EXPERIMENTAL AND NUMERICAL INVESTIGATION TO THE EFFECT OF EGG SHELL NANOPARTICLES ON THE PERFORMANCE OF DOUBLE PIPE HEAT EXCHANGER

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ABSTRACT

Thermal systems are involve in many applications the most important of which are cooling system, so most research aims to improve heat transfer, the enhancement of egg shell nanoparticles with distilled water in double pipe counter flow heat exchanger was studied. The results show that Nusselt number increase as volume concentration and Reynolds number increase. The increasing percentage of Nusselt number, friction factor, performance factor of (10.95, 11.878 and 6.88)% respectively than plain tube for maximum volume concentration and volume flow rate 5%,1.6 l/min respectively, fully developed turbulent flow, homogeneous Nano fluid with ranging Reynolds number from 3256.27 to 5202.33. ANSYS FLUENT 2015 package is used to simulate the heat transfer and fluid flow in the heat exchanger. The results show that the heat transfer increases when Nano fluid was added. The agreement observed with the experimental work with maximum discrepancy (12%).

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
Ø	Volume concentration	……
ρ	Density	Kg/m³
μ	Dynamic Viscosity	Pa.s
Ŋ	Thermal performance factor	……
Δp	Pressure drop	N/m²

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	Definition
CFD	Computational fluid dynamics
DGA	Dimensionless Groups analysis
PVC	Poly vinyl chloride

INTRODUCTION

Nanostructured metal oxides are promising materials, because of its unique properties CaO metal oxide is utilized in numerous applications similar as cement industry, biodiesel production, biosensors, tissue engineering, petroleum manufacture, electric lighting and power production Ashok et.al.[2015]. Abad et al., [2013] investigated experimentally the heat transfer and pressure drop characteristics of Cu - water and Al-water nano fluid in a spiral coil in laminar stream administration the experiments were carried out under the laminar flow regime and constant wall temperature condition were done for water and two types of nano fluid with different volumetric concentrations of (0.55%, 1.12% and 2.23%) the results showed that the thermal conductivity of Cu –water nano fluid is approximately 18% higher than Al –water the Nusselt number increments by augmenting the Dean number. This mean, all nano fluids have a higher Nusselt number compared to distilled water, the use of nano fluids with the higher volume fraction provides considerably higher Nusslet number and pressure drop comparing two enhancement heat transfer technique show that utilizing spiral coil is effect than using nano fluids. Selvam et al., [2016] investigated experimentally the convective heat transfer qualities of water– ethylene glycol blend fluids with silver nano fluids under laminar, transitional and turbulent administration, the volume concentrations of silver nano particles 0.05%, 0.1%, 0.15%, 0.3% and 0.45% are considered. Thermo physical properties measurements such as thermal conductivity, viscosity, density and specific heat capacity were executed utilizing conventional techniques. Convection estimations were done in a tube in tube counter flow heat exchanger using nano fluids as the hot fluid. The impact of nano fluid mass flow rate extending from 5 g/s to 45 g/s and inlet temperature of nano fluid at 35 °C and 45 °C and cold water is 25c results showed the convective heat transfer coefficient increased with increasing Reynolds number, particle concentration and inlet temperature. The highest enhancement of convective heat transfer coefficient of nano fluid is noticed to be 42% at 0.45 vol% compared with pure base fluid the pressure drop of nano fluid increases slightly for a nano fluid volume concentration up to 0.15 vol%. when contrasted and the base fluid, the distinction of pressure drop is insignificant so that the utilization of nano fluid has constrained punishment on the pressure drop up to 0.15 vol% loading of silver nanoparticles. Be that as it may, past 0.15 vol%, the pressure drop increments significantly which confines the utilization of these nano fluids at higher concentration for engineering applications. Xuan and Li [2003] investigated convective heat transfer and flow features of the nano fluid in a tube. Both the convective heat transfer coefficient and friction factor of the sample nano fluids for the turbulent flow are measured ,cu nanoparticles below 100nm with constant heat flux boundary condition and volume concentration0.3,0.5 0.8,1,1.2,1.5,2% from volume with range of Reynolds number (10000-25000) results shows that the suspended nanoparticles remarkly improve heat transfer procedure and the nano fluid has biggest heat transfer coefficient than that of the original base liquid under the same Reynolds number. The heat transfer feature of a nano fluid increases with the volume fraction of nano particles the friction factor for the dilute nano fluid comprising of water and Cu-nanoparticles is roughly the same as that of water the nano fluid with the low volume fraction of the suspended nanoparticles brings about no additional punishment of pump power. Hong et al., [2005] studied experimentally the enhanced thermal conductivity of Fe with concentration (0.2,0.3,0.4,0.55)% vol and Cu with (0.1,0.55)% vol the

results showed the thermal conductivity of a Fe nano fluid is increased nonlinearly Fe nano fluid displays 18% improvement with 0.55 volume fraction of nanoparticles while the Cu nano fluid presents 2% and 14% upgrade with 0.1 and 0.55 vol. % nanoparticles with little agglomeration therefore the highly thermally conductive material increases with increase sonication time thermal conductivity of nano fluids increases nonlinearly with the volume fraction of particles, nano particle at high concentration easily to agglomeration since none of the conventional theories for two segment blend predicts the improved thermal conductivity of nano fluids. Aliabadi et al., [2014] studied experimentally the   Copper–water nano fluid flow through different plate-fin channel heat exchanger seven plate-fin channels, including plain, perforated, offset strip, louvered, wavy, vortex generator, and pin, were fabricated and tested. The copper–waterb nano fluids were produced by a one-step method with five nano particles weight fractions (0%, 0.1%, 0.2%, 0.3%, and 0.4%) The required properties of the nano fluids a constant wall temperature as   boundary condition   and with volume flow rate (2,2.5,3,3.5,4,4.5,5) L/min temperature (298 to 313K) the results delineated that both the convective heat transfer coefficient and the pressure drop estimations of the considerable number of channels upgrade with augmenting the nanoparticles weight fraction. The suitable thermal hydraulic performance and maximum extreme diminishment of surface region were found for the vortex generator channel the utilization of nano fluids in this channel has more advantage at the lower flow rates and high nanoparticles weight fractions and gave the considerable esteems to choose the ideal channel for use in the plate fin heat exchangers and distinguish the impact of nano fluid flow with various nano particles weight fraction on the performance of each channel. Sadeghinezhad et al., [2014] an experimental investigation was performed to estimate the heat transfer characteristics and the pressure drop of a graphene nano plate nano fluid in a horizontal stainless steel tube that was subjected to a uniform heat flux at its outer surface. at concentrations of 0.025, 0.05, 0.075, and 0.1 wt % Thermal conductivity increases as the nano fluid temperature increases, and the enhancement in thermal conductivity for the nano fluid was between 7.96% and 25%. The increase in the pressure drop ranged from 0.4% to 14.6%.The nano fluid viscosity was highly dependent on the temperature. It decreased at upper temperatures, and their increment was 9−38% compared with distilled water, The use of the nano fluid provides significantly higher heat transfer coefficients up to 160%. The convective heat transfer coefficient increases as the flow rate and heat flux increase .An increase of the thermal performance could be obtained as 1.66, 1.70, and 1.77 for the nano fluid at 8231, 10 351, and 12 320 W/m² , respectively, for a 0.1 wt % concentration. Sultan[2012] Studied theoretically and experimentally investigated the heat transfer and flow of nano fluids through a horizontal and an slanted rounded tube heated by an axial fixed heat flux, fully developed regime with laminar flow, three types of nano fluids Al (25nm) – distilled water, Al_2O_3 (30nm) – distilled water and CuO (50nm) – distilled the single phase, the finite difference methods using the alternating direction implicit method and Gauss elimination technique. The results based on the fact that the secondary flow created by natural convection has significant effects on the heat transfer process, reveal an increase in the Nusslet number values as the heat flux increases and as the angle of inclination moves from the vertical to the horizontal position the experimental study includes construction of experimental test rig range of Reynolds number are chosen (100 – 900), heat flux ranging between (588– 7910W/m^2) which cover a large range of Rayleigh number between ($1{\times}10^3$– $4{\times}10^6$)and volume fractions from (0.25–2.5 vol %). Nusselt number ratio were estimated to be (45%, 31%, 25%) for the three nano fluids (Al, Al_2O_3, CuO) – distilled water respectively, with the uniform heat flux and (36%, 27%, 22%) constant wall temperature .It should be observed that the augment of heat transfer highly depend on volume fraction, nanoparticles type, nanoparticles size, flow regime. Nano fluids that include metal nanoparticles show more improvement than oxide nano fluids. The experimental and numerical study show good agreement with   ±9 % for average Nusselt

number. The previous study are carried out to improve the performance of heat exchanger to saving in energy, reduce process time, raise thermal rating and lengthen the working life of equipment   by using different technique in the present study we used natural egg shell nanoparticle the advantage of it are   cheap, less dangerous , give an enhancement in heat transfer coefficient . The objective of the present work study the effect of using egg shell nanoparticles on enhancement heat transfer in double pipe heat exchanger in different volume concentration (1, 3, 5%) of eggshell nanoparticles.

EXPERIMENTAL APPARATUS AND PROCEDURE
Test Section Regime:
The test section regime consists of the following parts
The inner pipe.
 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 length of the tube   is 1057 mm . 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**.**
The outer tube
External tube made from PVC with the inner   and outer diameters of   (44.4 and 50) mm respectively and length of 1032mm. The outer surface of the tube was insulated by 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 figure1 shows the model of the double pipe heat exchanger and figure2 shows schematic of the expermental work,
Experimental test rig
        The experimental test rig consists of the following parts as shown in figure. 2:
  1.   Test section (double pipe heat exchanger).
  2.   Chilled water unit: its capacity 1 ton comprises of the following parts: condenser, evaporator, solenoid, fan, capillary tube, accumulator.
  3.   Control board: comprises of temperature controls to control the temperature hot and cold fluid inlet in the test section. Also two switches on the two sides of the board used to open the pump for circling the hot and cold fluid.
4. Hot fluid supply system: the hot fluid supply system consists tank 3 litter capacity made from galvanized steel with thickness 1 mm insulated by fibre glass, pump (Marqose , power consumption 120w,power supply220V/50HZ ,maximum capacity 12 l/min ,maximum Head 12M,RPM 2850),Flow meter used to measure the hot fluid in the range 25-250 l/hr., by bass valve, heater (1500 W).
 5.Cold water supply system: the cold fluid water cycle (water only) system consists of tank (16) litters made of galvanized steel with thickness 1 mm insulated by fiber glass ,by pass valve, flow meter used to measure the cold water in the range 4-16 l/min ,pump (Marqose, power consumption 60 w , power supply 220V/50 HZ ,maximum capacity 20 l/min max head 10m, RPM2850) cold water put in annulus side to cold hot water and obtain height heat transfer .

Procedure of experimental data

To evaluate the thermal performance of the double pipe heat exchanger counter flow the following operational variables were utilized for hot fluid the flow rate: (1, 1.2, 1.4and1.6) l/min , the inlet temperature (45, 55, and 65) ℃, cold fluid in annular side is 25℃ with 4 l/min flow rate.   Preparation   the Nano fluid with all concentration (1,3,and 5%) volume concentration, running the chilled water unit, the pump of cooled water open by the switch in the control board. After the cooled unit is running during this time the heater remains turn off. Reduce the velocity for cooled water until settled flow rate 4l/min. The heater turn on and pump of hot water opened using the switches in the control board to circle high temp water in the internal cycle. Adjust the fluid flow by control valve until the flow rate achieve Reynolds number required .The procedure becomes steady state after 30-40 minute, 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 the thermocouple within range (-270 to 370) ℃ were recorded by temperature recorder device model BTM-4208SD,12 channels within SD card. Record the pressure difference on both ends of the test section by differential manometer model Lurton PM-9100(2000mbar*1mbar) also the inlet and outlet pressure were recorded by pressure gages within range (1-10)bar are used .

Physical properties of egg shell:
Table   (1) shows the physical properties of egg shell

PREPARATION OF NANO FLUID

For proper preparation of nanofluid with stable nanoparticles and low agglomeration, the required nanopartical concentration can be calculated as:

$$\emptyset\% = \frac{\dfrac{mp}{\rho p}}{\dfrac{mp}{\rho p} + \dfrac{mbf}{\rho bf}} \tag{1}$$

amount for distilled water for above condition the nanoparticle amount are(67.3 ,206.3, and 351.1)grams for concentration 1,3,5% respectively for egg shell nanoparticles the flask containing with (3) litter distilled water, put the required weighting of nanoparticles then put the flask into the bath of ultrasonic vibration homogenizer device, put on the device and impeller the impeller diminish the level of blending time 10% and get the homogeneous arrangement without sedimentation. Nanoparticles blending with distilled water at (13) hours to give homogeneous Nano liquid.

2.6 Performance Variables in Double Pipe Heat Exchanger:
The operational condition as: fully developed flow turbulent regime with constant   wall temperature insulation outer surface for annular side . the hot fluid (with and without nanoparticles) flow inside inner tube and cold fluid flow in annuli, heat transfer rate for Nano fluid   through tube side (hot fluid) the mathematical calculation for heat exchanger pointed by Kakac and Liu [2002]and Incropera and Doot [1986]as:

$$Q_h = \dot{m}_h \, C_{p_h} (T_{ho} - T_{hi}) \tag{2}$$

Heat transfer rate for water flow through annular side (cold water)

$$Q_c = \dot{m}_c C_{PC} (Tc_o - Tc_i) \tag{3}$$

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The Logarithmic Mean Temperature Difference (ΔT_{lm})

$$\Delta T_{lm} = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln \frac{(T_{h1} - T_{c_2})}{(T_{h2} - T_{C1})}} \tag{4}$$

Average heat transfer rate

$$Q_{ave} = \frac{Q_h + Q_c}{2} \tag{5}$$

Overall heat transfer coefficient (U)

$$U = \frac{Q_{ave}}{A_s \Delta T_{lm}} \tag{6}$$

Reynolds number for the annular side

$$Re = \frac{\rho u_a D_h}{\mu} \tag{7}$$

The Nusselt number for the annular side calculated by Dittus –Boelter correlation :

$$Nu = \frac{ho \times D_h}{k} = 0.023\, Re^{0.8} pr^{0.3} \tag{8}$$

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

$$U_O = \frac{1}{\frac{1}{h_o} + \frac{d_o \ln \frac{d_o}{d_i}}{2k} + \frac{d_o}{h_i d_i}} \tag{9}$$

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

$$Nu = \frac{h_i d_i}{k_i} \tag{10}$$

Friction factor calculated

$$f = \frac{\Delta p}{(\frac{L}{D})(\frac{\rho u^2}{2})} \tag{11}$$

The performance of heat exchanger (η) Shrirao et al. [2013]:

$$\eta = \frac{Nu_{nf}/Nu_{pt}}{(f_{nf}/f_{pt})^{1/3}} \tag{12}$$

Thermal conductivity Mostafizur[2014]:
Patel et al   [2010] model

$$\frac{keff}{kbf} = (1 + 0.135(\frac{kp}{kbf})^{0.273} \emptyset^{0.467} \left(\frac{T}{20}\right)^{0.547} (\frac{100}{dp})^{0.234} \tag{13}$$

Density:
Ho et al [2010]

$$\rho nf = 1001.064 + 2738.6191\emptyset - 0.2095T \tag{14}$$

Specific heat:

palm et al [2006]

$$(cp)nf = (1 - \emptyset)(cp)f + \emptyset(cp) \tag{15}$$

Viscosity:
Sekher   et al [2015]

$$\mu r = 0.935(1 + \frac{Tnf}{70})^{0.5602} (1 + \frac{dp}{80})^{-0.05915} (1 + \frac{\emptyset}{100})^{10.51} \tag{16}$$

EXPERIMENTAL RESULTS

Validation of Experimental Results:

 Plain tube was validated with Gliniski correlation[1976]:

$$Nu = \frac{\left(\frac{f}{8}\right) (Re - 1000)pr}{1 + 12.7(\frac{f}{8})^{0.5}(pr^{\frac{2}{3}} - 1)} \tag{17}$$

also comparesion with Ferrouillat et.al.[2011] and Murugesan [2011] for Nusselt number for plain tube shows discrepancy 3%, 10% , 3% respectively as shown in figure (3 ,4). The friction factor for plain tube are validated with   Blasius correlation and Petukhov [1970]as shown in figure (5). The validation results show that an agreement with maximum deviation 8% 10% respectively. Figure(6) shows the compression between expermental and numerical work with discrepency of 12%. The expermental data for plain tube are correlated for Nusselt number and friction factor through equation 18 ,19 by using dimensionless group Analysis program (DGA).

$$Nu = 9.7339Re^{0.96397} \tag{18}$$
$$f = 0.467823Re^{-0.31263} \tag{19}$$

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

Effect the Nanofluid insert on heat transfer charastiares and friction factor

Figure(7) shows the variation Nusselt number versus Reynolds number for (egg shell/ distilled water) Nano fluid with different volume concentration (1,3and 5)% volume concentration. The percentage of increasing Nusselt number is higher than that of plain tube by (2.837-3.669%),(7.395-9.252%),(10.957-13.818%)   for   1,3,5%   volume   concentration respectively, the increament that's because to increase the heat transfer coefficent and thermal condctivety of based liquid. Figure 8 shows the variation of friction factor versus Reynolds number. The   percentage of increasing friction factor is higher than that of plain tube   by (4.545-6.771%), (9.221-11.364%) , (11.878-14.548%) for (1,3,5%)volume concentration respectively .This increase due to the viscosity increase shear strees on the tube wall that is increase in friction factor. The experimental work for Nanofluid was correlated for Nusselt number and friction factor by using Dimensionless Group Analysis program (DGA).

$$Nu = 1.6Re^{0.927} \emptyset^{5.032} \tag{20}$$
$$f = 7.4157Re^{-0.044} \tag{21}$$

The above correlations are valid for ranging (3256.276-5202.335) of Reynolds number. Also the correlation predicted with experimental data for Nusselt number with the discrepancy 0.629 %.

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Performance Evaluation with Nano fluid

Figure9 show the variation of thermal performance factor versus Reynolds number. It can be observed that the thermal performance factor decreased with volume flow rate increase, the maximum thermal performance factor at 5% volume concentration.

NUMERICAL RESULTS

ANSYS FLUENT 15 package is carried out in order to predict the temperature, velocity and pressure drop distribution for straight tube. The following assumption is applied on flow in heat exchanger for water and Nano fluid:

- Turbulent flow.

- Three dimensional, single phase.

- Water alone, Nano fluid consider a Newtonian fluid .

- Steady state condition. Radiation, free convection, chemical reactors are negligible

- Fully developed flow, incompressible fluid flow. The fluid flow can be depicted by the continuity, Naviers stokes equations or momentum equations and energy equation in differential form of   the flow .

1. Continuity equation:

$$\frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} = 0 \tag{22}$$

2. Momentum equation:

x- Momentum

$$\rho(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}) = \rho X - \frac{\partial P}{\partial x} + \frac{1}{3}\mu\frac{\partial}{\partial x}(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}) + \mu\nabla^2 u \tag{23}$$

Y-momentum

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right) = \rho Y - \frac{\partial P}{\partial y} + \frac{1}{3}\mu\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) + \mu\nabla^2 v \tag{24}$$

z- Momentum

$$\rho\left(u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) = \rho Z - \frac{\partial P}{\partial z} + \frac{1}{3}\mu\frac{\partial}{\partial z}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right) + \mu\nabla^2 w \tag{25}$$

3.Energy equation:

$$\rho c_p(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z}$$
$$= \left(u\frac{\partial p}{\partial x} + v\frac{\partial p}{\partial y} + w\frac{\partial p}{\partial z}\right) + k\nabla^2 T + 2\mu\left\{\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right\}$$
$$+ \left\{\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z}\right)^2\right\}$$
$$- \frac{2}{3}\mu\left\{\frac{\partial v}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}\right\} \tag{26}$$

In the current study $(K - \varepsilon)$ model is used to simulate the fluid flow and heat transfer in the heat exchanger, tetrahedral cell are used for three dimensions design for a good mesh with higher accuracy solution as shown in figure 10 for complex model . 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).The boundary conditions are applied in CFD simulation for each zone to definition the physical model as:

a- Inlet boundary condition: three temperature were studied for inner tube of (45,55 and 65) ºC. while one temperature was used for outer tube 25 ºC, four velocities were selected in the inner tube. while one velocity in outer tube.

b- Initial condition: its important to start the iterations and solutions .

c- Wall boundary condition: there is no slip wall boundary condition, insulation annular side surface , solid region and bound the fluid condition are applied.

Figures (11,12,13,14,15,16,17) shows the velocity vectors in case Nano fluid 5% volume concentration in various location at case 65ºC and 1.6 l/min. Observed that the secondary flow increase, additional motion and increase swirling in case Nano fluid.

 Figures(18,19,20,21,22,23,24) shows temperature contours in case5% nano fluid at various location the results shows that the temperature is less than the temperature at the same position for case plain tube this is due to increase in heat transfer coefficient, enhancement and increase the number of swirling provides increasing in mixing of fluid which leads increasing heat transfer rate. Figure 25 shows contours of pressure in case 5% Nano fluid its clearly that the pressure drop in case Nano fluid higher than plain tube this is due increase in density and increase viscosity of Nano fluid effect to increase shear stress on tube the pressure drop increase when volume concentration increase, and volume flow rate increase that is due to increase in velocity.

CONCLUSIONS

1. The use of the inner copper tube with egg shell / distilled water Nanofluid at volume concentration 5% and 1.6 l/min give an enhancement up to (10.95, 11.878 and 6.88)% in Nusselt number, friction factor, performance factor respectively.

2. The used of Nano fluid egg shell/distilled water show the heat transfer increase the enhancement and increase pressure drop.

3. The maximum heat transfer occurs at high concentration nanoparticle in based   fluid

4. The nanoparticle in base fluid carried out different physical properties effect in heat transfer coefficient

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

Table   (1) Physical properties of egg shell as presented by ref. [9] and [10].

Physical properties	ρ (kg/m³)	Cp (j/kg .k)	µ(kg/m .s)	K(W/m . ºC)
Egg shell	2300	888	0.45×10^{-3}	2.25

Fig.1. Model of the double pipe heat exchanger.

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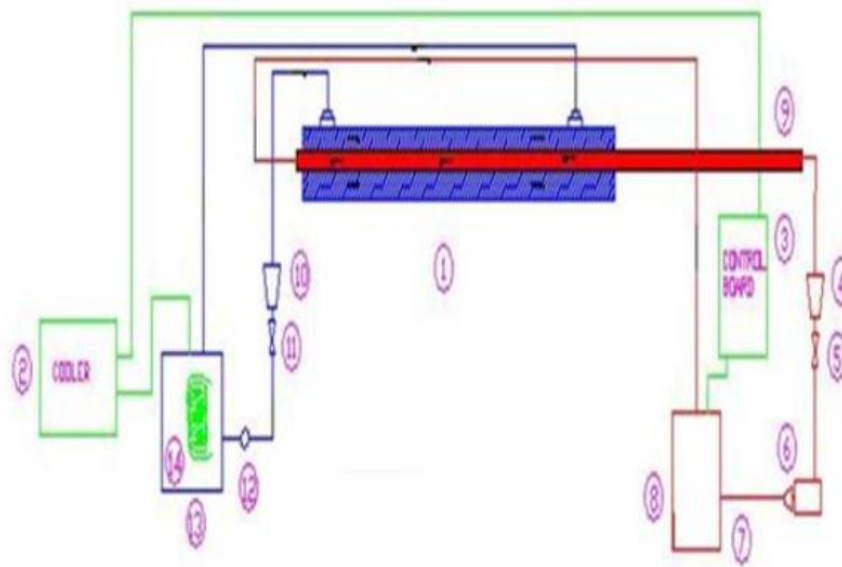
Fig. 2. Schematic of the expermental work.
1..Test section, 2.Cooler, 3.Control bord,4. Hot water flow meter,5. Valve, 6.HotWater pump, 7.Hot Water tank,8. Heate, 9.Fully developed region, 10,Cold water flow mete, 11.Valve,12. Cold water pump, 13.Coldwater tank,14. Evaporater
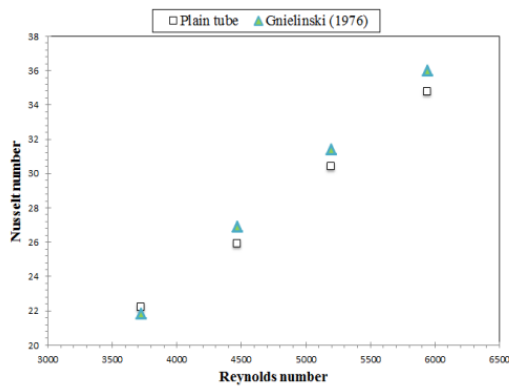

Fig.3. Compression the experimental work for plain tube with Gnielinski correlation
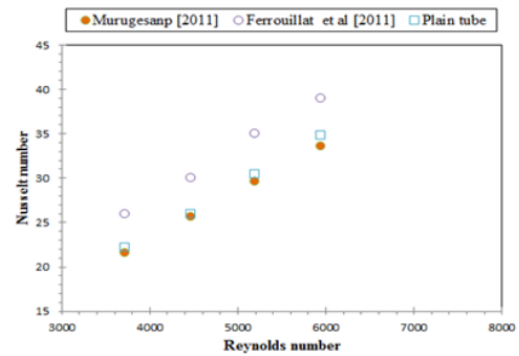
Fig. 4. Comperation the experimental work with reference studies for plain tube.
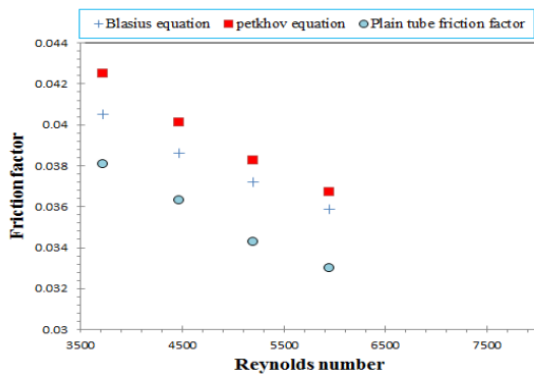
Fig.5. Compressions with correlations for friction factor.
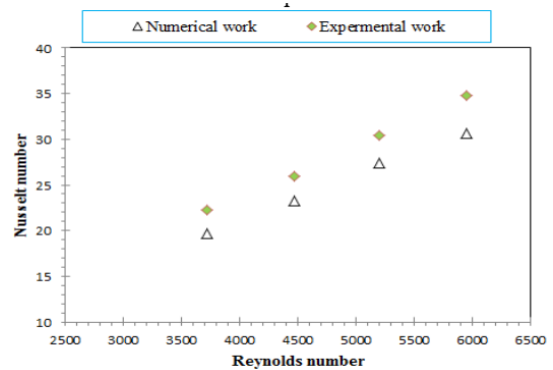
Fig.6. Compression experimental and numerical work
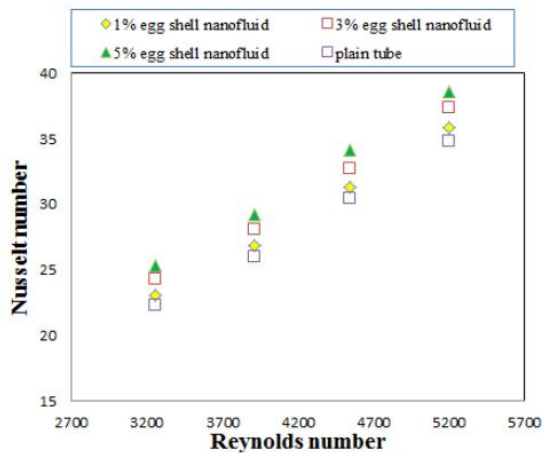
Fig.7. Variation Nusselt number versus Reynolds number for egg shell Nano fluid
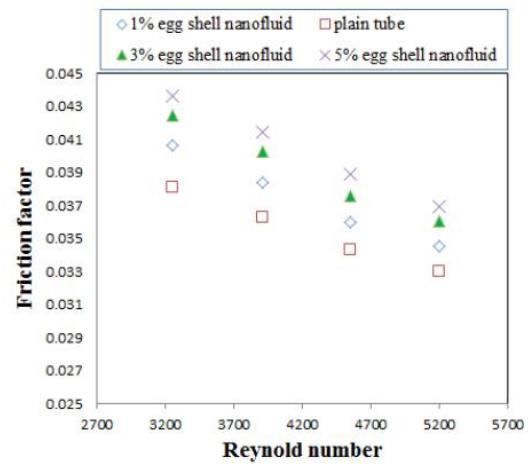
Fig.8. Variation friction factor versus Reynolds number for egg shell nano fluid
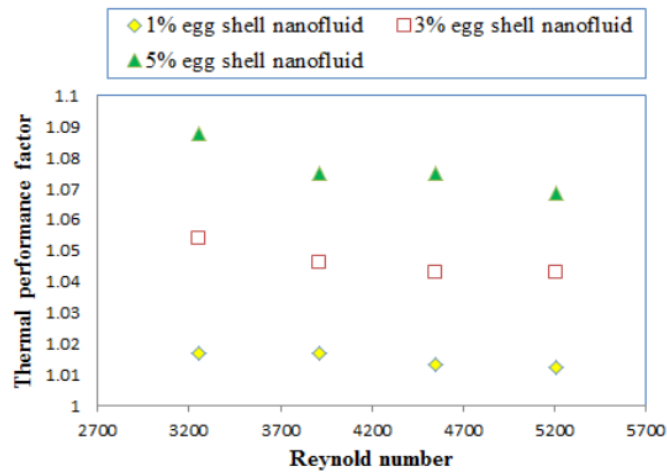
Fig.9. Variation performance factor versus Reynolds number for egg shell nano fluid
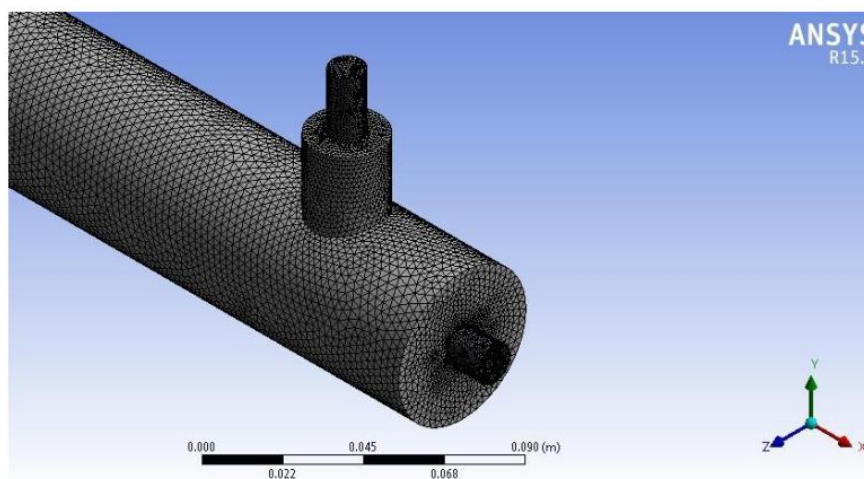
Fig. 10. The mesh of the present model.

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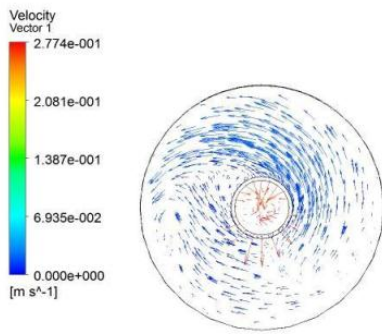
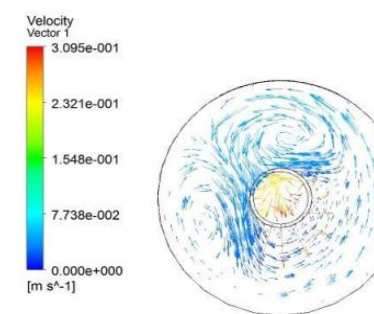Figure11Velocity vector contours   in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.02 m from inlet
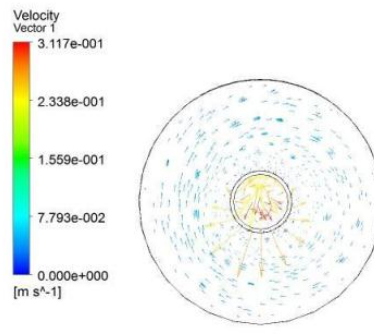
Figure12Velocity vector contours   in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.13m from inlet.
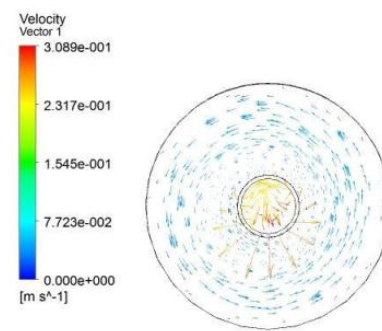
Fig.13. Velocity vector contours   in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.3 m from inlet.
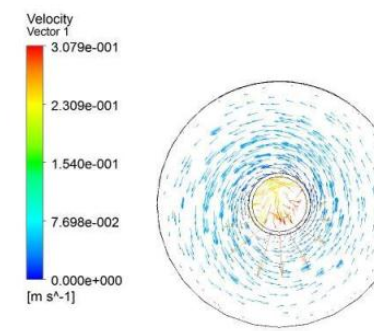
Fig.14. Velocity vector contours   in case 5% volume concentration T=65ºC and   volume flow rate 1.6 l/min at x= 0.46 m from inlet.
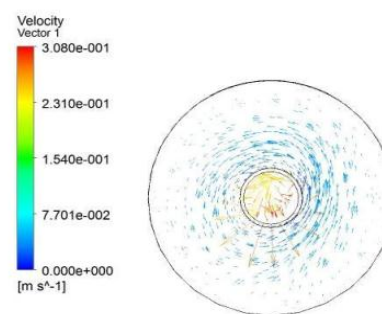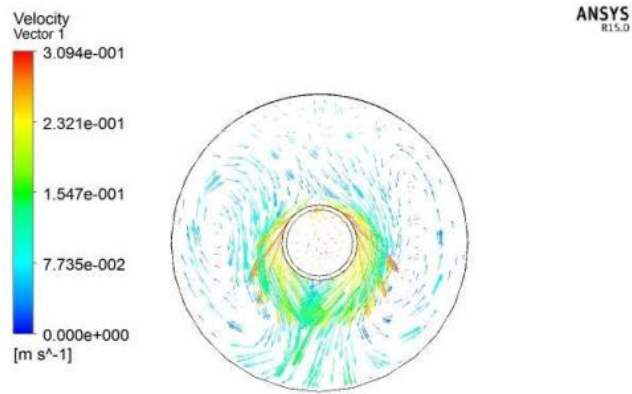

Fig.15. Velocity vector contours   in case 5% volume concentration T=65ºC and   volume flow rate 1.6 l/min at x= 0.65 m from inlet.

Fig. 16. Velocity vector contours   in case 5% volume concentration T=65ºC and   volume flow   rate 1.6 l/min at x= 0.84 m from inlet.

Fig.17.   Velocity vector contours   in case 5% volume concentration T=65ºC and   volume flow rate 1.6 l/min at x= 1.012 m from inlet.
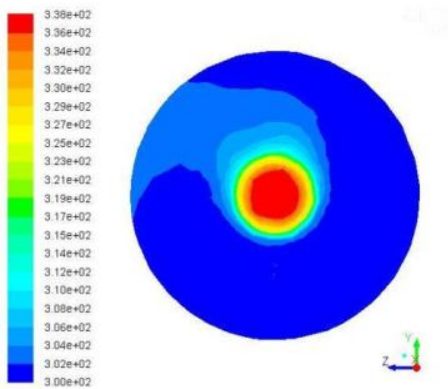

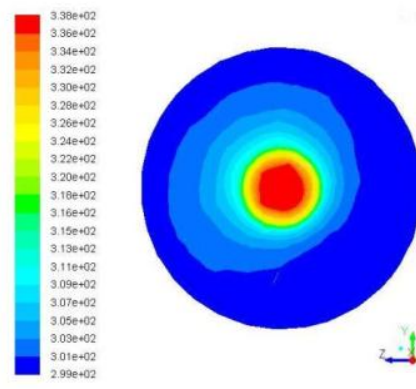Fig.18. Temperature contours in case 5% volume concentration T=65ºC and volume flow   rate 1.6 l/min at x= 0.02 m from inlet.

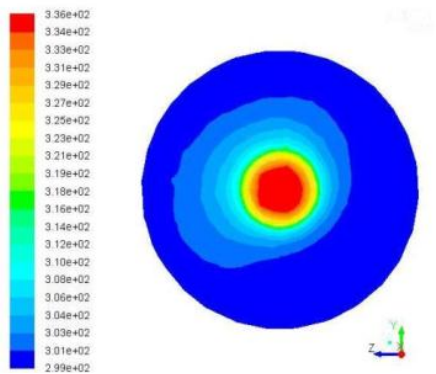Fig. 19. Temperature contours in case 5% volume concentration T=65ºC and volume flow   rate 1.6 l/min at x= 0.13 m from inlet.

Fig.20.   Temperature contours in case 5% volume concentration T=65ºC and volume flow   rate 1.6 l/min at x= 0. 3 m from inlet.
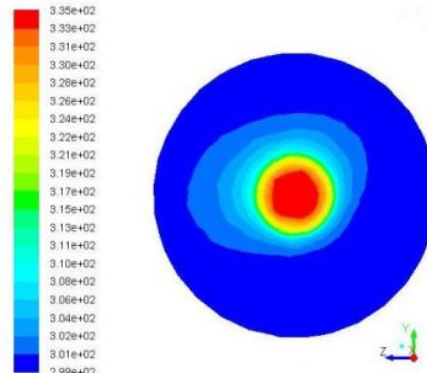

Fig.21. Temperature contours in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.46 m from inlet.
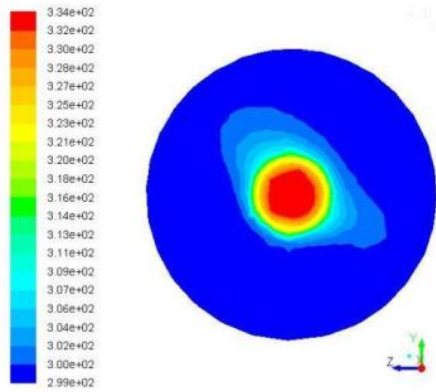
**EXPERIMENTAL AND NUMERICAL INVESTIGATION TO
THE EFFECT OF EGG SHELL NANOPARTICLES ON THE
PERFORMANCE OF DOUBLE PIPE HEAT EXCHANGER**

Muna Sabah Kassim

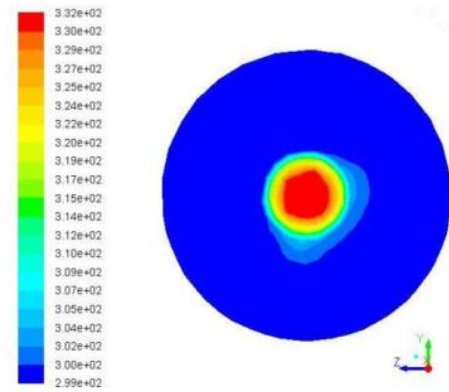Fig. 22. Temperature contours in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.65m from inlet.

Fig.23. Temperature contours in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 0.84 m from inlet.

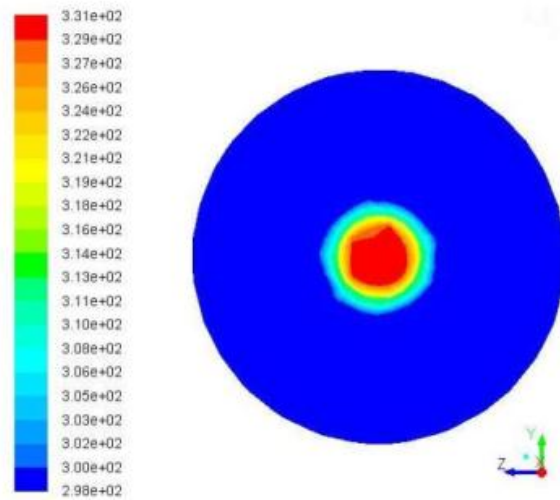. Fig. 24. Temperature contours in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min at x= 1.012 m from inlet

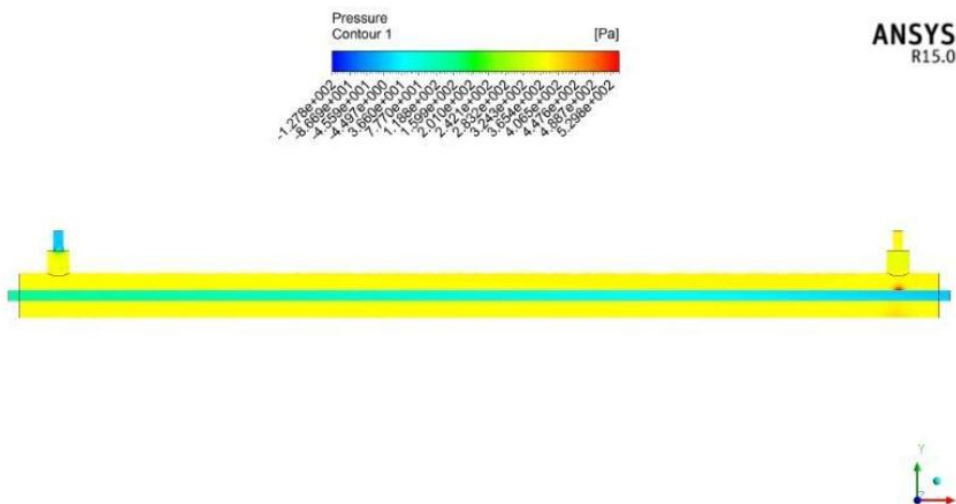
Fig.25. Pressure contours in case 5% volume concentration T=65ºC and volume flow rate 1.6 l/min.

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