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NUMERICAL AND EXPERIMENTAL STUDY THE EFFECT OF (SiO_2) NANOPARTICLES ON THE PERFORMANCE OF DOUBLE PIPE HEAT EXCHANGER

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Abstract: Numerical and experimental investigation of the heat transfer and the friction factor characteristics in a counter flow double pipe heat exchanger with and without Nano fluid was studied**.** Tests are performed for fully developed turbulent flow ranges, homogeneous Nano fluid. The double pipe heat exchanger consisting of two tubes, the first inner plain tube which has been manufactured from copper materials with inner and outer diameters of (11.2 and 12.7) mm respectively, the length of first inner plain tube is 1057 mm. The second insulated an external tube which has been manufactured from PVC material with inner and outer diameters of (44.4 and 50)mm respectively, with length is 1032mm. The flows has been tested at Reynolds number ranging (3019.43-4824.22). Nano particle size ranging (15-20)nm. The results show that the heat transfer rate increases as Reynolds number and volume concentration of Nano fluid increase. The increasing percentage of Nusselt number, friction facto and performance factor of (15.72,11.51 and 11.57) % respectively for maximum volume concentration and volume flow rate of 3% and 1.6 lpm respectively. ANSYS FLUENT 2015 package used to simulate the heat transfer and the fluid flow in the heat exchanger. The results show that the heat transfer increases when Nano fluid was added. The agreement observed with experimental work with maximum discrepancy 12%.

Keywords: *Investigation, Nanoparticles, SiO_2, performance, heat transfer, double pipe, heat exchanger.*

الدراسة العملية والنظرية لتأثير الجزيئات النانوية لثنائي اوكسيد السيليكون (SiO_2) على أداء المبادل الحراري

الخلاصة: تمت الدراسة النظرية والعملية لانتقال الحرارة ومعامل الاحتكاك في مبادل مزدوج الانبوب مع أو بدون المائع النانوي .الأختبارات تمت في مدى الجريان المضطرب وثبوت الخواص والمائع النانوي متجانس. المبادل الحراري مزدوج الانبوب يتألف من انبوبين متمركزين داخليا الانبوب الداخلي تم صناعته من النحاس وبقطر داخلي وخارجيmm (11.2 and 12.7) على التوالي وطول (1057mm) ، أما الانبوب الخارجي معزول وتمت صناعته من مادة PVC وبقطر داخلي وخارجي mm(44.4 and 50) على التوالي وطول 1032 mm في هذه الدراسة تم تحسين انتقال الحرارة في مبادل مزدوج الانبوب باستخدام ثنائي اوكسيد السيليكون كجزيئات نانوية في الماء المقطرمع عدد رينولدز يتراوح بين (3019.43-4824.22) ويتراوح حجم جزيئات النانوnm(15-20) . أوضحت النتائج أن معدل أنتقال الحرارة يزداد مع زيادة كلا من عدد رينولدز والتركيز الحجمي في المائع الاساس . النسبة المئوية للزيادة لكلا من عدد نسلت، معامل الاحتكاك، ومعامل الاداء للمبادل الحراري مزدوج الانبوب %(15.72, 11.51and 11.57) على التوالي عند تركيز 3% وتدفقlpm 1.6 . الدراسة النظرية تمت بأستخدام ANSYS FLUENT 2015 package استخدم لنمذجة المبادل الحراري ودراسة انتقال الحرارة وسلوك المائع . أوضحت النتائج زيادة انتقال الحرارة عند اضافة المائع النانوي لزيادة الخواص الفيزيائية للمائع الاساس، هناك توافق مع الجانب العملي مع اعلى نسبة انحراف 12%.

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1. Introduction

The thermal devices are important and used in many applications such as cooling systems, food industries, transportation, medical applications, etc. Therefore, research and development activities are carried out to improve the heat transfer process to reduce the energy losses and reduce the volume of thermal devices، hence saving in energy, reduce process time, raise thermal rating and lengthen the working life of equipment[1].

Senthilraja et al (2014)[2] studied experimentally the heat transfer coefficient of CuO /water Nano fluid flowing in a horizontal double pipe counter flow heat exchanger in the presence and absence of electric field. The CuO nanoparticles are approximated 27nm were utilized in the heat exchange, happens between a CuO /water nano fluid circling internal tube and a hot air stream coursing through the external tube. The electric field was provided reporting in real time side of the heat exchanger. Its voltage was shifted from 0kV to 6kV،the range of Reynolds number 3000-20000,mass flow rate of high temperature water 3l lpm . The experimental results indicated that the convective heat transfer coefficient of the nano fluid increased up to the volume fraction of 0.15%، also the heat transfer rate was enhanced while applying the high voltage supply to the electrode. The convective heat transfer coefficient of the nano fluid was increased with increasing the electric field intensity and nano fluid volume concentration.

Ebrahimi et al. (2014)[3] investigated experimentally the forced convection heat transfer in auto radiator (compact heat exchanger). The effect of adding SiO_2 nano particles to base fluid water to enhancing thermal efficiency of engine prompts،increment the engine's performance. Studied effects of fluid inlet temperature (43, 52, 60) and the flow rate، the accuracy of 0.2 lpm, nanoparticles concentration 0.04%. Results showed that using nano fluid as working fluid prompts to higher heat transfer performance which is propelled the auto engine execution, would decrease fuel utilization, diminish the pollution emissions, Nusslet number increases with increase of liquid inlet temperature, nano particle volume fraction and Reynolds number.

Raei et al (2017)[4] investigated experimentally overall heat transfer coefficient and friction factor of water based γ -Al_2O_3 nano fluid in a double tube, counter flow heat exchanger under turbulent flow regime. At the concentrations of 0.05 and 0.15 vol. % with variation of flow rates in the range of 7–9 lpm. Nano fluid enters the inner tube of the heat exchanger at different temperatures including 45, 55, and 65 °C. Results showed that increasing the nano fluid flow rate, concentration and inlet temperature can improve the overall heat transfer coefficient and heat transfer rate. Also, the ratio of the overall heat transfer coefficient of nano fluid to that of pure water decreased with increasing the nano fluid flow rate, the friction factor of nano fluid increased with increasing the volume fraction of nano particles.

Mohammed et al (2013)[5] studied numerically the effect of using louvered strip inserts placed in a circular double pipe heat exchanger on the thermal and flow fields utilizing various types of nano fluids with a uniform heat flux boundary condition two distinctive louvered strip embed arrangements (forward and in reverse), are utilized as a part in this study with a Reynolds number range of 10,000 to 50,000. The effects of various louvered strip slant angles and pitches are also investigated. Four different types of nano particles, Al_2O_3, CuO, SiO_2, and ZnO with different volume fractions in the range of 1% to 4% and different nano particle diameters in the range of 20 nm to 50 nm, dispersed in a base fluid (water) are used.

The continuity, momentum and energy equations are solved by means of a finite volume method. The numerical results showed that the forward louvered strip course of action can advance the heat transfer by roughly 367% to 411% at the highest slant angleα = 30, and least pitch of S = 30 mm. The maximal skin friction of the improved tube is around 10 times than that of the smooth tube, and the estimation of performance assessment criterion lies in the range of 1.28– 1.56. It is discovered that SiO_2 nano fluid has the most noteworthy Nusselt number value, trailed by Al_2O_3, ZnO, and CuO Nusselt number increments with diminishing the nano particles diameter and it increments somewhat with increasing the volume concentration of nano particles.

Yarmand etal 2014[6] investigated numerically heat transfer enhancement in a rectangular heated pipe, for turbulent nano fluid and uniform heat flux. The symmetrical rectangular channel at the top and bottom at constant heat flux, while the sides wall are insulated four different types of nano particals Al_2O_3, ZnO, CuO and SiO_2 at different volume fractions of nano fluids in the range of 1% to 5%, Reynolds number in range 5000-25000. The numerical results showed that SiO_2-water has the most noteworthy Nusselt number contrasted with other nano fluid, while it has the least heat transfer coefficient because of low thermal conductivity, the Nusselt number increments with the increase of Reynolds number.

Kumar et al (2014)[7] studied Numerically the concentric of dimpled tube with Al_2O_3 nano fluids. The inner tube specifications of the concentric tube heat exchanger, with ellipsoidal and spherical dimple geometries. The results showed that the Nusselt number for the spherical dimpled tube and ellipsoidal dimpled tube are 35.7% and 63.59% higher than that for dimpled tube, increase by 52.7% and 87.27% for ellipsoidal and spherical dimples compared with the smooth tube with Al_2O_3 nano fluid. The heat transfer rate for ellipsoidal dimpled better than spherical and plain tube.

Rahmah (2015)[8] investigated experimentally and numerically the enhancement of heat transfer characteristics of distilled water, 100% ethylene glycol, distilled water 50% - ethylene glycol 50% and metal oxide nano fluid type Al_2O_3 with distilled water at concentrations of φ = 0.1, 1, 3% by volume in a double-pipe heat exchanger counter flow have been studied،the used inlet temperature of hot fluid are (40, 50, 60 and 70°C) with flow rates of the fluid (1.1, 1.3 and 1.5 lpm) while the temperature of fluid in annulus is the atmospheric temperature (23-33°C) with flow rate (10.75lpm). Result showed the best working fluid for practical applications is the Al_2O_3 which gives (1.625) enhancement، for concentration(3 %). The Nusselt number increases with increasing concentration (0.1, 1and 3%), the change in fluid properties increases the values of Nusselt number. Ethylene glycol gave highest Nusselt number among the other working fluids due to the high Prandtle number, the mixture of ethylene glycol and water has Nusselt number lower than for the ethylene glycol alone, the numerical study includes using of ANSYS FLUENT package 15**,** good agreement between experimental and theoretical work.

Samar mad (2015)[9] investigated experimentally and numerically the convective heat transfer enhancement in a spiral fluted tube heat exchanger with and without twisted tape insert. The performance of the heat exchanger with and without nano fluid was also studied, constant wall temperature condition (110˚C) was achieved, Reynolds numbers ranging from (7000 to 15000) flows through the inner tube, the tests were conducted by using a twisted tape of constant twist ratio (y/w = 4). Titanium Oxide (TiO_2) nanoparticle powder with 50 nm diameter was dispersed in distilled water with different volume concentrations 0.08, 0.1, 0.2, and 0.3 % to the experimental results showed an increase in the convective heat transfer coefficient by increasing both volume concentration of nano particle and Reynolds number.

When using spiral fluted tube at 0.3% volume concentration (27-31) %. The heat transfer coefficient increased higher than the pure water in spiral fluted tube. Also, when using spiral fluted tube equipped with twisted tape inserted at 0.3% volume concentration (28-32). Numerical simulation was carried using ANSYS 14, FLUENT package. Steady state, Newtonian flow, incompressible and three dimensional when compared the experimental and numerical, good agreements were obtained with a maximum deviation of 12%.

The objective of the present work to study the effect of using silicon dioxide Nanoparticles on enhancement the heat transfer in double pipe heat exchanger with different volume concentration (0.1, 1 and 3)%.

2. Experimental Apparatus and Procedure

2.1 Test Section Regime

1. **The inner pipe :** its length tube of 1057 mm . The internal tube is concentric in the external tube . The inner tube manufacture from copper with internal and external diameters of (11.2 and 12.7) mm respectively . The hot fluid (water only or Nano fluid) flow in inner tube. The thermocouples fixed along the test section at various position on surface, on inner tube the thermocouples fixed by thermal Teflon material to measuring the temperature of tube side at each position of thermocouple location. The pressure drop on both ends of heat exchanger are measured by using differential manometer .Also at the inlet and outlet pressure of the heat exchanger is measured by using pressure gauge**.**
2. **The outer tube:** its length of 1032mm, external tube made from PVC with the inner and outer diameters of (44.4 and 50) mm respectively. The outer insulated by using Polyurethane with thermal conductivity0.02 W/m. k to anticipate heat dispersal to the surrounding.The cooled fluid flow in annulus side as shown in figure 1 schematic of expermental work,figure2 shows the model of the double pipe heat exchanger and figure 3 shows the model of the double pipe heat exchanger cross section.

2.2 Description of Experimental Test Rig

The schematic diagram of the experimental test rig is shown in Figure 1. The experimental test rig was carried out the double pipe heat exchangers and the accessories as shown below:

1. Test rig (double pipe heat exchanger

2. Hot fluid supply system: The hot fluid supply system consists of tank with (3) litters capacity، made from galvanized steel with thickness (1mm) .Also the tank insulated by fiber glass with thermal conductivity of (0.04W/m .k), pump (Marcus type, control utilization 120W,power supply220V/50HZ, most extreme limit 12 lpm, greatest head 12m, RPM 2850), flow meter used to masure the hot liquid in the range from 25to 250 lph. control valve and heater with 1500 W.
3. Cold water supply system: The cold water supply system consist of tank with 16 litters capacity،made from galvanized steel with thickness (1mm) protected by fiber glass insulation with thermal conductivity of (0.04W/m .k), control valve, flow meter used to measure the cold water in the range from 4 to 16 lpm, pump (Marcus, control utilization 60W, control supply 220V/50 HZ, most extreme limit20 lpm maximum head 10m, RPM2850) The cold water flow in annulus side to reduce the high temperature of the water that flow in inner tube and obtained height heat transfer.
4. Chilled water unit: The cold water storage tank associated with chilled unit capacity (1ton) to keep up the water temperature inside it roughly 25°C and it's comprises of the following parts: Condenser, evaporator, compressor, solenoid, fan, capillary tube (extension valve).accumulatorand control board.
5. Control board: Control board consist of the green bottom that used to open the cooler Two switches on the two sides of the board used to open the pump for circling the hot and cold fluid. Temperature controls to control hot and cold fluid inlet temperature .

2.3 Measuring Devices

1. Thermocouple: use to mesure the temperature in various position thermocouple (type – k) use with ranges from (-270 to 370)°C. Nine thermocouples are utilized and fixed in the following position: Five thermo couples fixed along the test at various position on surface of inner tube fixed by (thermal Teflon material) to measure the temperature of tube side at each position of thermocouple location. Two thermo couple fixed at inlet and outlet annulus side to measure the temperature inlet and outlet for cold water. Two thermo couple fixed at inlet and outlet of the tube side to measure inlet and outlet hot fluid temperature at tube side.
2. Temperature recorder: the temperature data logger model BTM-4208SD, 12 channels, with SD card, and range of channels type k (-100 to 1300ºC) .Data logger used to measure the temperature of fluid, surface for all position, inlet and outlet temperature for both fluid.
3. Flow meter: Two flow meters device for hot fluid and cold fluid used to measure the waterflow rate in test rig with range (4-16 lpm)for cold water in annulus side and rang (25- 250 lphr.) for hot fluid in tube side.
4. Pressure manometer and Pressure gage: differential Manometer (2000mbar*1mbar)model Lutron PM-9100 range model is used to measure the pressure difference along heat exchanger.Two pressure port position fixed at inlet and outlet also pressure gage within range (0-10)bar are used

2.4 Uncertainty of Experimental Work

The experimental work investigation errors are based upon the accuracy of measurement apparatus, the uncertainty for current work are calculated for Nusselt number and Reynolds number for tube side [20]

$$Re = \frac{4\dot{m}_h}{\pi\mu d_i} = \frac{4\rho\dot{V}_h}{\pi\mu d_i}$$

$$\Delta Re = [[\frac{\sigma}{\sigma\dot{V}_h}[Re].\Delta\,\dot{V}]^2 + [\frac{\sigma(Re)}{\sigma d_i}.\Delta di]^2 \text{ ، } \frac{\sigma(Re)}{\sigma\dot{V}_h} = \frac{4}{\pi\mu d_i} \text{ and } \frac{\sigma(Re)}{\sigma d_i} = \frac{4\rho\dot{V}_h}{\pi\mu}$$

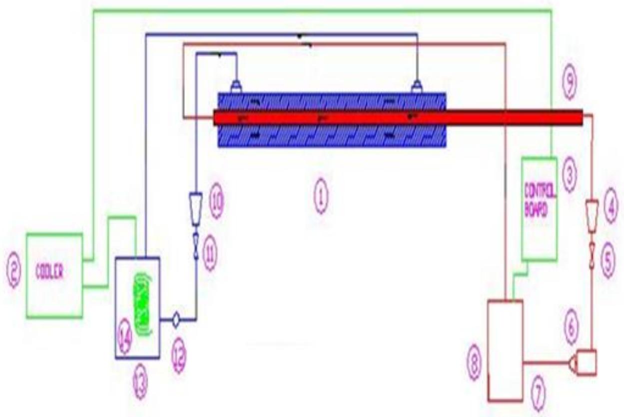
$$\frac{\Delta Re}{Re} = [[\left[\frac{\frac{4}{\pi\mu d_i}}{\frac{4\rho\dot{V}_h}{\pi\mu d_i}}\right].\Delta\,\dot{V}]^2 + \left[\frac{\frac{4\rho\dot{V}_h}{\pi\mu}}{\frac{4\rho\dot{V}_h}{\pi\mu}}\right]\Delta di]^2]^{0.5}$$

$$\frac{\Delta Re}{Re} = [[\frac{\Delta\dot{V}}{\dot{V}_h}]^2 + [\frac{\Delta d_i}{d_i}]^2]^{0.5} = [[0.0234]^2 + [\frac{7\times10^{-4}}{11.2\times10^{-3}}]^2]^{0.5}$$

=0.0665

$$\frac{\Delta Nu}{Nu} = [[\frac{\Delta h_i}{h_i}]^2 + [\frac{\Delta d_i}{d_i}]^2]^{0.5} \text{، } \frac{\Delta Nu}{Nu} = [[7.01923\times10^{-7}]^2 + [\frac{7\times10^{-7}}{11.2\times10^{-3}}]^2]^{0.5}$$

=0.0625



1. Test section
2. Cooler
3. Control bord
4. Hot water flow meter
5. Valve
6. HotWater pump
7. Hot Water tank
8. Heater
9. Fully developed region
10. Cold water flow mete
11. Valve
12. Cold water pump
13. Coldwater tank
14. Evaporater

Figure1. Schematic of expermental work.

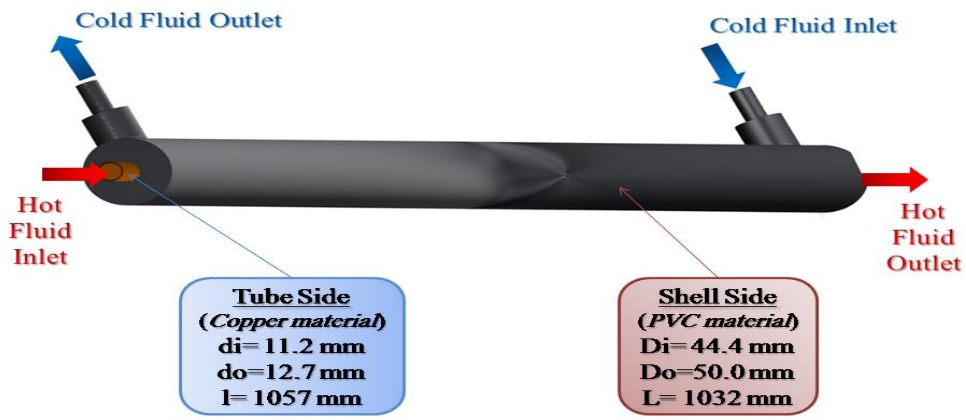


Figure2. Model of the double pipe heat exchanger.

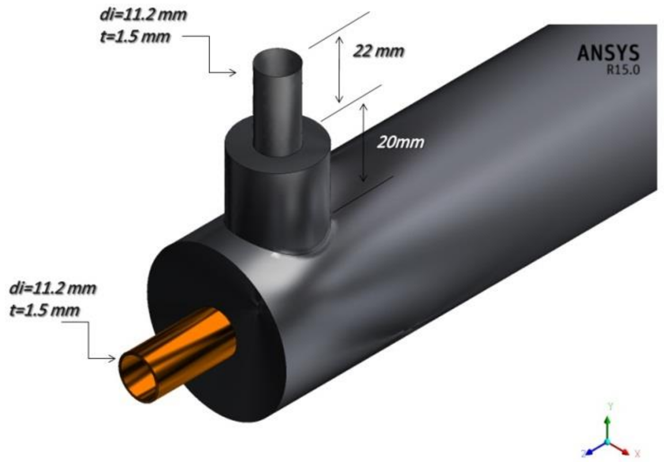


Figure 3. Model of the double pipe heat exchanger cross section.

2.5 Procedure of the Experimental Data

To estimate the performance of the double pipe heat exchanger, the following operation conditions were used : the flow rate for hot fluid of (1, 1.2, 1.4 and 1.6) lpm, the inlet temperatures of (45, 55, and 65)ºC and cold fluid in annular side is 25ºC with 4lpm flow rate.

a. Preparation the artificial nano fluid with all concentration (0.1, 1and 3)% volume concentration.
b. Running the chilled water unit
c. The pump of cooled water open by the switch in the control board .
d. After the cooled unit is running, during this time the heater remains turn off.
e. Reduce the velocity for cooled water until settled flow rate 4L/ min .
f. The heater turn on and pump of hot water opened by using the switches in the control board to circulate water or Nano fluid with high temperature in the internal cycle. Adjust the fluid flow by control valve until the flow rate achieve the required Reynolds number.

g. The procedure becomes steady state after 30-40 minute when $\frac{\partial T}{\partial t} = 0$,
h. Record the inlet and outlet temperatures for cold and hot water. Also the surface temperature Ts in the test section at all the thermocouple location were recorded.
i. Record the pressure difference on both ends of heat exchanger, also at inlet and outlet of heat exchanger the pressure were recorded .

2.6 Performance Variables in Double Pipe Heat Exchanger

In this study, two fluids flow in the double pipe heat exchanger, the hot fluid (with and without nanoparticles) flow inside inner tube and cold fluid flow in annular side .The operational conditions assumed that، fully developed flow،turbulent regime with constant wall temperature . The heat transfer rate for nano fluid through tube side (hot fluid) [10], [11].

$$\mathrm{Q}_h = \dot{\mathrm{m}}_\mathrm{h}\, \mathrm{C_p}_h (\mathrm{T}_{ho} - T_{hi}) \tag{1}$$

The heat transfer rate for water flow through annular side (cold water):

$$Q_C = \dot{\mathrm{m}}_c C_{PC} (Tc_o - Tc_i) \tag{2}$$

The Logarithmic Mean Temperature Difference (ΔT_{lm}):

$$\Delta T_{lm} = \frac{(\mathrm{T_{h1}} - \mathrm{T_{c2}}) - (\mathrm{T_{h2}} - \mathrm{T_{c1}})}{\ln \frac{(\mathrm{T_{h1}} - \mathrm{Tc_2})}{(\mathrm{T_{h2}} - \mathrm{T_{C1}})}} \tag{3}$$

Average heat transfer rate:

$$\mathrm{Q}_{ave} = \frac{\mathrm{Q}_h + \mathrm{Q}_c}{2} \tag{4}$$

Overall heat transfer coefficient (U):

$$\mathrm{U} = \frac{\mathrm{Q_{ave}}}{A_s \Delta T_{lm}} \tag{5}$$

Reynolds number for the annular side:

$$Re = \frac{\rho u_a D_h}{\mu} \tag{6}$$

Where Dh =Di-do Nusselt number for the annular side calculated by using Dittus – Boelter correlation:

$$\mathrm{Nu} = \frac{ho \times D_h}{k} = 0.023\, Re^{0.8} pr^{\,0.3} \tag{7}$$

Heat transfer coefficient for the tube side
The heat transfer coefficient for tube side can be calculated as following equation:

$$U_O = \frac{1}{\frac{1}{h_o} + \frac{d_o ln\frac{d_o}{d_i}}{2k} + \frac{d_o}{h_i d_i}} \tag{8}$$

So, can be calculated the tube side Nusslet number from this simple equation:

$$Nu = \frac{h_i d_i}{k_i} \tag{9}$$

Friction factor calculated:

$$f = \frac{\Delta p}{(\frac{L}{D})(\frac{\rho u^2}{2})} \tag{10}$$

The performance of heat exchanger (η) [12]:

$$\eta = \frac{Nu_{nf}/Nu_{pt}}{(f_{nf}/f_{pt})^{1/3}} \tag{11}$$

Thermo physical properties of Nano fluid
Thermal conductivity:
Patel et al (2010) [13] model

$$\frac{\mathrm{keff}}{\mathrm{kbf}} = (1 + 0.135(\frac{\mathrm{kp}}{\mathrm{kbf}})^{0.273}\emptyset^{0.467}\left(\frac{\mathrm{T}}{20}\right)^{0.547}(\frac{100}{\mathrm{dp}})^{0.234} \tag{12}$$

Density:
Ho et al (2010) [14]

$$\rho\mathrm{nf} = 1001.064 + 2738.6191\emptyset - 0.2095T \tag{13}$$

Specific heat:
palm et al (2006)[15]

$$(\mathrm{cp})\mathrm{nf} = (1-\emptyset)(\mathrm{cp})\mathrm{f} + \emptyset(\mathrm{cp}) \tag{14}$$

Viscosity:
Sekher et al (2015)[16]

$$\mu\mathrm{r} = 0.935(1+\frac{\mathrm{Tnf}}{70})^{0.5602}(1 + \frac{\mathrm{dp}}{80})^{-0.05915}(1+\frac{\emptyset}{100})^{10.51} \tag{15}$$

1. Numerical Analysis

The temperature, velocity and pressure drop distribution for straight tube were predicted by using ANSYS FLUENT 15 package . The following assumption is applied on flow in heat exchanger for water and nano fluid

- Three dimensional, single phase**.**
- Turbulent flow.
- Fully developed flow, incompressible fluid flow,
- Water alone, Nano fluid consider a Newtonian fluid .
- Steady state condition, radiation, free convection, chemical reactors are negliable.

The fluid flow can be depicted by the continuity، Naviers stokes and energy equations in differential form can show as following [17].

1. Continuity equation

$$\frac{\partial \rho u}{\rho x}+\frac{\partial \rho v}{\partial y}+\frac{\partial \rho w}{\partial z}=0 \qquad (16)$$

2. Momentum equation

x- Momentum

$$\rho(\mathrm{u}\frac{\partial \mathcal{U}}{\partial \mathrm{x}}+\mathrm{v}\frac{\partial \mathcal{U}}{\partial \mathrm{y}}+\mathrm{w}\frac{\partial \mathcal{U}}{\partial \mathrm{z}}) = \rho\mathcal{X}-\frac{\partial \mathcal{P}}{\partial \mathcal{X}}+\frac{1}{3}\mu\frac{\partial}{\partial \mathcal{X}}(\frac{\partial \mathcal{U}}{\partial \mathcal{X}}+\frac{\partial \mathcal{V}}{\partial \mathcal{Y}}+\frac{\partial \mathcal{W}}{\partial \mathcal{Z}}) + \mu\nabla^2\mathcal{U}) \qquad (17)$$

Y-momentum

$$\rho\left(\mathcal{U}\frac{\partial \mathcal{V}}{\partial \mathcal{X}}+\mathcal{V}\frac{\partial \mathcal{V}}{\partial \mathcal{Y}}+\mathrm{w}\frac{\partial \mathcal{V}}{\partial \mathcal{Z}}\right) = \rho\mathcal{Y}-\frac{\partial \mathcal{P}}{\partial \mathcal{Y}}+\frac{1}{3}\mu\frac{\partial}{\partial \mathcal{Y}}\left(\frac{\partial \mathcal{U}}{\partial \mathcal{X}}+\frac{\partial \mathcal{V}}{\partial \mathcal{Y}}+\frac{\partial \mathcal{W}}{\partial \mathcal{Z}}\right) + \mu\nabla^2\mathcal{V} \qquad (18)$$

z- Momentum

$$\rho\left(\mathcal{U}\frac{\partial \mathcal{W}}{\partial \mathcal{X}}+\mathcal{V}\frac{\partial \mathcal{W}}{\partial \mathcal{Y}}+\mathcal{W}\frac{\partial \mathcal{W}}{\partial \mathcal{Z}}\right) = \rho\mathcal{Z}-\frac{\partial \mathcal{P}}{\partial \mathcal{Z}}+\frac{1}{3}\mu\frac{\partial}{\partial \mathcal{Z}}\left(\frac{\partial \mathcal{U}}{\partial \mathcal{X}}+\frac{\partial \mathcal{V}}{\partial \mathcal{Y}}+\frac{\partial \mathcal{W}}{\partial \mathcal{Z}}\right) + \mu\nabla^2\mathcal{W} \qquad (19)$$

energy equation

$$\rho cp(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z} \\ =\left(u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y}+w\frac{\partial p}{\partial z}\right)+k\nabla^2 T+2\mu\left\{\left(\frac{\partial u}{\partial x}\right)^2+\left(\frac{\partial v}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right\} \\ +\left\{\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)^2+\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)^2+\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)^2\right\} \\ -\frac{2}{3}\mu\left\{\frac{\partial v}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}\right\} \tag{20}$$

3.1 Mesh Generation

The description of physical procedure by using the numerical solution that utilized the grid generation as the first step. The result of the solution depends upon the quality of grid. In the present study *tetrahedral* cell are used for three dimensions design for a good mesh with higher accuracy solution as shown in figure 4. For complex model and good mesh generation، the number of cell increase with the time resolution conversely the accuracy increase that's depend on the ability of computer, memory and process to solve and offer good mesh in this study the number of nodes (328322) and number of element (1555576) . In the present study($К - \boldsymbol{\varepsilon}$) model is used to simulate the fluid flow and the heat transfer in the heat exchanger.

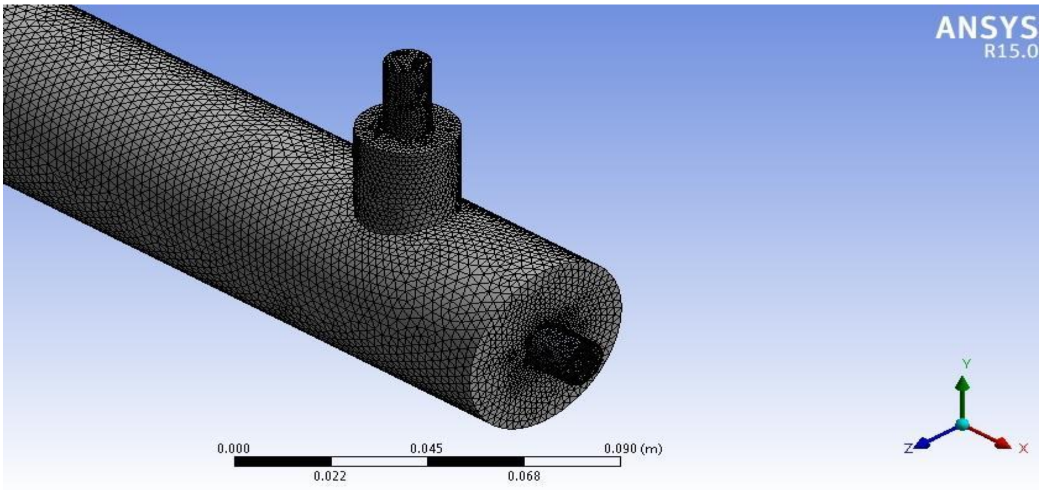


Figure 4. Mesh Generations.

1.2 Implementation of Boundary Condition

For each zone of computational domain, the boundary conditions are applied in CFD simulation is consider the main conditions to definition the physical model as following

a. Inlet boundary condition: Four velocities were selected in the inner tube while one velocity in outer tube . Three temperatures were studies for inner tube of(45،55,65)C. While one temperature was used for outer tube of 25C

b. Initial condition: The initial condition important to start the solution and the iteration .

c. **Wall boundary condition:** The wall boundary conditions can be summarized as: there is no slip wall boundary condition, insulation annular side surface, solid reign and bound the fluid the condition are applied.

4. Experimental Results

4.1 Validation of Experimental Results

Plain tube was validated with Gliniski correlation(1976)[18]:

$$Nu = \frac{\left(\frac{f}{8}\right)(Re - 1000)pr}{1 + 12.7\left(\frac{f}{8}\right)^{0.5}\left(pr^{\frac{2}{3}} - 1\right)} \qquad (21)$$

also comparesion with [19] and [20] to Nusselt number for plain tube shows discrepancy(3, 10 and 3)% respectively as shown in figure 6‹5.

The friction factor for plain tube are valid with Blasius correlation and petukhov (1970) [18], figure 7 shows the validation. This results shows that an agreement with maximum deviation (8 and 10)% respectively,

Figure 8 shows the compression expermental and numerical work for plain tube with discrepency 12%.

Figure 9 shows the validation expermental work with1% concentration nanofluid with refrence work maximum discrepency 5%.

The empirical correlations of Nusselt number and friction factor for plain tube were estimated from the experimental results. This is accomplished by using Dimensionless Groups Analysis program through equation22‹23.

$$\mathrm{Nu}=9.7339Re^{0.96397} \qquad (22)$$

$$f = 0.467823Re^{-0.31263} \qquad (23)$$

Nusselt number for plain tube was predicted with Nusselt number for expermental data found discrepency 0.5%.

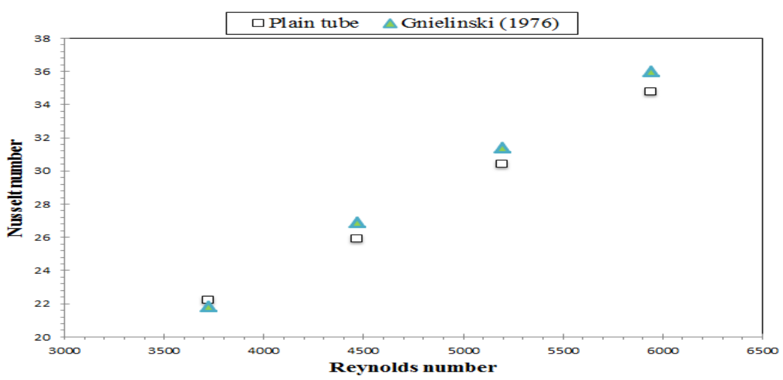


Figure5. Compression the experimental work for plain tube with Gnielinski correlation

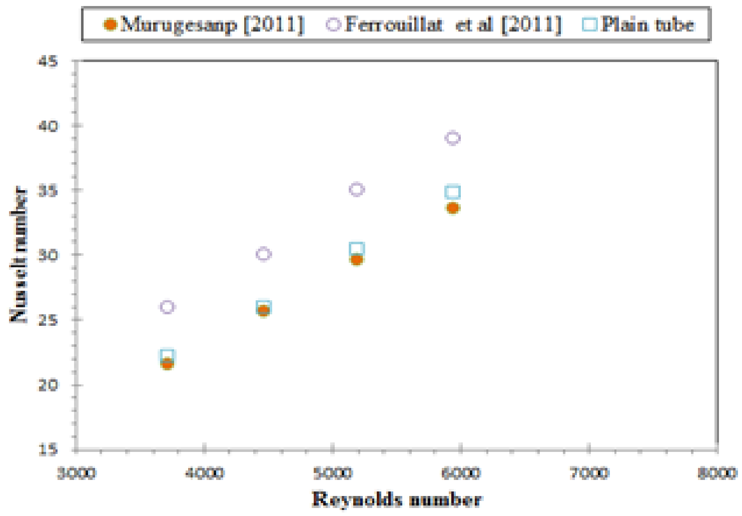


Figure 6. Comperation the experimental work with reference studies

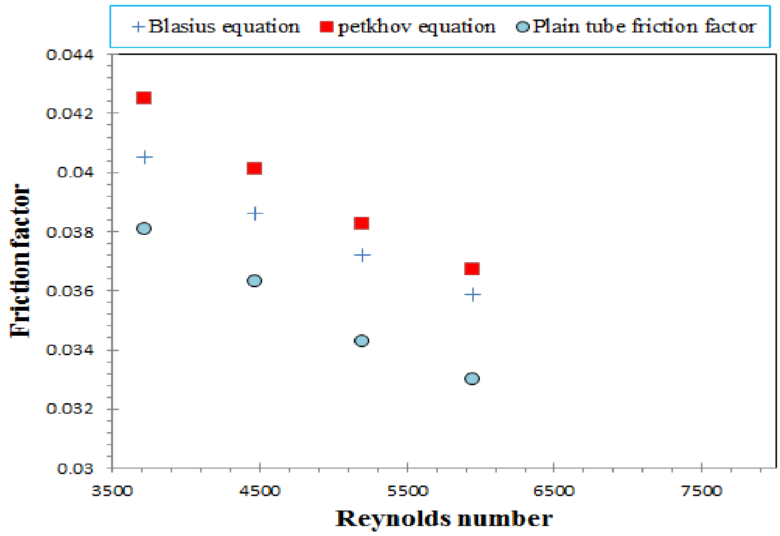


Figure 7. Compressions with correlations for friction factor.

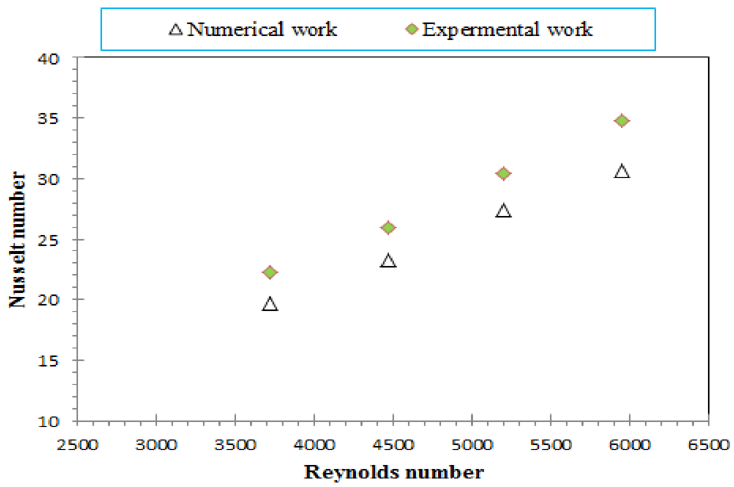


Figure 8. compression experimental and numerical work

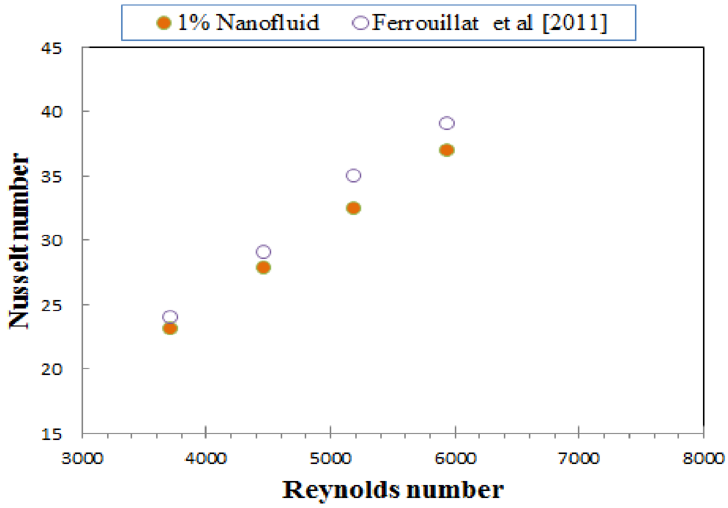


Figure 9. validation experimental work 1% nano fluid with reference work.

4.2 Effect of the Nanofluid insert on heat transfer charastiares and friction factor

Figures 10 shows the relationship between Nusselt number and Reynolds number for (SiO_2 nanofluid / distilled water)with different volume concentration of (0.1، 1and 3)%. The percentage of increasing Nusselt number is higher than that of plain tube by 2.40-4.04%، 6.43-7.96%، 15.72-16.47% for artificial nanofluid with concentration (0.1,1and،3)% respectively.

Figure11 shows the variation of friction factor for nanofluid versus Reynolds number the percentage of increasing friction factor is higher than that of plain tube by 4.84-6.56%،5.45-7.611%, 11.51-13.12% for volume concentration of (0.1،1 and 3) % respectively.

The empirical correlations of Nusselt number and friction factor for nanofluid were estimated from the experimental results. This is accomplished by using Dimensionless Groups Analysis program (DGA) through equation24 and 25.

$$Nu=2.1138Re^{0.8844}\emptyset^{2.0281} \quad (24)$$

$$f = 0.7366Re^{-0.34827}\emptyset^{1.63633} \quad (25)$$

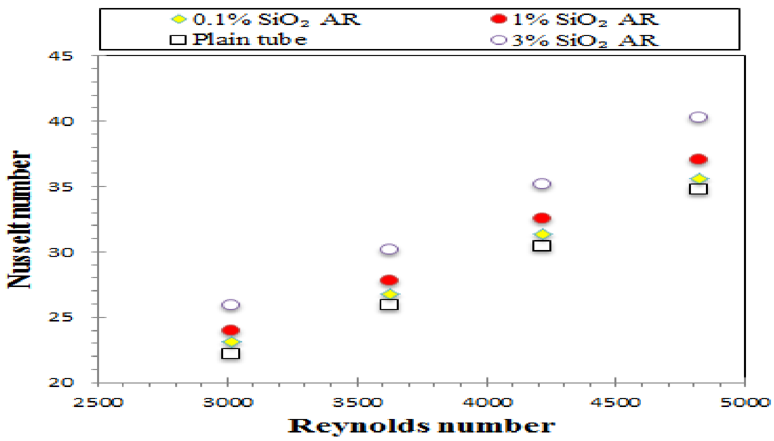


Figure10. Variation of Nusselt number versus Reynolds number for nano fluid.

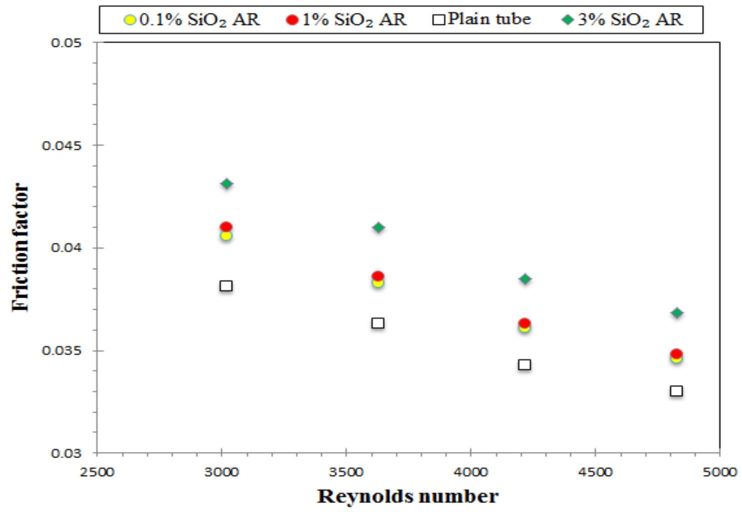


Figure 11. variation of friction factor versus Reynolds number for nano fluid.

4.3 Performance Evaluation with Nano fluid

Figure12 shows the variation of the thermal performance factar with Reynolds number .It can be observed that the performance factor decreases with Renoldes number increases .While the thermal performance increases with the volume concentration of nanoparticles increases، due to the volume concentration changing that leads to change in both Nusselt number and friction factor. The maximum thermal performance factor of Nano fluid at 3%.

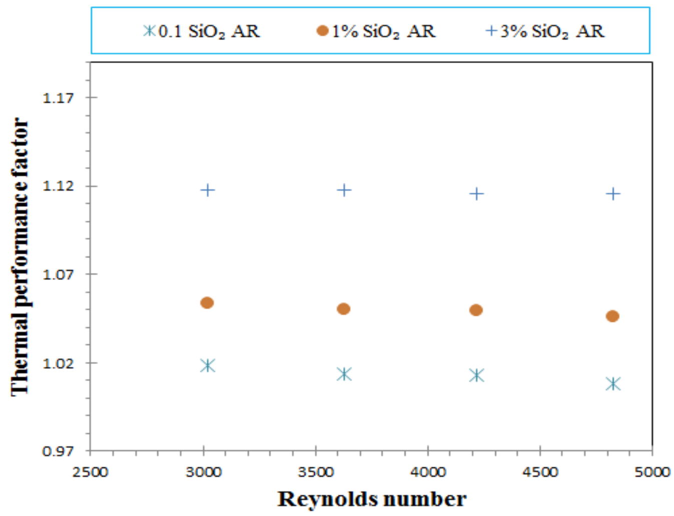


Figure12. Thermal performance factor of the double pipe heat exchanger with nana fluid.

5. Numerical Results

The simulation results from ANSYS FLUENT 15 / package are obtained to show the heat transfer characteristics and fluid flow.

Case 1 -velocity vectors

Figures 13,14,15,16,17,18,19 show the velocities vectors at different locations of X direction with Nano concentration of 3%, temperature of hot fluid of 65ºC and 1.6lpm flow rate. Observed that the secondary flow increases, additional motion and increases swirling in case Nano fluid.

Case 2- temperature contours

Figures20,21,22,23,24,25,26 show the temperature contours in different locations of X direction، with 3% Nano fluid concentration .The results show that the temperature is less than that temperature at the same position for case plain tube this is due to increases in heat transfer coefficient, enhancement and increases the number of swirling provides, increasing in mixing of fluid which leads to increase heat transfer rate.

Case 3-Pressure contours

Figure 27 shows contours of pressure at case with 3% Nano fluid concentration . It's clearly that the pressure drop in case Nano fluid higher than that of plain tube .This is due increase in density and increase viscosity of Nano fluid lead to increase shear stress on tube. The pressure drop increases when volume concentration increases and volume flow rate increases that is due to increase in velocity.

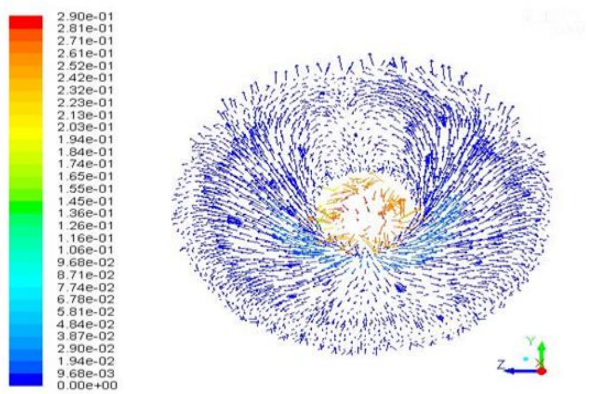


Figure13. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.02 m from inlet.

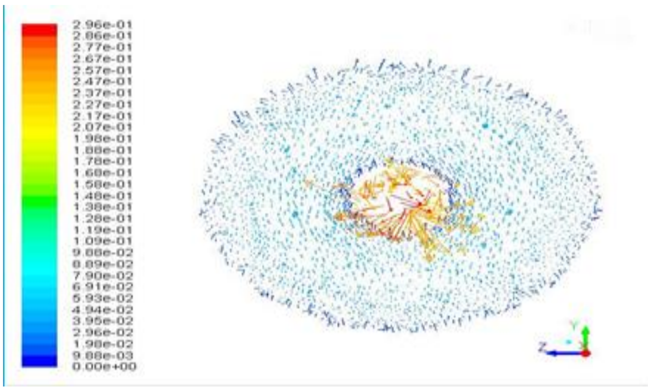


Figure14. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.13 m from inlet.

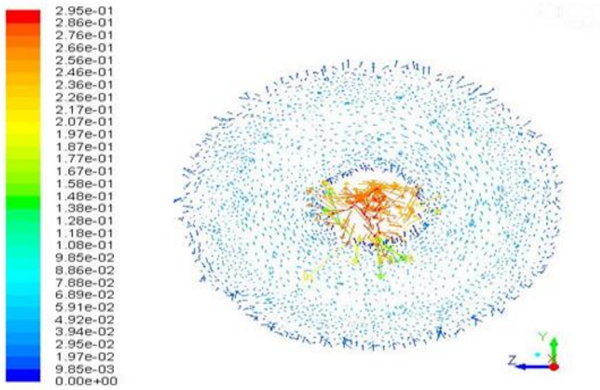


Figure15. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.3 m from inlet.

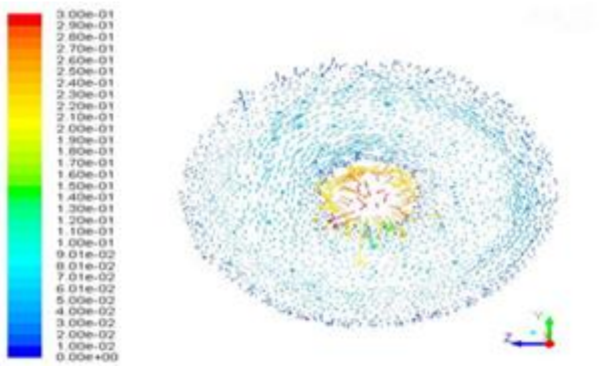


Figure16. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.46 m from inlet.

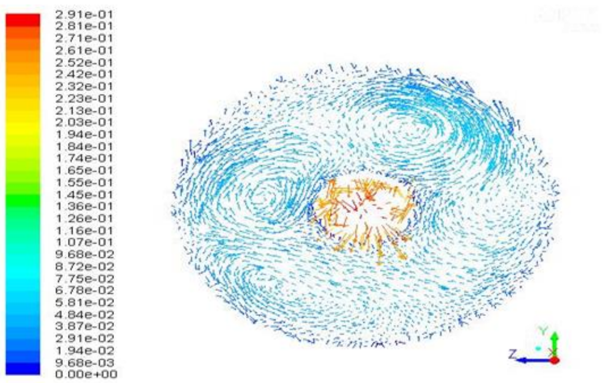


Figure17. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.65 m from inlet.

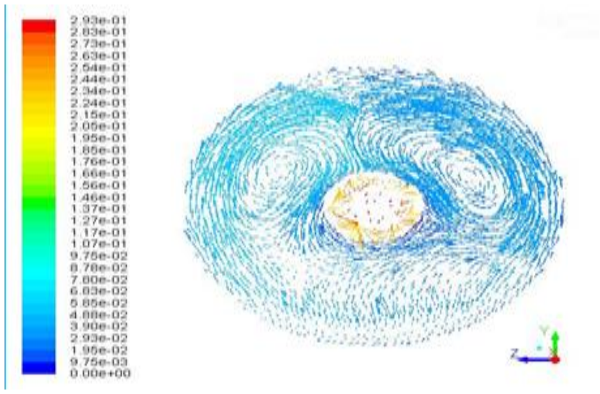


Figure 18. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.84 m from inlet.

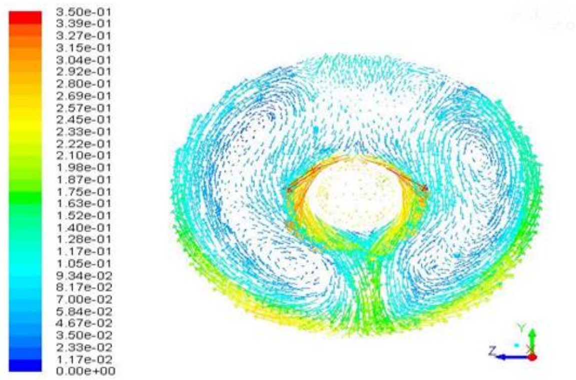


Figure 19. Velocity vector contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 1.012 m from inlet.

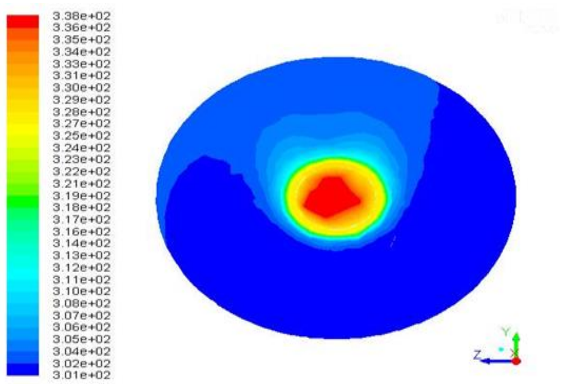


Figure 20. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.02 m from inlet.

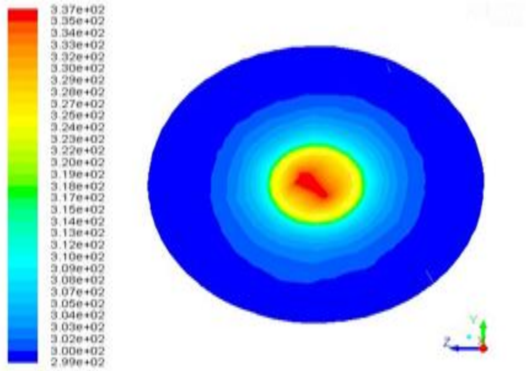


Figure 21. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.13 m from inlet.

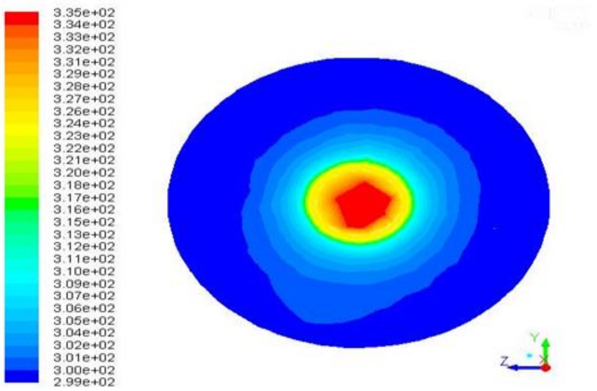


Figure22. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.3 m from inlet.

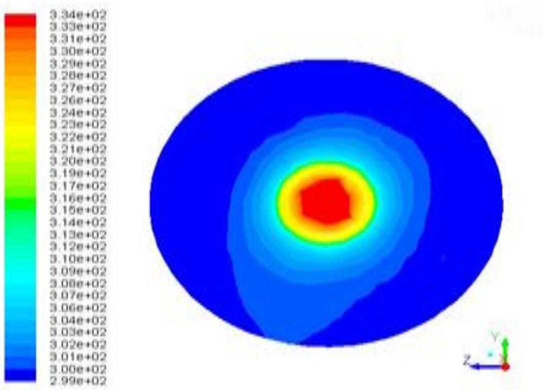


Figure 23. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.46 m from inlet.

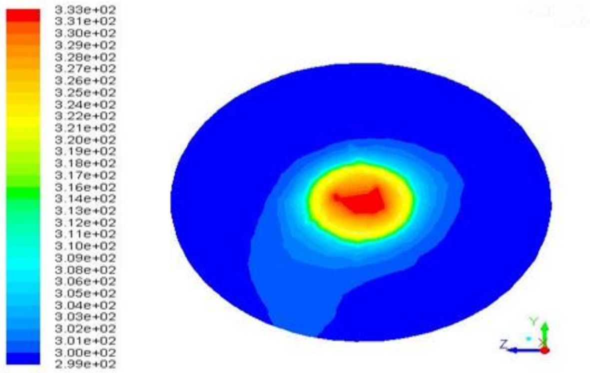


Figure24. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 0.65 m from inlet.

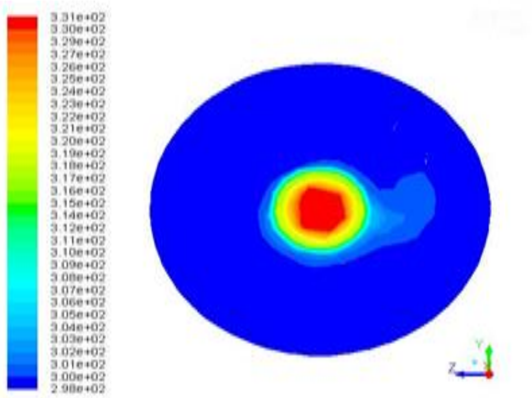


Figure25. Temperature contours in case 3% volume concentration at temperature of hot Nano fluid T=65ºC and volume flow rate 1.6 lpm at x= 0.84 m from inlet.

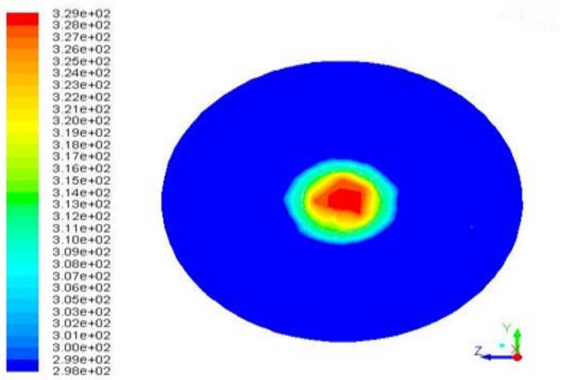


Figure26. Temperature contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm at x= 1.012 m from inlet.

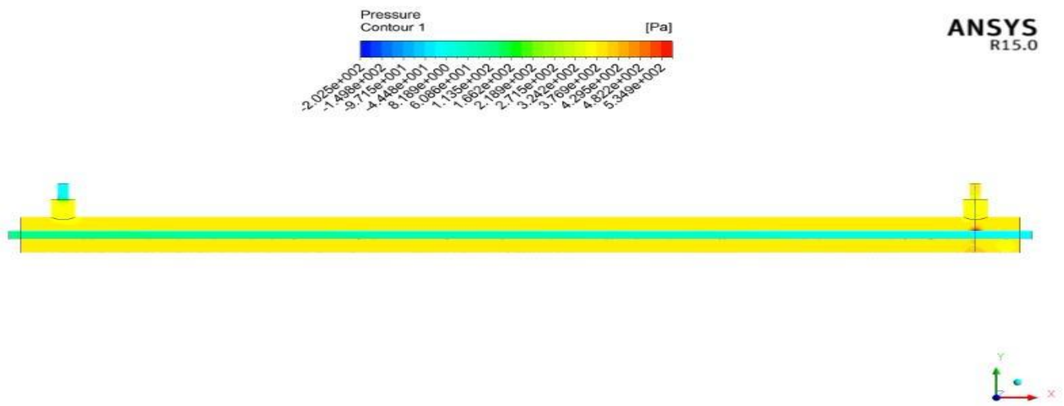


Figure27. Pressure contours in case 3% volume concentration at temperature of hot Nanofluid T=65ºC and volume flow rate 1.6 lpm

6. Conclusions

The used of the inner copper tube with Nano fluid at concentration 3% and volume flow rate of 1.6 lpm give an enhancement up to (15.72, 11.51, 11.57) % for Nusselt number, friction factor and performance factor respectively. The used of Nano fluid SiO_2 / distilled water show that the heat transfer enhancement compared with plain tube . The maximum heat transfer occur at high concentration nanoparticle in based fluid, nanoparticle in base fluid carried out different physical properties effect in heat transfer coefficient .

The simulation results by using ANSYS FLUENT 2015 package gives in agreement with experimental work the correlation are predicted.

Nomenclature

Symbol	Description	Units
A_S	Surface area	m²
C_P	Specific heat at constant pressure	J/k. g
d_o	Outer diameter of inner tube	m
D_i	Inner diameter of annuli side	m
D_h	Hydraulic diameter	m
f	Friction factor	……
h	Convective heat transfer coefficient	W/m².C
k	Thermal conductivity	W/m. C
L	Length	m
m	Mass	kg
$\dot{m}_C$	Mass flow rate in annuals side	Kg/s
$\dot{m}_h$	Mass flow rate in tube side	Kg/s
Nu	Nusselt number	……
pr	Prandtle number	……
Q	Heat transfer rate	W
Re	Reynolds number	……
T_o	Outlet temperature	C
T_i	Inlet temperature	C
U	Overall heat transfer coefficient	W/m².C

Greek symbols

Symbol	Description	Units
∅	Volume concentration	……
ρ	Density	Kg/m^3
μ	Dynamic Viscosity	Pa.s
η	Thermal performance factor	……
Δp	Pressure drop	N/m^2

Subscript

Symbol	Title
a	annular
bf	Base fluid
c	cold
eff	Effective
f	Fluid
h	Hot
i	Inner
o	Outer
nf	Nano fluid
p	Particle
r	Relative value
Pt	Plain tube

Abbreviations

Symbol	**Detentions**
AR	Artificial
PVC	Poly vinyl chloride
DGA	Dimensionless Groups Analysis

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