselt number. It is clearly seen that siO_2 – water nanofluid possess higher Nusselt number compared to pure water.

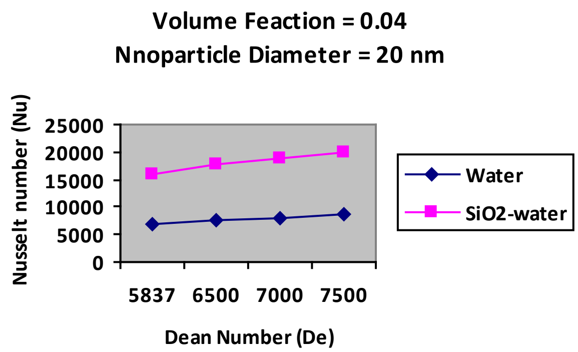

Fig.(5) Variation of the effect of SiO_2-water nanofluid under Dean Number range on the average Nusselt number.

3.2.2 The effect of volume fraction:

In this section, the graph is plotted between Nusselt Number and Dean Number at different volume fraction. It can be seen that Nusselt number enhancement is achieved by adding low volume fraction of nanoparticle (0.01-0.04) with 20 nm particle size of SiO_2 to the water as base fluid. Because the presence of nanoparticle provides the larger surface area for molecular collisions and higher momentum is achieved by increasing concentration of nanoparticle. This momentum carries and transfer thermal energy more efficiently and enhances the heat absorption of coolant causing its temperature to increase. It is clear that the Nusselt number increases with increased value of volume fraction.

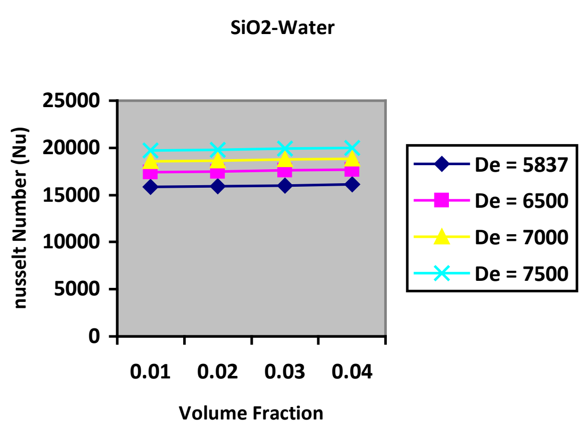

Fig. (6) The effect of different volume fraction under the Dean number range on the average Nusselt Number

4. Conclusion:

Numerical simulation with ANSYS FLUENT 15 has been carried out for double pipe helical coil heat exchanger subjected to insulated outer wall condition. Nusselt number and heat transfer coefficient are computed. The tube side fluid is SiO_2-water nanofluid and the flow condition is turbulent. Following are the outcomes of the above numerical study;

- In the central region of inner pipe, the velocity is found to be maximum due to the maximum distance from boundary layer.
- It is observed that heat transfer coefficient decreases with increasing diameter and pitch of coil. Because by increasing these parameters, the effect of secondary flow diminishes and fluid behaves like a flow in a straight pipe**.**
- For convective heat transfer, increase in Dean Number results in increased heat transfer coefficient. Because at increased flow rates, the dispersion effects and random movement of the nanoparticle enhances the mixing fluctuations and results in increased heat transfer coefficient.
- Higher heat transfer coefficient is achieved by nanofluid containing small amount of nanoparticles compared to the base fluid and it increases with increasing particle volume fraction. Because by increasing nanoparticle volume fraction, the interaction and collision of nanoparticle intensifies and a rapid heat transfer from wall to nanofluid occurs due to relative movement of these particles.

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