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CFD Study Based on Effect of Employing the Single-Walled Carbon Nanotube (SWCNT) and Graphene Quantum Dots (GQD) Nanoparticles and a Particular Fin Configuration on the Thermal Performance in the Shell and Tube Heat Exchanger

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ABSTRACT

In this research, the thermal attributes of shell and finned tube heat exchanger such as thermal efficiency, pressure drop, heat transfer rate and average temperature in the tube side of heat exchanger with using the different volume concentration of nanoparticles (SWCNT and Graphene quantum dot) at the various Reynolds number by applying either fin blades and without fin blades have been conducted numerically. In this heat exchanger the hot fluid or nanofluid flows in the tube section and cold fluid or pure water moves in the shell side. As regarding to results obtained the majority of thermal characteristics like heat transfer rate, pressure drop and effectiveness enhanced with augmentation of Reynolds number and increasing of volume concentration of nanofluids to 1% volumetric of working fluid whereas at the higher volume concentrations of nanoparticles (upper from 1% volumetric) the thermal properties of heat exchanger decreased generally. Also pressure drop intensifies with increment of Reynolds number and volume concentration of nanoparticles that at higher Reynolds number the effects of nanoparticles on the pressure drop were more noticeable. The average temperature of heat exchanger in the end section of inside tubes increased with augmentation of Reynolds number and nanoparticles. Finally, according to the results obtained in this study, most impression on the thermal attributes enhancement was found by employing of finned tubes compared to other factor which this factor increased heat transfer rate of heat exchanger by almost 188% also the effects of nanoparticles at the high levels of volume concentration especially for 5% of SWCNT nanoparticle on the pressure drop obtained about 80% compared to the base fluid.

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1. Introduction

Today, considering to the increasing consumption of fossil fuels and their pollution from one side and on the other hand with reduction of energy sources and increment costs of energy around the world, finding new methods for energy retrieval and preventing heat losses of equipment are needed. One of the most used devices in the industry is heat exchanger that has responsible to conduct heat transfer between hot and cold fluids via the solid wall. As regards the importance of reducing heat transfer time and higher effectiveness in the heat transfer systems, energy transfer and heat exchange are important problem in the heat exchangers which done with various ways between two fluids such as liquid-liquid [1], liquid-gas [2] and gas-gas [3]. There are various types of heat exchangers such as double tube heat exchangers [4], shell and tube heat exchangers [5], plate heat exchangers [6] and etc. among of overall heat exchangers, shell and tube heat exchangers more important rather than other due to their more heat transfer surface and lesser complex. Also for increasing heat transfer rate of heat exchangers especially shell and tube heat exchangers used different methods such as increment Reynolds number, increasing of heat transfer surface or using nanoparticles with various volume concentration which causes enhancement of base fluid thermal properties. One of the most important work that has been done in this field using together and simultaneous finned tube and nanoparticles in the base fluid which caused an augmentation thermal attributes such as thermal conductivity, density and viscosity [7-12]. According to this, several researchers have studied on the enhancement heat transfer rate and reduction of heat losses that the following is a summary of them.

Some studies have been performed on the thermal properties of nanofluids in the heat exchangers that all those studies have indicated the enhancement thermal attributes of base fluid with addition nanoparticles. Lee *et al.,* [13], Das *et al.,* [14], and Wang *et al.,* [15] studied on the thermal characteristics of working fluid by adding $\mathrm{Al}_2O_3$and CuO nanoparticles that observed improvement thermal attributes of base fluid. Also Pak and Cho [16] investigated on the convective heat transfer and heat transfer rate using TiO_2-water and Al_2O_3-water nanofluids which obtained results indicated by adding nanoparticles and augmentation Reynolds number, thermal characteristics of heat exchanger such as Nusselt number and heat transfer rate picked up. Ranjbarzadeh *et al.,* [17] carried out one empirical investigation on the forced convective of nanofluids that has obtained with augmentation volume concentration of nanoparticle and Reynolds number enhance Nusselt number of nanofluids about 51.4%. Anoop *et al.,* [18] measured thermal properties of base fluid by using SiO_2nanoparticles in shell and tube heat exchanger and resulted increasing heat transfer rate of heat exchanger at the different flow rates by using the various volume concentration of nanofluid. Farajollahi *et al.,* [19] dealt with experimental study on the heat transfer attributes of shell and tube heat exchanger by using TiO_2-water and γ-Al_2O_3-water nanofluids at the different volume concentration of nanoparticles that based on the obtained results, enhanced thermal properties of base fluid so that effects of TiO_2nanoparticle compared with other nanoparticle was better. Karimi *et al.,* [20] conducted an empirical assessment on the effectiveness of heat exchanger with adding MgO-$\mathrm{MWCNTs/EG}$nanofluid to the base fluid and results showed an enhancement about 20% on the thermal efficiency of heat exchanger. Elias *et al.,* [21] performed the study on the effectiveness of shell and tube heat exchanger with γ-Al_2O_3-water nanofluid under different baffle angles of heat exchanger that their experimental results indicted the best of baffle angle of setup for highest thermal efficiency was 20 degrees. Fotukian and Nasr esfahany [22] studied on the heat transfer characteristics of shell and tube heat exchanger with use CuO-water nanofluid that have obtained an enhancement about 25% for heat transfer coefficient. Shahrul *et al.,* [23,24] evaluated an empirical study on the thermal efficiency of shell and tube heat exchanger by using

TiO_2,Al_2O_3,Fe_3O_4,ZnOand CuO nanoparticles that obtained conclusion showed by increasing volume concentration nanoparticles of 0.01 to 0.04, thermal effectiveness of heat exchanger augmented approximately by 23-52%. Mehrali *et al.,* [25] measured a case study on the effectiveness of shell and tube heat exchanger with application of grapheme nanoplatelet nanoparticle in the base fluid and obtained by adding nanoparticle at the various volume concentration to the base fluid, thermal efficiency of heat exchanger improved about 83-200%. Vajjha *et al.,* [26] dealt with numerical investigation of two nanoparticles that include copper and aluminum oxide in the base fluid (ethylene glycol and water). According to their results, Nusselt number and heat transfer coefficient have enhanced with increasing volume concentration of nanofluid and Reynolds number so that effect of Reynolds number rather than volume concentration of nanofluid was greater. In addition, they found with augmentation of volume concentration both nanofluids, coefficient of friction surface increased compared to the base fluid. Peyghambarzade *et al.,* [27], Chavan and Pays [28] have performed in one numerical study on the impression adding of alumina nanoparticles with 1% volume concentration in the water and their obtained results showed increment 45% in heat transfer rate rather than pure water as base fluid. Choghol and Sahoo [29] have conducted an empirical study on the alumina nanoparticle at the 1% volume concentration and have resulted enhancement heat transfer rate amount 52.03%. Also Hussein *et al.,* [30] in the another research, have measured effect of TiO_2nanoparticles on the heat transfer rate and friction coefficient that their results have showed increasing two mentioned factor by raising of volume concentration nanoparticle 1-4% volumetric. Lv *et al.,* [31] have added copper nanoparticle with 5% volume concentration to the water and obtained augmentation approximately 45% of heat transfer rate compared to pure water. Also Naraki *et al.,* [32] conducted similar experiments on the copper oxide nanoparticles with 0.4% volume concentration of nanofluid and have indicated increasing about 8% of total heat transfer coefficient rather than pure water as base fluid. In addition, several experimentally studies such as Ho and Gao [33], Dhaidan *et al.,* [34], Fan and Khdadadi [35], and Wu *et al.,* [36] have confirmed using suspended nanoparticles as the enhancement process of thermal characteristics in the base fluid.

Regarding the recent investigation of scholars, major of studies have been conducted on the thermal attributes of heat exchangers based on metal nanoparticles that possess high thermal conductivity due to their inherent characteristics. Although remarkable development has been obtained in this field of nanoparticles but there are some problems in using this nanoparticle so that results have indicated the propagation of metal and metal oxide nanoparticles in the nature was caused concerns [37] that due to remaining metal or metal oxide combinations without any exchanges in the environment [38]. In the present study, was used of SWCNT-water and Graphene quantum dots-water nanofluid that are based on carbon nanoparticle and also used finned tubes at external section of heat exchanger simultaneously with mentioned nanoparticles and resulting thermal properties of shell and tube heat exchanger have been reported numerically.

2. Problem Statement and Methodology

2.1 Definition of Shell and Finned Tube Heat Exchanger

Designed heat exchanger in this research is a shell and finned tube heat exchanger that possess 21 finned tubes the internal of shell also this heat exchanger has not baffle segment. Figure 1 Geometry schematic of shell and finned tube heat exchanger.

The length and diameter of both shell and tube are 400 mm and 150 mm respectively. Figure 1 Represents the schematic of shell and finned tube heat exchanger that it should be noted hot fluid flows from center and exits from center whereas cold fluid enters from circumference and flows to out from perimeter as well as used nanofluid in heat exchanger (SWCNT and GQD) at the various volume concentration flows inside the tubes and in the shell, section flows pure water. Velocity of working fluid in the shell section is constant (Re=850) and inside the tube section is variable (Re=850-3300). Table 1 shows the geometry characteristics of shell and finned tube heat exchanger.

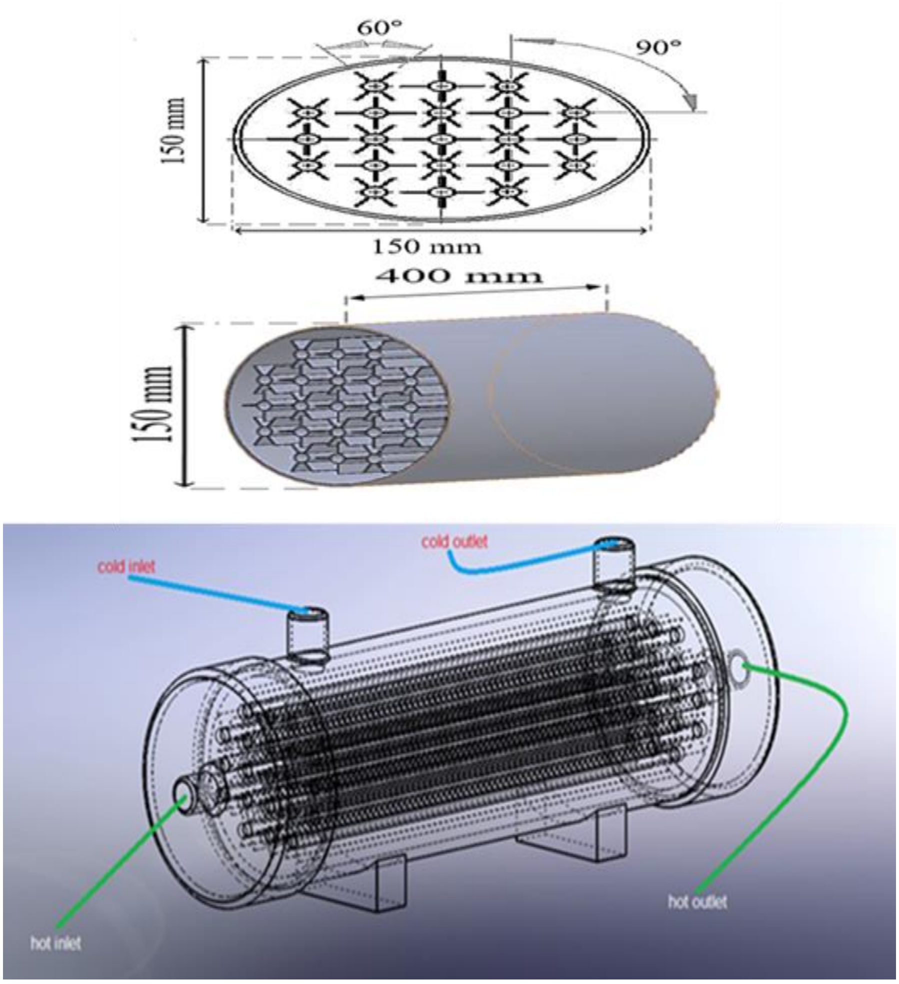

Fig. 1. Geometry schematic of shell and finned tube heat exchanger

Table 1
General characteristics of shell and finned tube heat exchanger

Parameters	Quantity
Shell inside diameter	150 mm
Shell thickness	2 mm
Shell length	400 mm
Number of baffles	-
Number of tubes	21
Tubes length	400 mm
Tubes arrangement pattern	Square
Tubes thickness	1 mm
Tubes outside diameter	10 mm
Fins length	400 mm
Fins thickness	1 mm
Fins width	1mm
Fins angle	60 deg
Fins angle	90 deg

2.2 Governing Equations

To evaluation of thermo-fluidic behaviour of the working fluid in shell and finned tube heat exchanger are used governing equations that include continuity, momentum and energy equations. Also problem suppositions are considered for working fluid to incompressible flow and Newtonian condition that are listed as following

Equation of continuity

$$\nabla.(\rho v) = 0 \tag{1}$$

Conservation of momentum

$$\nabla.(\rho\, vv) = -\nabla P + \nabla.(\mu\nabla\, v) \tag{2}$$

Energy equation

$$\nabla.(\rho\, vc_p\, T) = \nabla.(k\, \nabla\, T) \tag{3}$$

where ρ, k, P, T, μ and c_p indicate density, thermal conductivity, pressure, temperature, dynamic viscosity and specific heat, respectively.

2.3 Thermophysical Properties of Nanofluid

As regards thermophysical attributes nanoparticles and base fluid (water) is different thus for computing of mentioned properties of nanofluid should be used from new equations that included both their thermal characteristics. For this reason, used from following equations to determine thermal properties of nanofluids.

For determine the thermal conductivity of nanofluids is used Maxwell equation [39] accordance follows

$$\frac{k_{nf}}{k_{bf}} = \frac{k_{np}+2k_{bf}+2\varphi(k_{np}-k_{bf})}{k_{np}+2k_{bf}-\varphi(k_{np}-k_{bf})} \tag{4}$$

where k_{nf}, k_{bf}, k_{np} and φ are thermal conductivity of nanofluid, thermal conductivity of base fluid, thermal conductivity of nanoparticle and volume concentration of nanofluid, respectively. As well as for calculation of nanofluid density used Pak and Cho model [40] as below

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_{np} \tag{5}$$

where $\rho_{nf}, \rho_{bf}, \rho_{np}$ are density of nanofluid, base fluid and nanoparticles, respectively. To assessment the specific heat capacity of nanofluid from Xuan and Roetzel model [41] as follows was used

$$Cp_{nf} = \frac{(1-\varphi)\rho.Cp_{bf}+\varphi Cp_{np}}{\rho_{nf}} \tag{6}$$

where $cp_{nf}, cp_{bf}, cp_{np}$ represent specific heat capacity of nanofluid, base fluid and nanoparticles, respectively. Viscosity of nanofluid was gained from Einstein equation [42] as below

$$\mu_{nf} = (1 + 2.5\varphi)\mu_{bf} \tag{7}$$

where μ_{nf}, μ_{bf} and φ are viscosity of nanofluid, base fluid and volume concentration of nanofluid, respectively.

2.4 Boundary Condition

To prevent heat losses on the outside of the heat exchanger shell is used from insulation until heat losses decreased from inside heat exchanger to the out. Also for simulation of heat exchanger was assumed that temperature in the both section of shell and finned tube heat exchanger uniformly and temperature variations have been considered negligible. Furthermore, in the outlet boundary of cold and hot fluid of heat exchanger is employed zero relative pressure as well as was used no-slip condition at the walls of the shell and finned tube heat exchanger.

2.5 Numerical Method and Validation

For the numerical simulation of heat transfer studies use from different software such as ANSYS FLUENT, ANSYS CFX, COMSOL and etc. [43-46]. In current research, was used ANSYS FLUENT 19 for simulation and modeling of heat transfer problems related to the shell and finned tube heat exchanger. Also to discretization the governing equations was used Finite-volume method (FVM) with second-order upwind as well as semi-implicit method for pressure linked equations (SIMPLE) have been applied for pressure-velocity coupling with using PRESTO scheme for pressure correction. As regarding volume variation in the shell and finned tube heat exchanger was approximately equal to zero, was used PRESSURE BASED option for solving the conservation equations. In this study, the desired residual was determined for momentum and continuity equation equal to 10^{-4} and for the energy equation equal to 10^{-6} until end of the solving of governing equations. Further to evaluation of grid independency has been used the various grids that including different cell numbers with N=4,211,012, 6,514,362, 8,421,635, 10,145,774 and 12,329,104 which as regards obtained results, the best of grid systems for shell and finned tube heat exchanger model was found the grid system of N=10,145,774 so that by using this cell number was not seen remarkable change in the results of hot fluid temperature at the outlet section of finned tube as well as cold fluid temperature at the outlet region of shell for Reynolds number equal to 850 and inlet temperature of working fluid at the shell and finned tube equal to 293 K and 253 K, respectively. In fact, Table 2 shows with augmentation mesh density from 10,145,774 cells to 12,329,104 cells just computational cost have been increased and noticeable alteration in the results have not happened and this i.e. insensitivity of results to the grids.

Table 2
Gained results of various grid study on the heat exchanger model

N	Hot fluid temperature of finned tube (K)	Cold fluid temperature of shell (K)
4,211,012	333	309
6,514,362	336	311
8,421,635	335	312.5
10,145,774	335.5	313
12,329,104	335.7	313.1

Also for validation of numerical method used in this study, results gained of simulation the shell and finned tube heat exchanger compared with results of experimental conducted in this research. For this work, desired experiments have performed under conditions of various Reynolds number with pure water at the shell and finned tube region and conclusions of heat transfer rate obtained from experimental and simulation methods are listed in Table 3. Results gained of Table 3 represented that experimental results comparison with numerical conclusions for heat transfer rate characteristic there was no noticeable difference and applied model with desired grid system valid for further simulation of heat exchanger.

Table 3
The comparison results obtained from experimental and simulation methods for different Reynolds number

Re	U (w) Simulation	U (w) Experiment	Error (%)
850	412	430	4
127	493	511	3.5
1700	574	583	1.5
2500	1316	1310	.4
3300	2058	2073	.72

2.6 Data Processing

For calculations of heat transfer equations such as heat transfer rate, overall and inside-tube heat transfer coefficient, Nusselt number under conditions laminar and turbulent regime, pressure drop and effectiveness of heat exchanger with various volume concentration of nanofluids was used from following equations that entire these equations was obtained from ref [47,48]. Heat transfer rate of inside-tube was gained from following equation

$$Q = \dot{m}.Cp_h(T_{hi} - T_{ho}) \tag{8}$$

where $\dot{m}, cp_h, T_{hi}, T_{ho}$ represent mass flow rate, specific heat capacity of hot working fluid, temperature of hot fluid at inlet and outlet section, respectively. Also, for calculating of overall heat transfer coefficient was employed from Eq. (9) as below [49]

$$U = \frac{Q}{A_{t,f}.\Delta T_{lm}} \tag{9}$$

where heat transfer rate (Q)gained from Eq. (8) and $A_{t,f}, \Delta T_{lm}$were calculated from Eqs. (10) and (13), respectively.

$$A_{t,f} = A_t + A_f \tag{10}$$

where $A_t, A_f$are tube area and fin area that obtained from Eqs. (11) and (12), respectively.

$$A_t = \pi L d_o N_t \tag{11}$$

$$A_f = (2\,Lw)\,N_t \tag{12}$$

where L, d_o, N_t and w represent tube length, outer diameter of tubes, number of tubes and fin width, respectively. Moreover ΔT_{lm} has been showed as Eq. (13)

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{ln\frac{\Delta T_2}{\Delta T_1}} \tag{13}$$

So that for counter current inside the shell and tube was applied Eq. (14) as well as for agree current of shell and tube was used Eq. (15) as follows

$$\Delta T_2 = T_{hi} - T_{co}, \Delta T_1 = T_{ho} - T_{ci} \tag{14}$$

$$\Delta T_2 = T_{hi} - T_{ci}, \Delta T_1 = T_{ho} - T_{co} \tag{15}$$

Convective heat transfer coefficient gained from following equation that related to Dittus-Boelter correlation [50]

$$h_i = \frac{Nu_i k_{nf}}{d_i} \tag{16}$$

where for calculation of Nusselt number under conditions laminar and turbulent regime was used following equations [51,52], respectively.

$$Nu_i = 3.66 + \frac{0.0668(d_i/L)RePr}{1+0.4[(d_i/L)\,RePr]^{2/3}} \tag{17}$$

$$Nu_i = 0.037(Re^{0.75} - 180)\,Pr^{0.42}[\,1 + (d_i/L)^{2/3}](\frac{\mu_b}{\mu_w})^{.14} \tag{18}$$

where $Nu_i, k_{nf}, \mu_b, \mu_w$ are Nusselt number inside the tube, thermal conductivity of the nanofluid, viscosity of base working fluid and viscosity of near wall, respectively. As well as Reynolds number and Prantle number were calculated from Eqs. (19) and (20), respectively [49].

$$Re = \frac{\rho v d_i}{\mu} \tag{19}$$

$$Pr = \frac{Cp\mu}{k} \tag{20}$$

For obtain the pressure drop in heat exchanger used from Darcy-Weisbach equation [53] that friction coefficient (λ) was gained from Darcy Eq. (22) for laminar regime and Blasius Eq. (23) for turbulent current as follows

$$\Delta P = L\lambda\frac{\rho v^2}{2d_i} \tag{21}$$

$$\lambda = \frac{64}{Re} \tag{22}$$

$$\lambda = \frac{0.3164}{\sqrt[4]{Re}} \tag{23}$$

Also, thermal efficiency (ε) and NTU index have been determined from the below equations

$$\varepsilon = f\ (\mathrm{NTU}, \underbrace{C_{\min} / C_{\max}}_{C_r}) \tag{24}$$

$$\varepsilon_{parallel-flow} = \frac{1 - exp[-NTU(1 + C_r)]}{1 + C_r}$$
$$\varepsilon_{counter-flow} = \frac{1 - exp[-NTU(1 + C_r)]}{1 + C_r\, exp[NTU(1 - C_r)]} \tag{25}$$

So that

$$NTU = \frac{UA}{C_{min}} \tag{26}$$

and

$$C_r = \frac{C_{min}}{C_{max} = \frac{(\dot{m}cp)_{min}}{(\dot{m}cp)_{max}}} \tag{27}$$

As regards in current research amount flow rate of hot fluid lesser than cold fluid, thus cp, h_{min} so that $c_{p,h}$ is specific heat capacity of hot working fluid in the shell and finned tube heat exchanger. Also, Table 4 represents thermophysical properties of nanofluids used in the shell and fined tube heat exchanger as follows

Table 4
Thermophysical properties of nanofluids

Nanoparticle Density (Kg.m^{-3})	Thermal conductivity (W.m^{-1}.k^{-1})	Specific heat (J.kg^{-1}.k^{-1})
SWCNT 2600 [54]	6600 [54]	425
Gqd$_s$ 400	3000	643

3. Results and Discussion
3.1 Average Temperature and Eddy Viscosity Analysis

In this study, thermal properties of desired shell and finned tube heat exchanger such as effectiveness, heat transfer rate, overall heat transfer coefficient and pressure drop by using the two nanoparticles (SWCNT and Graphene quantum dots) are examined so that hot fluid that included nanofluids, flows at the inside tubes and cold working fluid that involved pure water moves at the shell section. It should be noted that Reynolds number in the shell side was considered constant and equal to 850 whilst it was variable from 850 to 3300 at the inside tube section. Inlet temperature of inlets was equal to 353 K and 293 K for the tube and shell sections, respectively. Tow nanoparticles SWCNT and Graphene quantum dots with different volume concentration $w = 1\%, 3\%, 5\%$of volumetric with various Reynolds number was applied in this study that was employed in the tube side section. As a conclusion, a countercurrent flow was employed in the shell and finned tube heat exchanger and results have been showed in the contour plots as numerically. Figure 2 represents temperature contour plots of heat exchanger for different Reynolds number from 850 to 3300 at the tube section and constant Reynolds number equal to 850 for the shell section with using pure water at the two sides of heat exchanger and without fin configuration. As can be observed in Figure 2, outlet temperature of hot fluid has been increased with increment Reynolds number from 850 to 3300 at the tube section as follows.

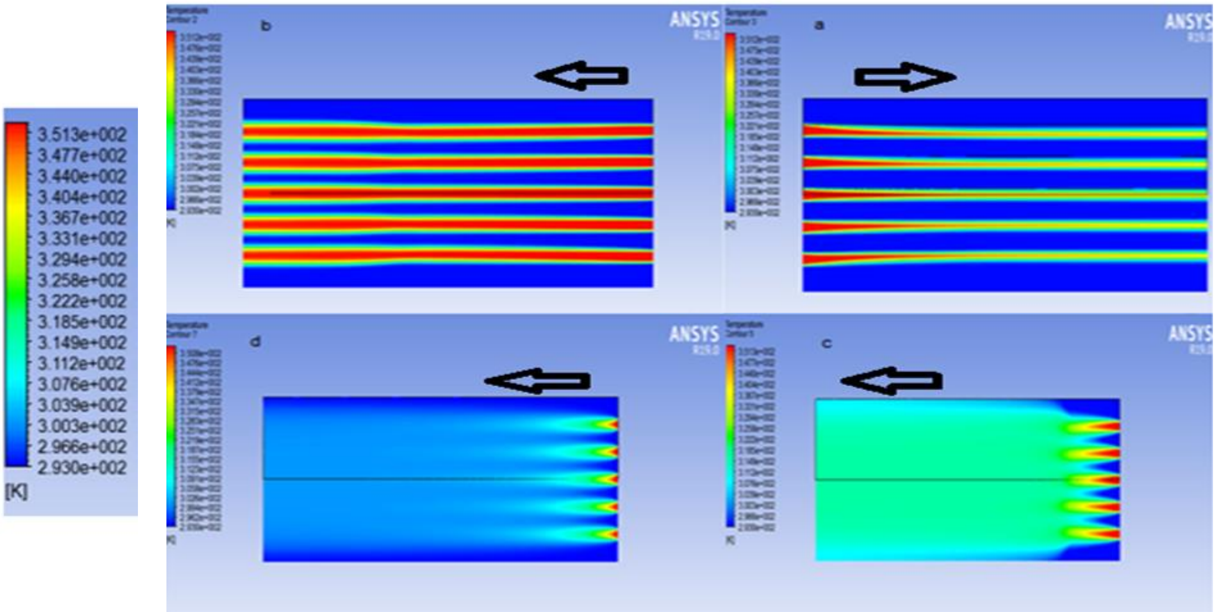

Fig. 2. Temperature contour plots: Reynolds number equal to: (b) 1700, (a) 850, (c) 3300 at tube section and (d) 3300 at the two sides of heat exchanger

In fact, Figure2 represents by increasing Reynolds number from 850 to 3300 at the tube side, outlet temperature of tube side declined to a greater amount which this is because the thickness of thermal boundary layer has been decreased further which caused convective heat transfer rate from tube to the out increases also due to more gradient temperature between inlet and outlet of tube, heat transfer rate the heat exchanger enhances. Moreover Figure 3 demonstrate temperature contour plots in the presence of finned tubes that according to the results obtained by using of finned tubes in the heat exchanger, outlet temperature at the tube section has been dropped significantly that was due to increasing of heat transfer surface at outer section of tubes. Also it can be observed with augmentation of Reynolds number from 850 to 3300 in the tubes as well as in the shell, outlet temperature of tubes has improved remarkably that it is because decreasing thickness of thermal boundary layer and increasing convective heat transfer coefficient in the tube section. Also it can be seen by adding of nanoparticles (SWCNT and GQD) at the different volume concentration to base fluid (pure water) under condition absence of fines enhanced outlet temperature in the tube side rather than without using of nanoparticles that was due to greater thermal conductivity of solid particles rather than base fluid. As can be observed in Figure 4, with augmentation volume concentration of SWCNT nanoparticle from 1% to 5% volumetric have been increased the outlet temperature in the tube section that it is because with addition volume concentration of nanofluids from 1% to 5% have been augmented viscosity and density of hot nanofluid. Indeed, increasing volume concentration of SWCNT nanoparticle augmented greatly conductive heat transfer of nanofluid that caused convective heat transfer of nanofluid to be severely suppressed and consequently heat transfer rate of heat exchanger from inside tubes to the out was decreased while the best volume concentration of GQD nanoparticle considering to the numerical and experimental results was found value of 3% volumetric since thermal conductivity and density of GQD nanoparticle related to SWCNT nanoparticle is lower and for improvement outlet temperature in the tube side need to higher percent of GQD nanoparticle. Also effect the application of SWCNT and GQD

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nanoparticles and finned tube option have been investigated simultaneously which results obtained from numerical and experimental methods indicated an improvement the outlet temperature of tube side that in the following contour plots has been showed.

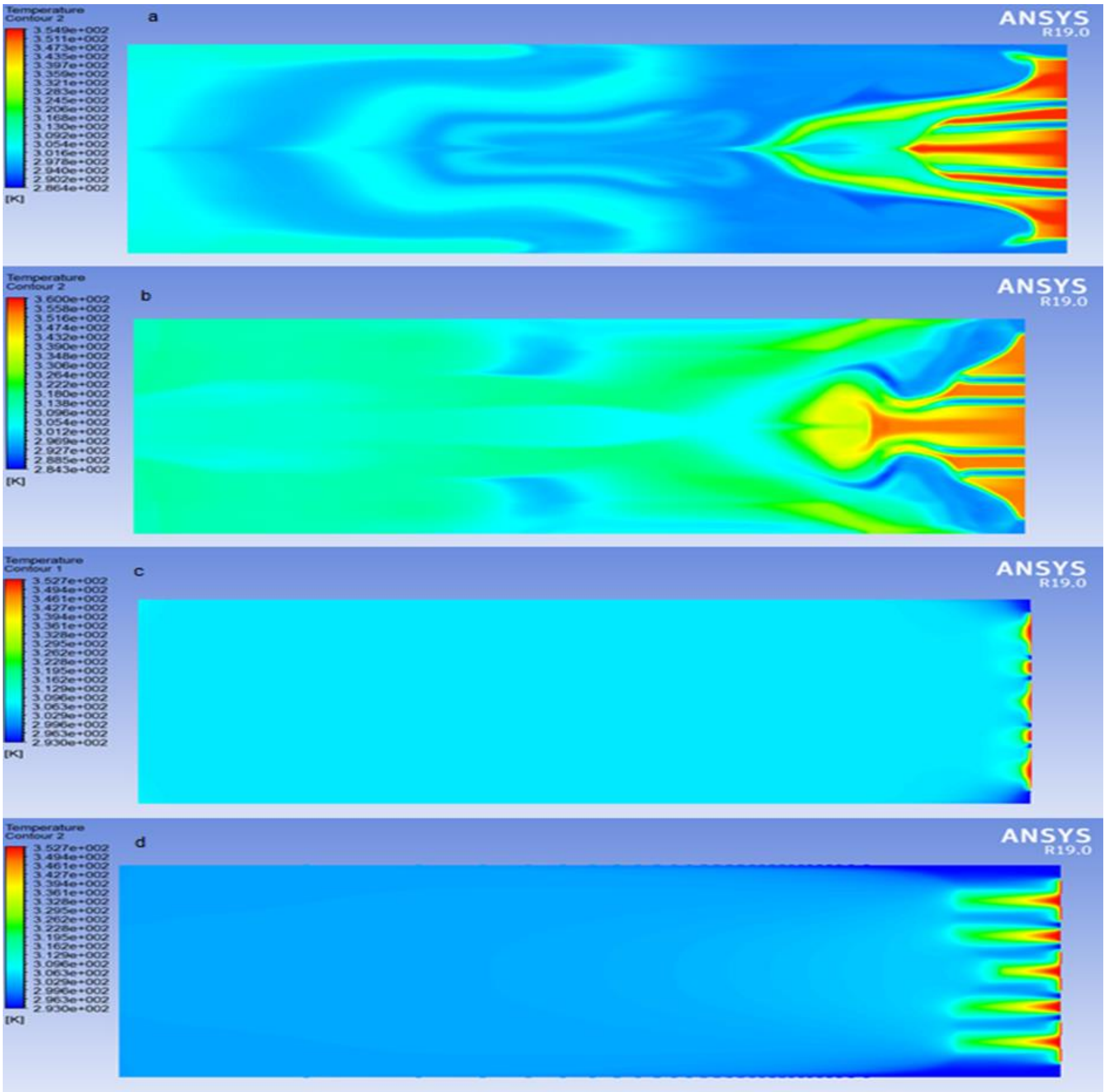


Fig. 3. Temperature contour plots of heat exchanger in the presence of finned tube at the various Reynolds number: (a) 850, (b) 1700, (c) 3300 at the tube section and (d) 3300 at the two sides of heat exchanger (shell and tubes together)

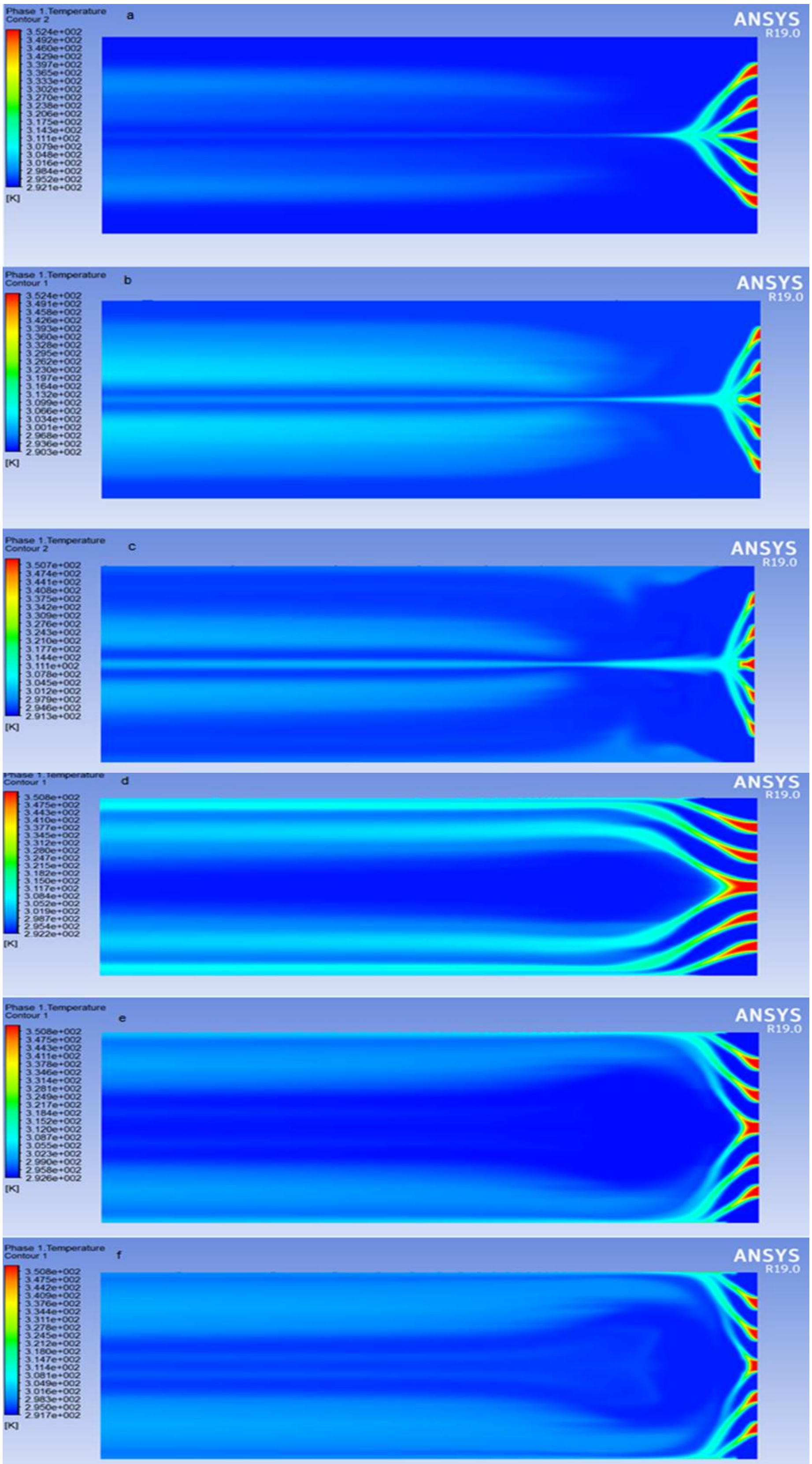


Fig. 4. Temperature contour plots of shell and tube heat exchanger with: (a) 1% SWCNT, (b) 3% SWCNT, (c) 5% SWCNT, (d) 1% GQD, (e) 3% GQD and (f) 5% GQD volume concentration of nanoparticles at the Reynolds number equal to 850 in the two sides (shell and tubes together)

As can be observed in Figure 5, the best combination of shell and tube heat exchanger for highest thermal properties and outlet temperature in the tube and shell side was found amount of 1% volume concentration of SWCNT nanoparticle along with finned tubes in the internal section of heat exchanger due to with this combination outlet temperature in the tube section was perceived to lowest its amount that causes highest thermal efficiency as well as it should be noted that increment of the volume concentration of nanoparticle increase friction losses between wall and nanofluids moreover causes augmentation pressure drop in the tube section and consequently, decrease thermal efficiency of heat exchanger. As regards calculation gained from numerical methods, the contour plots of (a) for SWCNT nanoparticle and contour plots of (e) for GQD nanoparticle are the best options for enhancement thermal attributes of shell and tube heat exchanger. Generally by increasing volume concentration of nanoparticles in the base fluid in this study was decreased the thermal properties of heat exchanger that it is because with addition of nanoparticles at the high levels, natural convective heat transfer suppressed by great conductive heat transfer due to high concentration of nanoparticles furthermore high percentages of nanoparticles increase pressure drop in the tube suddenly and incline effectiveness of heat exchanger with more intensely. Also considering to results obtained from temperature of nanoparticle surface could be found the best of volume concentration of nanoparticle as follows.

As can be seen in Figure 6 temperature of SWCNT nanoparticle at the 1% volume concentration more decreased related to the other volume concentration which it is because at the 1% volume concentration of SWCNT nanoparticle was found a balance between convective heat transfer and conductive heat transfer so that was caused highest desired thermal efficiency of heat exchanger also corresponding balance between two thermal factors obtained for 3% volume concentration of GQD nanoparticle. Moreover using the SWCNT nanoparticle rather than GQD nanoparticle had more effect on the outlet temperature at the low volume concentration that was due to greater its thermal conductivity related to the GQD nanoparticle but at the higher volume concentration, effect the GQD nanoparticle rather than the SWCNT nanoparticle was more that it is because density and pressure drop of GQD nanoparticle was lower related to the SWCNT nanoparticle also as regarding lower thermal conductivity of GQD nanoparticle related to the SWCNT nanoparticle consequently, natural convective heat transfer of heat exchanger enhanced and effectiveness of heat exchanger was better. On the other hand, the effect of turbulence eddy viscosity under condition turbulent flow equal to 3300 between two situations of turbulent regime only in the tube and turbulent regime in the tube and shell together have been studied and have been observed by applying turbulence current in the two sides of heat exchanger, turbulence eddy viscosity in the middle of the heat exchanger is increased that caused average temperature in the middle section enhances and consequently, thermal efficiency of heat exchanger increases as follows.

As can be seen in Figure 7 the average temperature in contour of (b) is lower related to contour of (a) that due to turbulence eddy viscosity in contour (b) at the middle section of heat exchanger represents a greater extent (red region) related to the other contour. For this reason, by applying tow turbulence regimes in the tubes and shell more effect on the average temperature and thermal efficiency of heat exchanger.

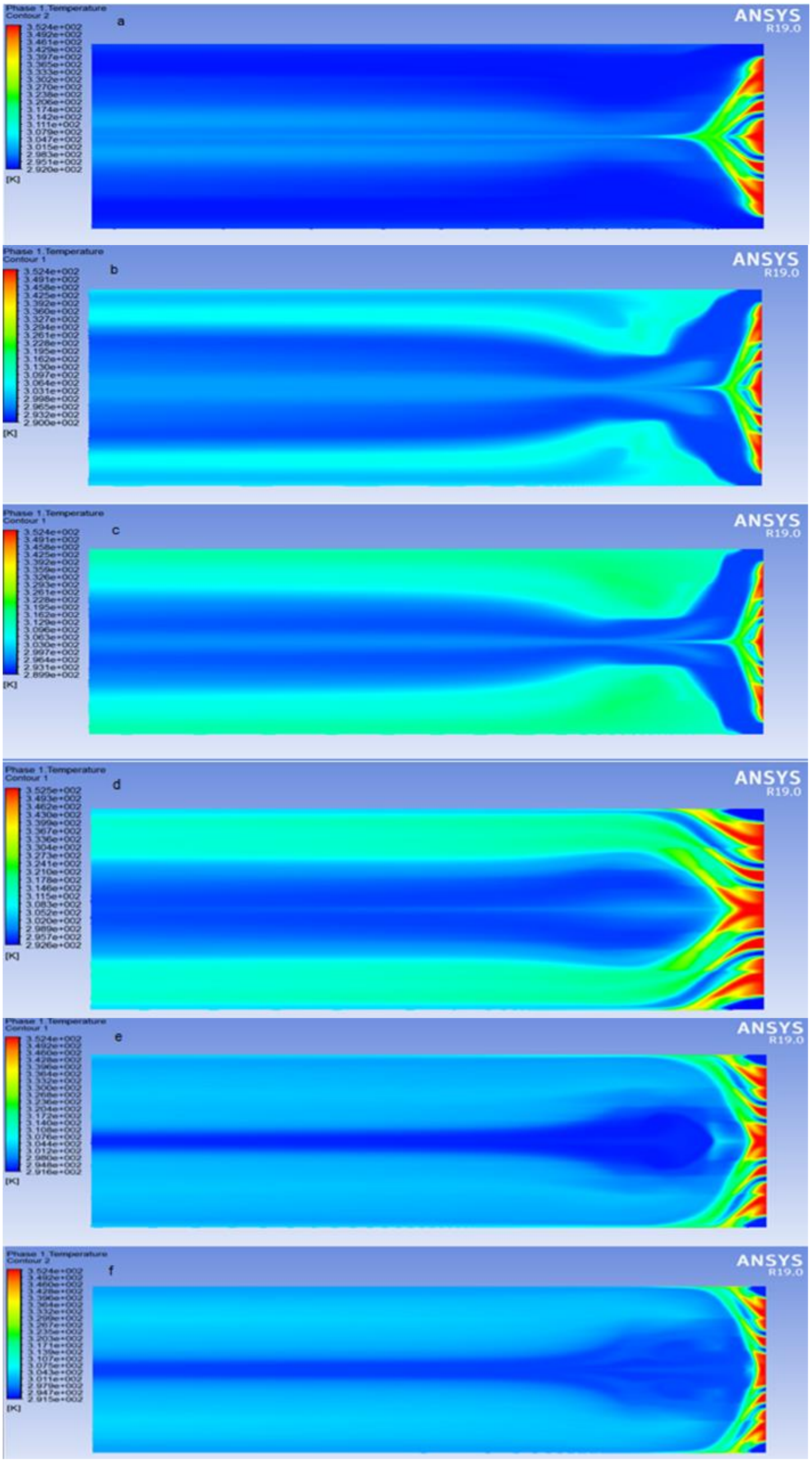


Fig. 5. Temperature contour plots for (a) 1%, (b) 3% and (c) 5% of SWCNT nanoparticle as well as (d) 1%, (e) 3% and (f) 5% of GQD nanoparticle under condition using of finned tubes and invariant Reynolds number equal to 850

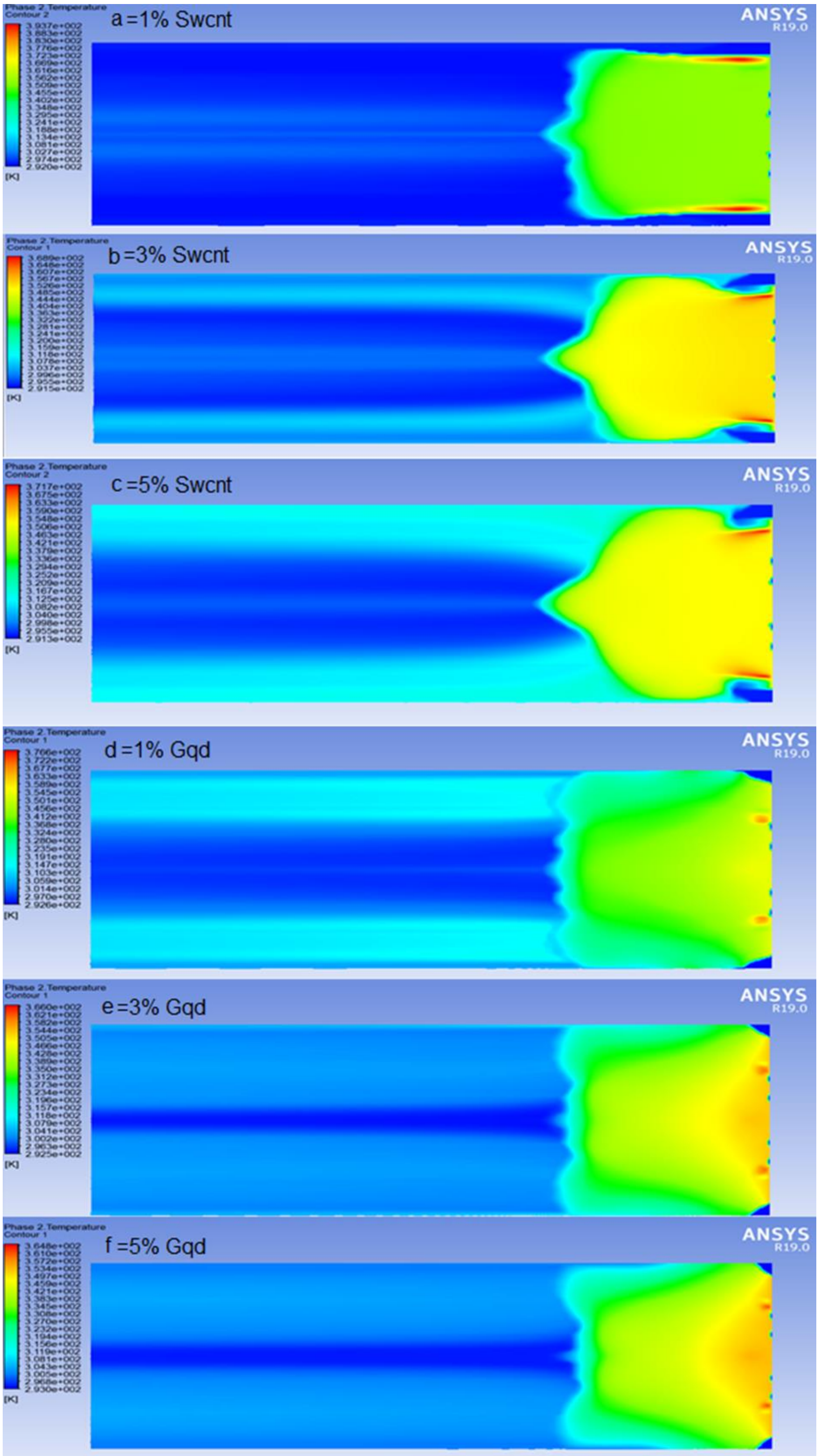


Fig. 6. Temperature contour plots of nanoparticles at Reynolds number equal to 850

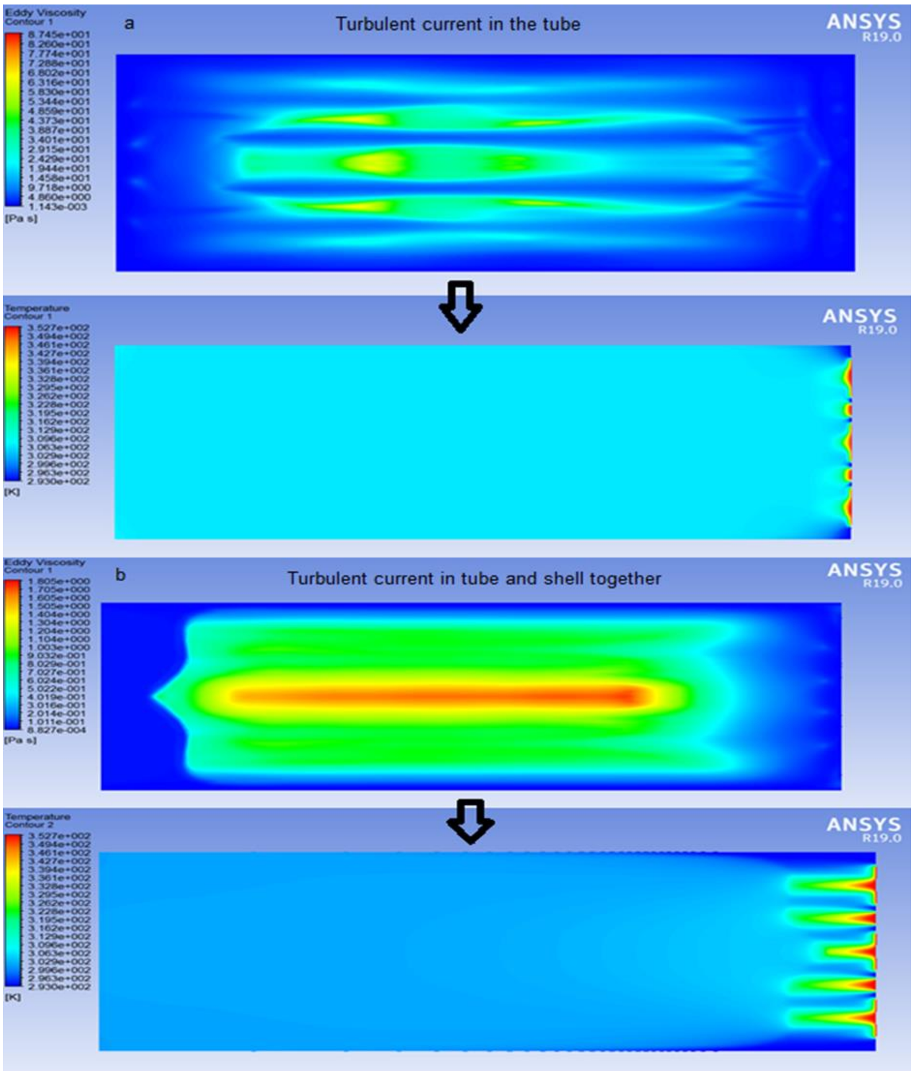
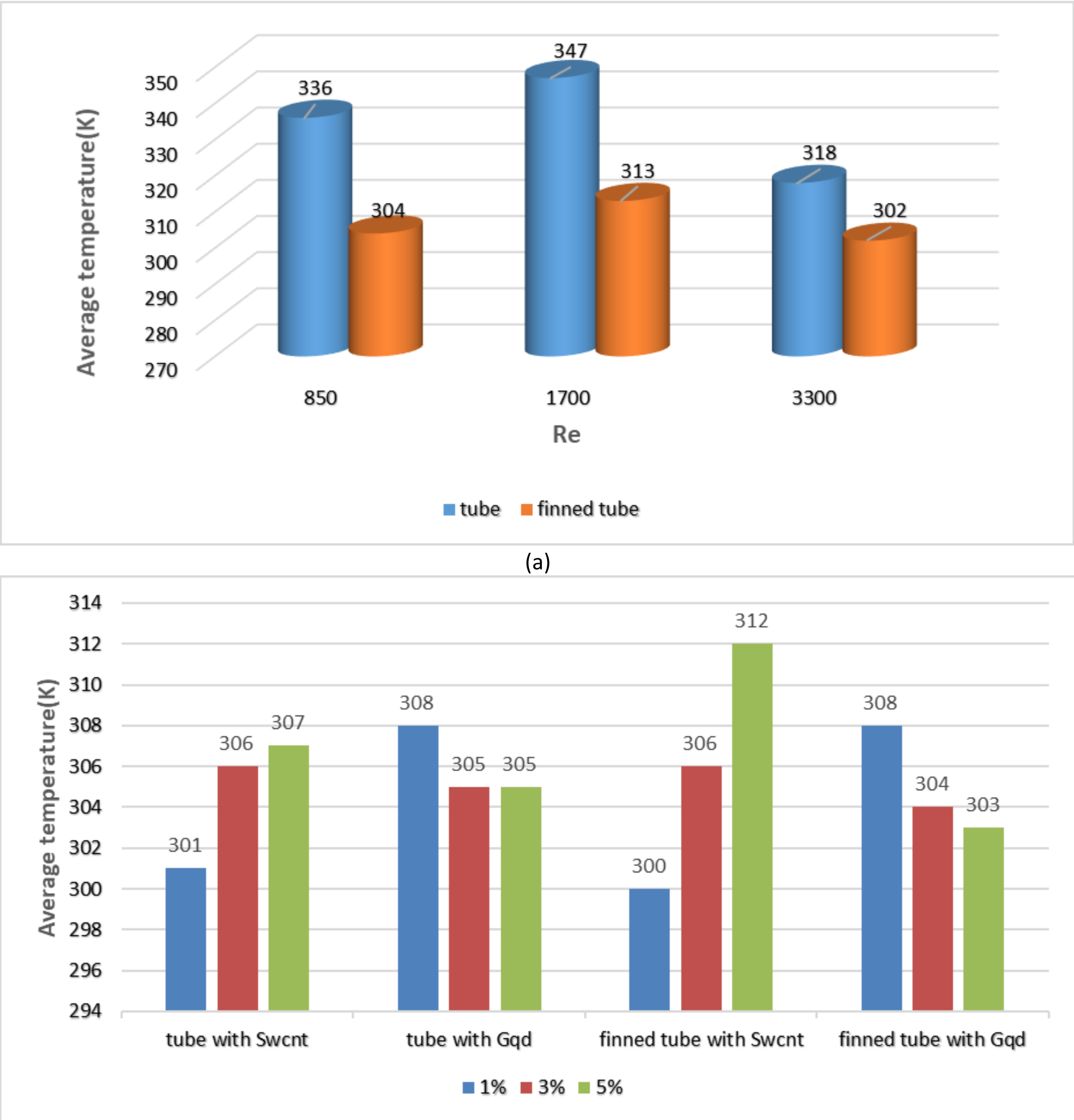


Fig. 7. The effect of turbulence eddy viscosity on the average temperature in the shell and tube heat exchanger at Reynolds number equal to 3300 for (a) in the tube and (b) in the tube and shell together

Figure 8 depicts the average temperature in the tubes side of shell and finned tube heat exchanger against of Reynolds number and different volume concentration for nanofluid and base fluid. It is obvious that with augmentation value of Reynolds number from 850 to 1700, is increased the average temperature is which due to low residence time of hot fluid in the heat exchanger whereas with increment the amount of Reynolds number from 1700 to 3300 and enters to the turbulence range, the average temperature is decreased that it is because turbulence eddy viscosity and turbulence kinetic energy have been augmented which those reduced thickness of boundary layer that consequently, improved average temperature of heat exchanger. On the one hand, by applying nanoparticles in the heat exchanger generally average temperature enhanced that due to nanoparticles possess great thermal conductivity than base fluid but on the other hand, with increment volume concentration of SWCNT nanoparticle from 1% to 5% volumetric of fluid the average temperature of hot fluid augmented which it is because by increasing volume concentration of SWCNT nanoparticle from 1% to 5% volumetric the convective heat transfer rate suppresses with the large increment in conductivity heat transfer is that resulting the augmentation volume concentration of nanoparticles. Also average temperature the GQD nanoparticle by increasing volume concentration of nanoparticle from 1% to 5% volumetric is decreased since the its conductivity heat transfer is lower than SWCNT nanoparticle and more positive influence on the heat transport from tubes to the out.



(b)

Fig. 8. The impression Reynolds number on the average temperature (top figure) (b) volume concentration of nanoparticle on the average temperature

3.2 Heat Transfer Rate

Figure 9 represents the heat transfer rate of shell and tube heat exchanger against of various Reynolds number with different volume concentration of SWCNT and GQD nanoparticles both with fin blades and without fin blades that based on results gained the heat transfer rate generally improved with augmentation of Reynolds number and volume concentration of nanoparticles so that the best of volume concentration of nanoparticle was found amount of 1% SWCNT nanoparticle volumetric of fluid since at the higher volume concentration of nanoparticles the conductive heat transfer suppresses the convective heat transfer and considering to greater thermal conductivity of

SWCNT nanoparticle comparison with GQD nanoparticle, its effect on the average temperature of heat exchanger is more. Also it should be noticed by using of finned tube the heat transfer rate enhanced by 188%.

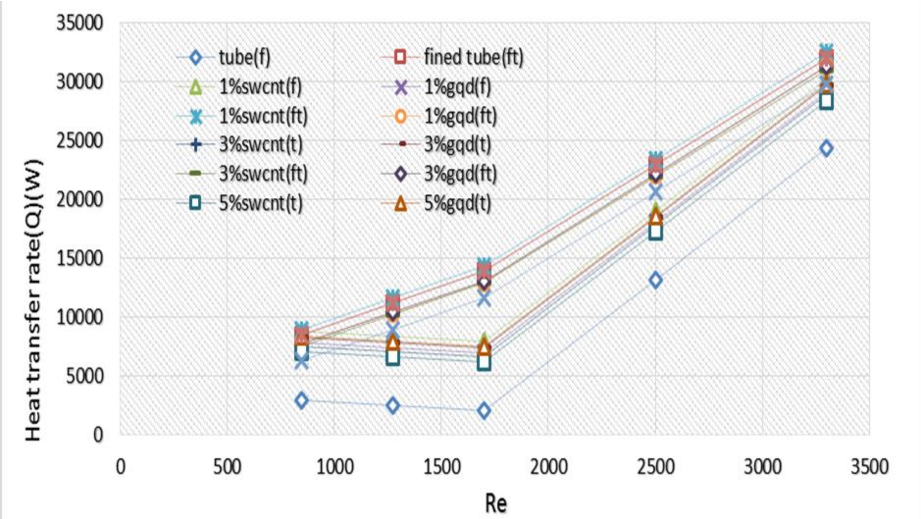


Fig. 9. Comparison of heat transfer rate at different Reynolds number and nanoparticles

3.3 Pressure Drop

As shown in Figure 10 the pressure drop of hot working fluid with augmentation Reynolds number and volume concentration of nanoparticle increased that the main reasons are increment friction factor between nanofluid and tube wall as well as solid nanoparticles with together, increasing the turbulence eddy viscosity of near wall in the tube side and augmentation density and viscosity of nanofluid. As regarding results obtained from experimental and numerical methods the highest relevant increasing of pressure drop was found for 5% SWCNT nanoparticle by about 80%.

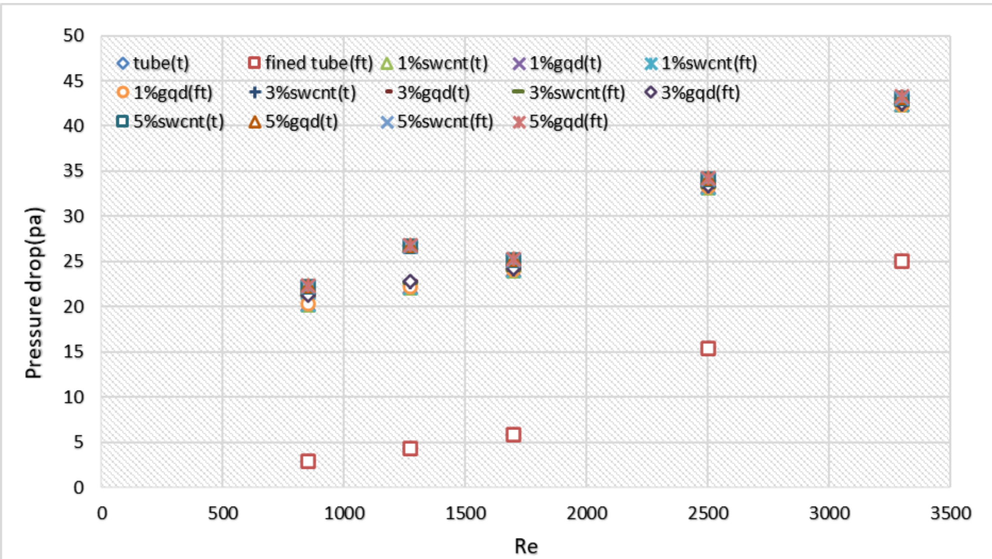


Fig. 10. The pressure drop of heat exchanger under various Reynolds number and volume concentration of nanoparticles

3.4 Effectiveness and Response Surface Method

As regarding calculations obtained from numerical method the effectiveness of shell and finned tube heat exchanger at the various Reynolds number and volume concentration of nanoparticles increased with increment the volume concentration of SWCNT and GQD nanoparticles and applying the finned tubes. It should be considered by increasing the volume concentration of nanoparticles from 1% to 5% volumetric of fluid the effectiveness has no significant enhancement that due to high pressure drop and reduced convective heat transfer which eliminate the positive influence of great thermal conductivity of nanoparticle at the high volume concentration. Moreover by increasing the Reynolds number the thermal efficiency decreased initially that the its main reason is low residence time of hot fluid in the heat exchanger but while hot fluid enters to the turbulence range the effectiveness of hot fluid increased that its goal is high turbulence eddy viscosity and kinetic energy which caused the thickness of boundary layer in the tubes decrease and heat transfer from inside of tubes to the out increase. As can be seen in Figure 11, the thermal efficiency of heat exchanger under different condition of Reynolds number and various volume concentration of nanoparticles has been augmented with different volume concentration of SWCNT and GQD nanoparticles along with finned tubes option by about 100% and 85%, respectively so that as regarding results obtained the influence of finned tubes was very more than the nanoparticles.

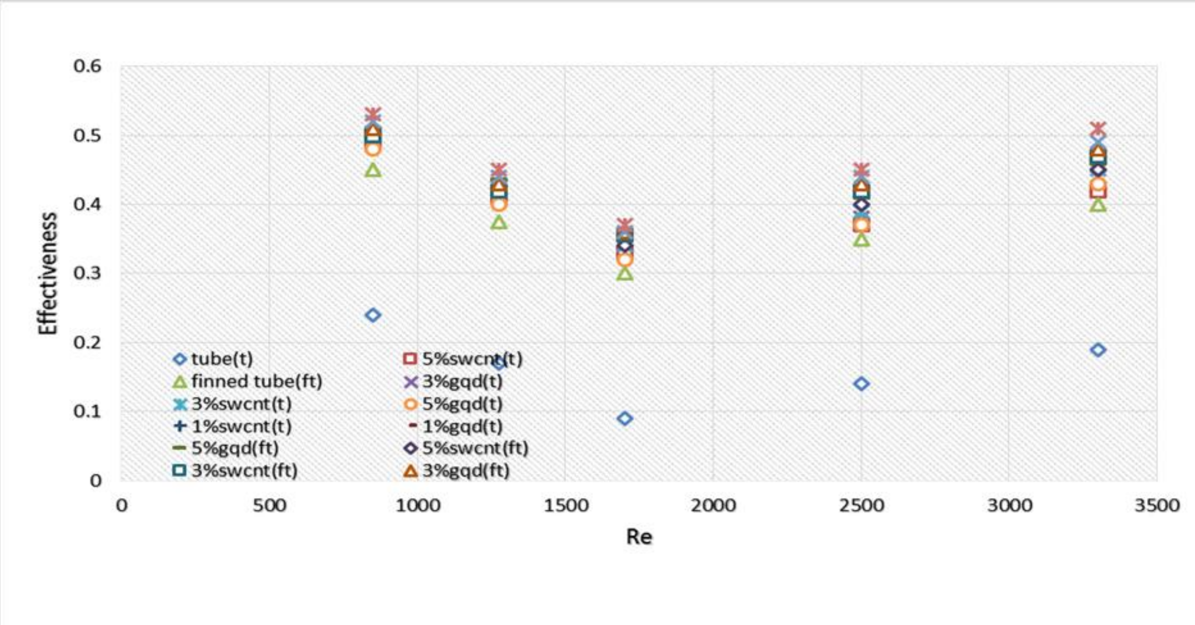
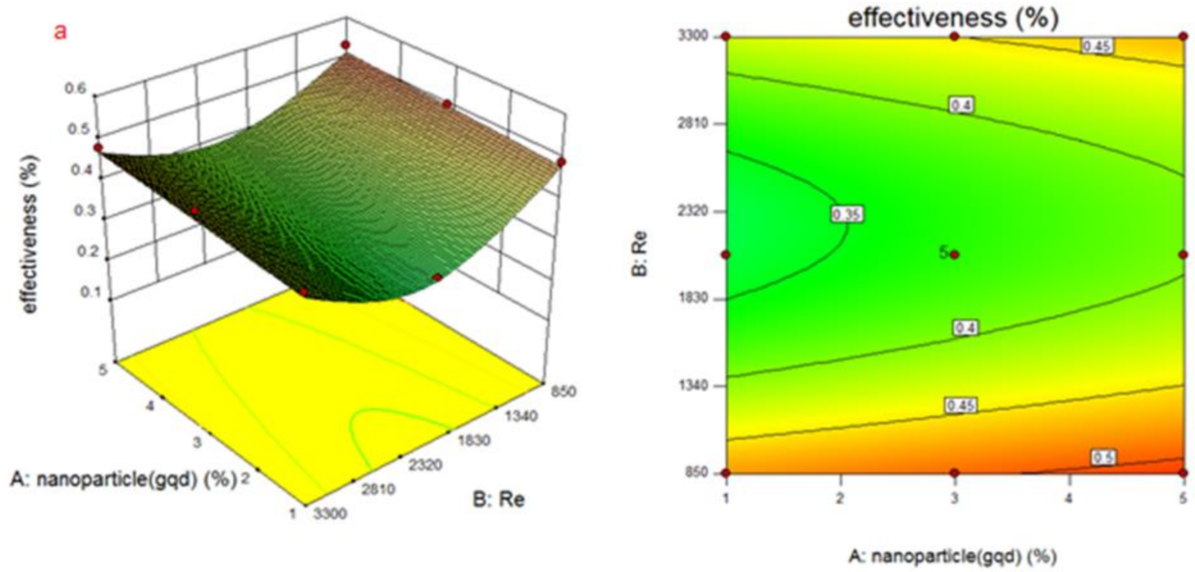
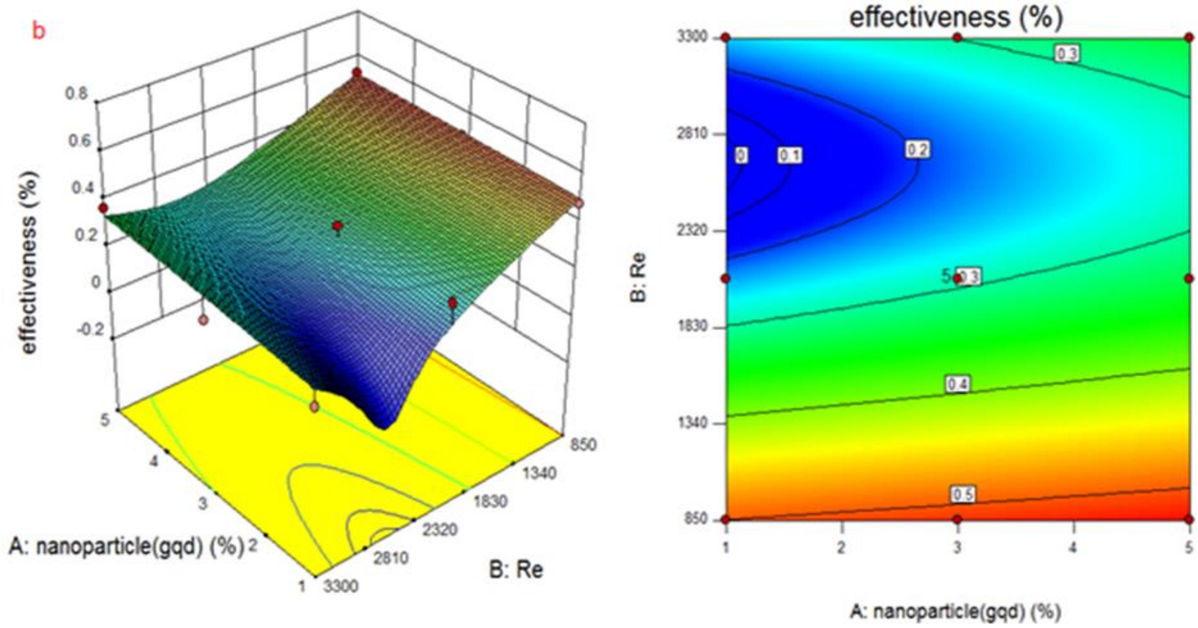


Fig. 11. The effectiveness of heat exchanger under different volume concentration of nanoparticles and Reynolds number

In this section, the thermal efficiency of heat exchanger is illustrated by using RSM software of two representing situation (three-dimensional surface and two-dimensional contour plots) that have been showed in Figure 12 as follows.



(a)



(b)

Fig. 12. (a) Three-dimensional and contour plots the effectiveness of heat exchanger without finned tube and (b) with finned tube

As can be seen in Figure 12 with increasing volume concentration of nanoparticles from 1% to 5% volumetric of fluid the effectiveness augmented while this increasing was no significant because by augmentation the volume concentration of nanoparticles in the base fluid the pressure drop and convective heat transfer decrease that its result is decreasing of thermal efficiency also with increment value of Reynolds number the thermal efficiency of heat exchanger initially decreased and then increased that due to by increasing the Reynolds number to below the turbulence range the time residence of hot fluid in the heat exchanger decreases and hot fluid has no remarkable time for heat transfer with cold fluid whereas by augmenting the Reynolds number from 2300 to 3300 and enters the turbulence range, the increasing of turbulence eddy viscosity and kinetic energy have been represented its influence that it reduces the thickness of boundary layer and consequently, thermal efficiency of heat exchanger increases.

4. Conclusion

In the present research, thermal attributes such as average temperature, heat transfer rate, pressure drop and effectiveness under using the both finned tube and without finned tube as well as different volume concentration($\omega = 1\%$, 3%, 5% of SWCNT and Graphene quantum dot) the within countercurrent shell and tube heat exchanger are examined and measured and following results were gained

I. As regarding to results obtained in this study, generally by increasing the volume concentration of nanoparticles and Reynolds number the thermal properties were augmented so that effect of Reynolds number was variable and volume concentration of nanoparticles from 1% to the 5% volumetric of working fluid were didn't noticeable.
II. Using the finned tubes in the shell and tube heat exchanger have been increased thermal attributes since using the finned tubes caused a higher contact surface between cold fluid and hot fluid and consequently, enhanced the thermal efficiency of heat exchanger.
III. Heat transfer rate have been increased by applying the finned tubes by about 188% compared with without finned tubes of heat exchanger.
IV. The nanofluids as base fluid indicated higher pressure drop especially at the grater Reynolds number comparison with the base fluid (pure water) qua the most relevant impact was found for 5% SWCNT nanoparticle equal to 85% compared to base fluid.
V. Initially by increasing the Reynolds number the effectiveness of heat exchanger decreased but then with more augmentation of Reynolds number from 2300 to 3300 and enters to the turbulence range and increment the turbulence eddy viscosity and kinetic energy the thermal efficiency of heat exchanger was increased.
VI. As regarding to numerical investigations, the influence of finned tubes comparison with other factors such as Reynolds number and volume concentration of nanoparticles was more.

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