Journal of Al Azhar University Engineering Sector
Vol. 9, No. 30, January, 2014, 213-225

HEAT TRANSFER ENHANCEMENT IN HEAT EXCHANGERS

Sherif H. Taher[1], Eed A. Abdel-Hadi[1], Ramadan Y. Sakr[1] and Mostafa S. Hemeda[2]
[1]Mech. Eng. Dept., Shobra Faculty of Eng., Benha University.
[2]Graduate Student

ABSTRACT
Heat transfer, friction and thermal performance characteristics of Al_2O_3/water nanofluid have been experimentally investigated. The nanofluid used was a stable colloidal suspension of Al_2O_3 nanoparticles of average diameter 50 nm. The convective heat transfer coefficient and friction factor characteristics of Al_2O_3 nanofluid during flow in a circular tube is investigated experimentally for Reynolds number ranged from 2,670 to 22,000 and volume concentration ranged from 0.3 % to 1%. The Nusselt number and friction factor of the base fluid are determined as reference data. The obtained results reveal that Nusselt number increases with increasing Reynolds number and nanofluid concentration. Correlations are developed based on the experimental measurements which can be used for the estimation of Nusselt number and friction factor of water and nanofluid flow in tubes. The heat transfer coefficient is enhanced by 45.18% and friction factor by 17.92% at 1% volume concentration compared to flow of water at similar operating conditions in same tube

KEYWORDS: Nanofluid, Forced Convection Heat Transfer, Circular Pipe Flow and Heat Transfer Enhancement.

NOMENCLATURE

A	Tube surface area	m^2
C	Specific heat at mean bulk temperature	J/kg K
D	Inner diameter of the test section	m
d_p	Nanoparticle diameter	
f	Friction factor = $\Delta P / ((L/D) \times \rho U^2/2))$	
g	Acceleration of gravity	m/s^2
h	Heat transfer coefficient	W/m^2 K
k	Thermal conductivity	W/m K
L	Heated length of the test pipe	m
$\dot{m}$	Mass flow rate	kg/s
Nu	Nusselt number = $(h \times D) / k$	
ΔP	Pressure difference in the U tube manometer	m water
Pe	Peclet number = $(U_m \times d_p) / \alpha$	
Pr	Prandtl number = $(c_p \times \mu) / k$	
Q	Discharge	m^3/s
q	Heat flux	W/m^2

Re	Reynolds number= $(\rho \times U_m \times D) / \mu$	
T	Temperature	K
U_b	Mean flow velocity	m/s
Δz	Head difference in the U tube manometer	m

Greek symbols

□	volume fraction of nanoparticle	
ρ	Density	kg/m^3
μ	Dynamic viscosity	kg /s m
□	thermal performance factor	
α	Thermal diffusivity	m^2/s
ν	kinematic viscosity	m^2/s

INTRODUCTION

The chemical processing industries, power stations, nuclear reactors, transportation industries, electronic industries and heating, ventilation and air conditioning (HVAC) face challenges in meeting out the cooling demand for the past decades. Therefore, the increase in efficiency of heat exchangers through the augmentation techniques has been investigated. There are three broad classifications of heat transfer augmentation techniques, passive techniques which do not require any external power such as treated surfaces, rough surfaces, extended surfaces, swirl flow devices, displaced enhancement devices, coiled tube, surface tension devices, additives for liquids, and additives for gases. Active techniques require external power to facilitate the desired flow modification for augmenting heat transfer such as mechanical aids, surface vibrations, fluid vibrations, electrostatic fields,....etc. Compound heat transfer technique is the combination of any two or three of the above mentioned techniques simultaneously. As the conventional heat transfer fluids have exhausted their cooling capacity, the new class of heat transfer fluids with 1-100 nm sized suspended nanoparticles has been introduced by Choi [1] (1995) and conceived the concept of nanofluid . It is observed that asignificant increase in thermal conductivity of liquids dispersed with nano size particles even at volumetric concentrations less than 1.0%. The use of these nanofluids for heat transfer augmentation in both laminar and turbulent flows is being investigated both theoretically and experimentally.

Several published researches have concluded that the use of nanofluid effectively improve the fluid thermal conductivity which consequently enhance heat transfer performance. Several types of nanoparticles have been employed for nanofluid preparation, including metals such as gold (Au), copper (Cu) and silver (Ag) In addition to metal oxides such as TiO_2, Al_2O_3, Al_2O_3 and CuO. Due to their significantly lower cost, metal oxides are more attractive for heat transfer enhancement applications compared to metals. The enhancement in thermal conductivity using Al_2O_3 nanofluid at different volume concentrations has been confirmed and showed increase of thermal conductivity compared to base fluid. Lee et al. [2] have observed 5% enhancement in case of use Al_2O_3 and CuO nanofluid in thermal conductivity and estimated using transient hot wire method. Choi et al. [1] observed 160% enhancement in thermal conductivity for 1.0% volume concentration of carbon nanotubes (CNTs) in engine oil. Das et al. [3] observed enhancement in thermal conductivity of nanofluid in the temperature range of 21 to 51 °C using Al_2O_3 nanofluid. Xuan and Roetzel [4] considered single phase and dispersion model for the estimation of heat transfer enhancement of nanofluids.

Xuan and Li [5] conducted experiments with CuO nanofluid under turbulent flow in a tube and reported that higher heat transfer coefficient was obtained compared to base fluid and developed a regression equation valid for the experimental range used. Numerical analysis of laminar flow heat transfer of Al_2O_3/ethylene glycol and Al_2O_3/water nanofluids in tube has been reported by Palm et al. [6] and Roy et al. [7] and observed that wall shear stress increases with the increase of volume concentration and Reynolds number. Pak and Cho [8] obtained heat transfer enhancement with $Al_2O_3$and TiO_2 nanoparticles in turbulent flow and presented regression equation which is independent of volume concentration. Maïga et al. [9]

investigated laminar and turbulent nanofluid flow inside circular tubes. Al_2O_3/water and Al_2O_3/ethylene glycol nanofluids were considered under constant wall heat flux boundary conditions. It was concluded that Al_2O_3/ethylene glycol nanofluid provides higher enhancement compared to Al_2O_3/water nanofluid.

Zeinali et al. [10] compared convective heat transfer coefficient for Al_2O_3/water and CuO/ water through square cross-section duct having 100 cm length in laminar flow under constant heat flux. The results show that a considerable enhancement of convective heat transfer coefficient for both nanofluids has been found, however, CuO/water nanofluid expresses more enhancements in convective heat transfer coefficient compared to Al_2O_3/water nanofluid at the same concentrations.

Nguyen [11] carried out an experimental investigation to study mixed convection of Al_2O_3/water nanofluid inside an inclined copper tube subjected to a uniform wall heat flux at its outer surface. The effects of nanoparticles concentration and power supply on the development of the thermal field were studied and discussed under laminar flow conditions. Results showed that the experimental heat transfer coefficient decreases slightly with the increase of particle volume concentration from 0 to 4%. Two correlations are proposed to calculate the Nusselt number in the fully developed region for horizontal and vertical tubes. Yimin Xuan and Qiang Li [12] investigated the convective heat transfer and flow characteristics of Cu/water nanofluid in a straight tube with constant heat flux under laminar and turbulent flow conditions. Sajadi and Kazemi [13] investigated experimentally turbulent heat transfer behavior of titanium dioxide/water nanofluid in a circular tube where the volume fraction of nanoparticles in the base fluid was less than 0.25%. Results indicated that addition of small amounts of nanoparticles to the base fluid augmented heat transfer remarkably. It was observed that there was no much effect on heat transfer enhancement with increasing the volume fraction of the nanoparticles. The measurements also showed that the pressure drop in case of using nanofluid was slightly higher than that in case of using base fluid and increased with increasing the volume concentration.

Chandrasekar et al. [14] examined the heat transfer and friction factor characteristics of Al_2O_3/water nanofluid in a circular tube under laminar flow with wire coil inserts. It was found that Nusselt number increased by 12.24% at Re=2275 using nanofluid with volume concentration of 0.1% compared to that of the base fluid (distilled water). By the use of two wire-coil inserts with pitch ratios of 2 and 3 together and nanofluid, Nusselt numbers were further enhanced by 21.5% and 15.9% respectively. Sundar and Sharma [15] studied the turbulent heat transfer and friction factor of Al_2O_3 nanofluid in circular tube with twisted tape inserts. The results revealed that in case of using nanofluid (0.5% volume) and twisted tape (twist ratio of 5) simultaneously the heat transfer coefficients at Reynolds numbers of 10,000 and 22,000 were higher than those of water in a plain tube by 33.51% and 42.17% respectively.

In the present study experimental study is conducted using stable suspension of Al_2O_3 nanoparticles in water as nanofluids and equations for the estimation of Nusselt number, friction factor were developed based on the experimental measurements

EXPERIMENTAL WORK

From the intensive review of the above literature, it is clear that the use of nanofluid efficiently improve the heat transfer rate with respect to the individual use of pure fluids. Also the heat transfer coefficient enhancement and friction factor increase with increasing nanoparticle volume concentration. The aim of the present experimental investigation is to estimate both the convective heat transfer and friction factor characteristics for the flow of Al_2O_3/water nanofluid for Reynolds number ranged from 2,670 to 22,000 with volume concentrations of 0.3, 0.5, 0.7 and 1 % respectively. So the present experimental work include studying the heat transfer phenomena in circular heated tube and investigate the effect of using nanoparticle with water as a working fluid instead of conventional fluids. A schematic diagram of the experimental apparatus is shown in Fig.1 and the apparatus consists of working fluid supply system, cooling coil, heating test tube and measuring devices.

1. Flow Circuit

A schematic diagram of the experimental setup is shown in Fig. 1. Fluid is supplied to the test section by a water pump. The amount of fluid flow rate was controlled by using valves and measured by using a flow meter.

2. Test Model

The heating test tube consists of the main tube (test section), two heaters and electric power supply. A horizontal circular copper tube of 24 mm inner diameter, 2mm wall thickness and length of 2000 mm only 1000 mm was considered as heated test section as depicted in Fig (1). The tube is connected to the working fluid supply system by means of PVC tubes. The test section is heated, under uniform heat flux condition, by means of an electrical heater wound around the wall of the tube. The test tube details are depicted in Fig. (2). The surface temperature has been measured by twenty four thermocouples made of copper-constantan (T-type) all these thermocouples have 0.1°C resolution and are calibrated before fixation. There are three pairs of similar thermocouples were fixed opposed to each other at equal distance on the two sides of the thermal insulation which was inserted between the main and the guard heaters to ensure steady state conditions and used to determine the heat lost . Fig. (2) Illustrates the details of the cross section of the test tube.

The thermocouples locations are shown in Fig. (3) The junctions of the thermocouples have been fixed in grooves milled in the surface parallel to the tube axis by soft hammering and then metal powder and epoxy steel were used to fix the thermocouples on the copper tube as shown in Fig (4). The junction of thermocouples is embedded in holes of 2 mm diameter and 0.5 mm from the inner surface of the tube, at distance of 0.05, 0.1, 0.15, 0.2, 0.35, 0.5, 0.65, 0.8, 0.85, 0.9 and 0.95 m respectively from entry of the tube, there are two thermocouples are fixed at inlet and outlet of test section to measure the working fluid temperatures

3. Experimental Conditions

Several experimental runs were carried out to investigate the convective heat transfer coefficient and friction factor at various volume concentrations for flow in circular tube having Al_2O_3 nanofluid. The parameters investigated include different concentrations of nanofluid which are 0.3%, 0.5%, 0.7% and 1% respectively and the Reynolds number ranged from 2,670 to 17,600. the flowing fluid was heated, under uniform heat flux condition. Along the tube which had 1000 mm length.

Initially experiments were made with water at different flow rates which are used for comparison of the results obtained.

Aluminum oxide (Al_2O_3) nanoparticles with physical properties presented in Table 1 were used in this study. Preparation of nanoparticle suspensions is the first step of the nanofluids. Al_2O_3 nanoparticles were dispersed in water without using any dispersant or stabilizer to prevent any changes of chemical properties of the nanofluid due to presence of additions. The nanoparticles with the required volume concentrations of 0.3%, 0.5%, 0.7% and 1% were dispersed in water using ultrasonic vibrator for 5 h. The prepared Al_2O_3/water nanofluid was characterized by SEM (Scanning Electron Microscope JSM 6360SEM). Figure (5) represents the SEM image of Al_2O_3 nanoparticles.

Table (1). Physical properties of Al_2O_3 nanoparticle used in this study

Nanoparticle	d_p (nm)	ρ (kg/m³)	C (J/kg K)	k (W/mK)
Al_2O_3	50	3880	773	36

4. INSTRUMENTATION

The fluid-flow rate is recorded by a calibrated flow-meter. A digital multi_ meter was used to measure the voltage drop and the electric resistance for each of the main and guard heaters. A voltage regulator (variac transformer) is used to control the heat flux applied on the tube wall via supplying both the main guard heaters by variation voltages. The pressure drop across the test section was measured by a U-tube manometer. The surface temperatures of the tube surface were measured at several axial locations using copper-constantan thermocouples. The thermocouples were fixed to the outer surface of the tube. Digital thermometer device is used to record the different values of surface temperature of the heated test tube, the bulk mean fluid temperature entering and leaving the test section.

DATA REDUCTION AND UNCERTAINTY ANALYSIS

The heat transfer and pressure drop across the tube using pure water and Al_2O_3/water nanofluid were measured according to the experimental procedure described previously. During the heat transfer runs, using pure water or Al_2O_3/water nanofluid. The convective heat transfer mechanism, measure the amount of heat received is equal to the convective heat transfer within the test section which can be written as:

$$Q = Q_c \quad (1)$$

The heat gained by the base fluid or Al_2O_3/water nanofluid in terms of enthalpy change can be expressed as:

$$Q = \dot{m}\, C\, (T_o - T_i) \quad (2)$$

Where, $\dot{m} = \rho A U_b$ is the mass flow rate that passes through the test section, C is the specific heat Al_2O_3/water nanofluid, T_o is the temperature of fluid at the outlet of test section, and Ti is the inlet temperature of the working fluid. The heat flux, q, is calculated as follows:

$$q = Q / \pi D_o L \quad (3)$$

Where, $\pi D_o L$ represents the outer surface area of the test section.

The local heat transfer coefficient (h) is evaluated from the following equation:

$$h = q / (Ts_{,x} - T_{b,x}) \quad (4)$$

Where $T_{s,x}$ is the surface wall temperature. Then, the local Nusselt number Nu_x is calculated as follows:

$$Nu_x = h_x D/k \quad (5)$$

Then the average Nusselt number is given by:

$$\overline{Nu} = \overline{h}\, D/k \quad (6)$$

In the present work, the friction factor is calculated under an isothermal flow condition as

$$f = \Delta P / ((L/D) * \rho U_b^2/2)) \quad (7)$$

The thermal performance factor (η) is defined as the ratio of the Nusselt number ratio (Nu/Nu_w) to the friction factor ratio (f/f_w) at the same pumping power which is expressed as follow:

$$\eta = (Nu/Nu_w)/(f/f_w)^{1/3} \quad (8)$$

The thermo physical properties of Al_2O_3/water nanofluid are estimated using the Eq.s. (9 to (12) at the average bulk temperature. Where 'np' indicates solid particle, without suffix indicates base fluid. Density of Al_2O_3/water nanofluid is determined by using the Eq. 9 as given by Pak and Cho (1998) [8].

$$\rho_{nf} = \square\, \rho_{np} + (1- \square)\, \rho_w \quad (9)$$

The specific heat capacity of Al_2O_3/water nanofluid is estimated using Eqn. 10 as given by Xuan and Roetzel [4].

$$(\rho C)_{nf} = (1- \square)(\rho C) + \square (\rho C)_{np} \quad (10)$$

The thermal conductivity was calculated from Maxwell model [16] which is recommended for the homogeneous liquid–solid suspensions with low volume concentration of randomly dispersed, uniformly sized and non interacting spherical particles [14]. The Maxwell equation is:

$$\frac{Knf}{Kw} = \frac{Knp+2Kw+2\phi(Knp-Kw)}{Knp+2Kw-\phi(Knp-Kw)} \quad (1)$$

The effective viscosity of nanofluid is found from the formula Eq.12 which is proposed by Einstein,(1956).This model is suitable for spherical particles and 5.0% volume concentration.

$$\mu_{nf} = (1- \square)\, \mu_w + \square\, \mu \quad (12)$$

Formulae for Comparison

1. Gnielinski's [17] correlation for single phase fluid, this relation valid for $2300 < Re < 5 * 10^6$, $0.5 < Pr < 2000$

$$Nu = \frac{(f/2)*(Re-1000)*Pr}{1+12.7*((f/2)^{0.5}*(Pr^{0.666})-1)} \quad (13)$$

Where $f = (1.58 \ln Re^{-3.82}) - 2$

2 Notter and Rouse [18] correlation for single phase fluid

$$Nu = 5 + 0.015\, Re^{0.856}\, Pr^{0.347} \quad (14)$$

3 Pak and Cho [8] correlation for Al_2O_3 and TiO_2 nanofluid, Valid in the range $10^4 < Re < 10^5$, $6.54 < Pr < 12.33$ and 0 < □< 3.0%.

$$Nu = 0.021\, Re^{0.8}\, Pr^{0.5} \quad (15)$$

4. Xuan and Li [5] correlation for Cu nanofluid Valid for Reynolds number ranged from t10, 000 to 22,500 and volume concentration ranged from 0 to 1.5%.

$$Nu = 0.0059\,(1 + 7.6286\square^{0.6886}\, Pe^{0.001})\, Re\, 0.9238\, Pr^{0.4}, \quad Pe = U_b \cdot d_p / \alpha \quad (16)$$

5 Blasius [19] using one-seventh power velocity distribution, developed, an expression for the determination of friction factor valid in the range $3000<Re<10^5$ for pure fluids as

$$Nu = 4 * 0.0791 * Re^{-0.25} \quad (17)$$

RESULTS AND DISCUSSION

1- Nusselt number of Al_2O_3 nanofluid

Measurements are conducted using water at different flow rates to validate the Nusselt number estimated from Eq. (5) as shown in Fig. 6 in comparison with the corresponding values obtained by Gnielinski [17] and Notter and Rouse [18]. As shown in Fig.6 a good agreement between the experimental results and the published data is obtained. The results of Nusselt number estimated from Eq. (6) are shown in Fig.7 and compared with those obtained by Dittus-Boelter correlation [20]. The experimental Nusselt number at various volume concentrations of Al_2O_3 nanofluid are estimated with Eq. (5) and shown in Fig. 8 with the data of water. It can be observed that Nusselt number increases with the increase of Reynolds number and volume concentration. The data obtained using Al_2O_3 nanofluid at different volume concentrations is shown and compared with those estimated from Eq. (15) given by Pak and Cho [8] for Al_2O_3 nanofluid and Eq. (16) obtained by Xuan and Li [5] for Cu nanofluid in Figs. 9 and 10 valid for turbulent flow conditions. It can be observed that the Nusselt number of Al_2O_3 nanofluid predicts higher values compared to the values estimated with Pak and Cho [8] and Xuan and Li [5] equations. A correlation is developed with the experimental data useful for the estimation of Nusselt number of both water and Al_2O_3 nanofluid.

$$Nu = 0.1107\, Re^{0.7583}\, \square^{0.1303} \quad (18)$$

Valid for Reynolds number ranged from 2,670 to 22,000, volume concentrations ranged from 0 to 1% and Prandtl number ranged from 3.72 to 6.50. The graph between the experimental data and the correlation Eq. (18) for Nusselt number is shown as Fig. 11.

2. Friction factor of Al_2O_3 nanofluid

Experiments are conducted with water and the values of friction factor estimated with Eq (7) are shown in Fig. 12 along with the values from Eq. (17) of Blasius [19]. The values of friction factor obtained with nanofluid is in the volume concentration is tested. A close agreement between the two is obtained which is due to the low volume concentrations employed. The values of friction factor of Al_2O_3 nanofluid at different concentrations is shown in Fig. 13. Generalized regression equation is developed for the estimation of friction factor of water and different volume concentrations of nanofluid in circular tube with given by:

$$f_{nf} = 1.0226\ Re^{-0.3668}\ \square^{\ 0.02828} \qquad (19)$$

The values of friction factor estimated with Eq. (19) are in good agreement with the experimental values as shown in Fig. 14, thus validating the equation developed.

3- THERMAL PERFORMANCE FACTOR OF AL_2O_3 NANOFLUID

The variation of thermal performance factor (η) with Reynolds number is illustrated in Fig. 15. As shown, thermal performance increases as the Reynolds number increases. The results also show that the thermal performance factor increases with increasing concentration of nanofluid. The use of nanofluid provides considerably higher thermal performance than that of using pure water for all Reynolds number studied. This is a result of the higher thermal conductivity of nano particle used with water as nanofluid. The thermal performances of nanofluids of 0.5, 0.7 and 1 % are higher than those of 0.3% of Al_2O_3 nanofluid over the range studied, the maximum thermal performance factor of 5.98 is found with the use of nanofluid of 0.7% by volume at Reynolds number of 10845.

Based on the present data correlation is developed for estimation of thermal performance factor for both water and Al_2O_3 nanofluid as follows:

$$\eta = 0.01278\ Re^{\ 0.649}\ \square^{\ 0.121} \qquad (20)$$

The values of thermal performance factor estimated with Eq. (20) are in good agreement with the experimental values as shown in Fig. 16, thus validating the equation developed.

CONCLUSIONS

From the above discussion it can be concluded that:

1- Nusselt number increases considerably with increasing Reynolds number. The average heat transfer coefficient using Al_2O_3 nanofluid with volume concentration of 1% give 1.5 time increase compared to that of using pure water. The friction factor measurements showed an increase of approximately 1.2 time compared to that obtained using pure water

2- Both the average heat transfer enhancement ratio (Nu/Nu_w) and the friction factor ratio (f/f_w) increase with the increase of volume concentration of nanofluid but decreases with increasing Reynolds number. The results show that the friction factor using Al_2O_3 nanofluid with volume concentration of 0.3% is lower than that with using volume concentration of 1%.

3- The maximum thermal performance factor is found to be 5.53 with the use of Al_2O_3/water nanofluid and 0.7% volume concentration at Reynolds number of 10845.

4- The Nusselt number estimated from Eq. (13) given by Gnielinski's [17] valid for single phase fluid shows lower values by 66 % for volume concentration of 1 % for Al_2O_3 nanofluid under similar operating conditions.

5- The Nusselt number estimated by Eq. (15) of Pak and Cho [8] valid for Al2O3 and TiO_2 is 27.3% for 1 % of Al_2O_3 nanofluid under same Reynolds number, Also, the Nusselt number estimated from Eq. (16) given by Xuan and Li [5] valid for Cu nanofluid shows 44 % compared to that given using data of 1 % volume concentration of Al_2O_3 nanofluid under same Reynolds number.

6- Friction factor for 1% volume concentration of Al_2O_3 nanofluid when compared with that given by using water is 1.05 times and 1.3 times at Reynolds number of 2,770 and 16, 000 respectively.

7- A generalized equation is deduced here based on experimental measurements for the estimation of friction factor , Nusselt number and thermal performance applicable for water and Al_2O_3 nanofluid for flow in a circular tube under const heat flux

$$Nu = 0.1107\ Re^{\ 0.7583}\ \square^{\ 0.1303}$$

$$F = 1.0226\, Re^{-0.3668}\, \square^{0.02828}$$

$$\eta = 0.01278\, Re^{0.649}\, \square^{0.121}$$

These are valid in the range of $2{,}670 < Re < 22{,}000$, $0 < \square < 1\%$ and $3.72 < Pr < 6.50$.

REFERENCES

1. Choi, S.U.S., Zhang, Z.G., Yu, W, Lockwood, F.E., Grulke, E.A., "Anomalous thermal conductivity enhancement in Nanotube suspensions" Applied Physics Letters, vol. 79, No 14, pp 2252–2254, 2001.
2. Lee, S., Choi, S.U.S., Li, S., Eastman, J.A., "Measuring thermal conductivity of fluids containing oxide nanoparticles", Journal of Heat Transfer, vol. 121, 280–289 , 1999.
3. Das, S.K., Putra N., Thiesen P., Roetzel W., "Temperature dependence of thermal conductivity enhancement for nanofluid", Journal of Heat Transfer, vol. 125, 567–574, 2003.
4. Xuan, Y., Roetzel, W., "Conception for heat transfer correlation of nanofluid", International Journal of Heat and Mass Transfer, vol. 43, No 19, pp 3701–3707, 2000.
5. Xuan, Y., Li, Q., "Investigation on convective heat transfer and flow features of nanofluids", Journal of Heat Transfer, vol .125, pp151–155, 2003.
6. Palm, S.J., Roy, G., and Nguyen, C.T., "Heat transfer enhancement in radial flow cooling system-using nanofluid, in Proceeding of the ICHMT Inter. Symp. Advance Comp. Heat Transfer, Norway, CHT-04-121, 2004.
7. Roy ,G., Nguyen, C.T., Lajoie, P.R., "Numerical investigation of laminar flow and heat transfer in a radial flow cooling system with the use of nanofluids", Superlattices Microstruct, vol. 35, No 3 , pp 497–511, 2004.
8. Pak, B.C., Cho, Y.I., "Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles", Experimental Heat Transfer, vol .11, pp 151–170, 1998.
9. Maiga, S.E.B., Palm, S.J., Nguyen, C.T., Roy G., Galanis, N.,"Heat transfer enhancement by using nanofluids in forced convection flows", International Journal of Heat and Fluid Flow, vol. 26, pp 530–546, 2005.
10. Zeinali Heris, S., Taofik H. Nassan, Noie , S.H., "A comparison of experimental heat transfer characteristics for Al_2O_3/water and CuO/water nanofluids in square cross-section duct", International Communications in Heat and Mass Transfer , vol. 37 , pp 924–928, 2010.
11. Nguyen ,C.T., Ben Mansour, R., Galanis, N., " Experimental study of mixed convection with water Al_2O_3 nanofluid in inclined tube with uniform wall heat flux ", International Journal of Thermal Sciences , vol .50, pp 403–410, 2011.
12. Yimin Xuan, Li., and Qiang. 2003. "Investigation on Convective Heat Transfer and Flow Features of Nanofluids", Journal of .Heat and Mass Transfer125, pp.151-155.
13. A.R. Sajadi, M.H. Kazemi, "Investigation of turbulent convective heat transfer and pressure drop of TiO_2/water nanofluid in circular tube", International Communications in Heat and Mass Transfer xxx, xxx–xxx, 2011.
14. Chandrasekar ,M., Suresh, S., Chandra Bose ,A., " Experimental studies on heat transfer and friction factor characteristics of Al_2O_3/water nanofluid in a circular pipe under laminar flow with wire coil inserts", Experimental Thermal and Fluid Science, vol. 34, pp 122–130, 2010.
15. Syam Sundar ,L. and Sharma , K.V., "Laminar convective heat transfer and friction factor of Al_2o_3 nanofluid in circular tube fitted with twisted tape inserts", International Journal of Automotive and Mechanical Engineering (IJAME) January-June 2011
16. Maxwell, J. C., "A Treatise on Electricity and Magnetism, Clarendon Press", Oxford.
17. V. Gnielinski, "New equations for heat and mass transfer in turbulent pipe and channel flow", Int. Chem. Eng. Vol.16, pp 359–368, 1976.
18. Notter, R.H., Rouse, M.W., "A solution to the Graetz problem – III. Fully developed region heat transfer rates", Chemical Engineering Science, vol 27, pp 2073–2093, 1972.
19. Blasius, H., "Grenzschichten in Flussigkeiten mit kleiner Reibung (German)", Z. Math. Physics vol.56, pp 1–37, 1908.
20. Dittus F.W., and Boelter. L.M.K. (1930) 'Heat transfer for automobile radiators of the tubular type', University of California Publications in Engineering, Vol.2, p. 443.

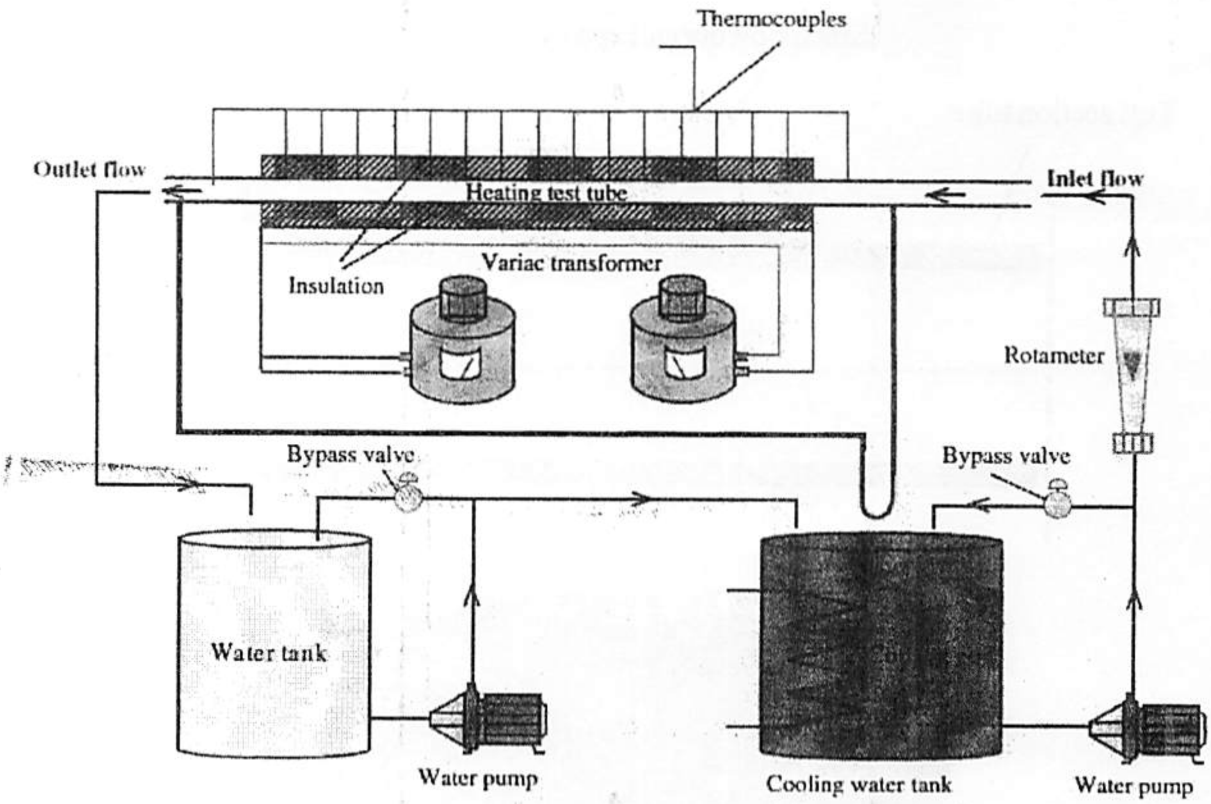

Fig. 1 Schematic diagram of experimental set-up.

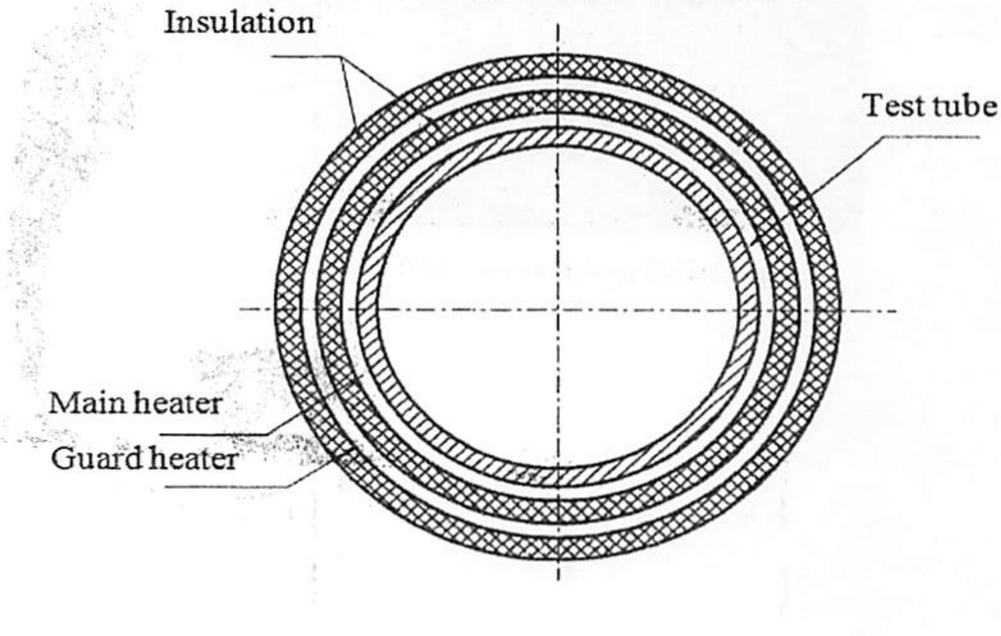

Fig. 2 Schematic of the test tube

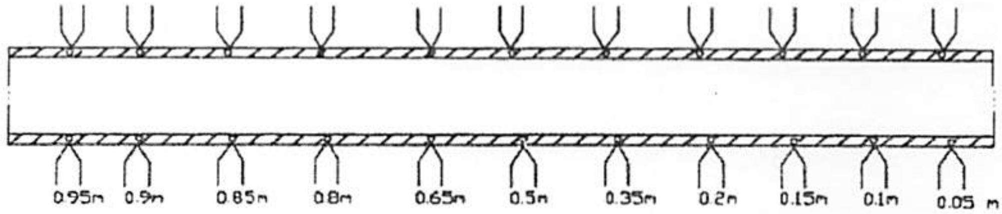

Fig. (3) Thermocouples locations on the test tube

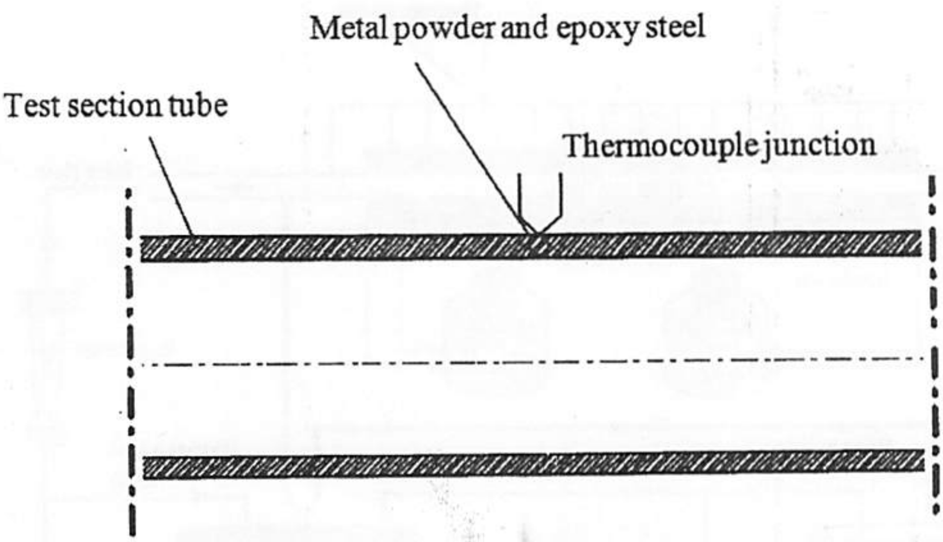

Fig. (4) Thermocouple fixation

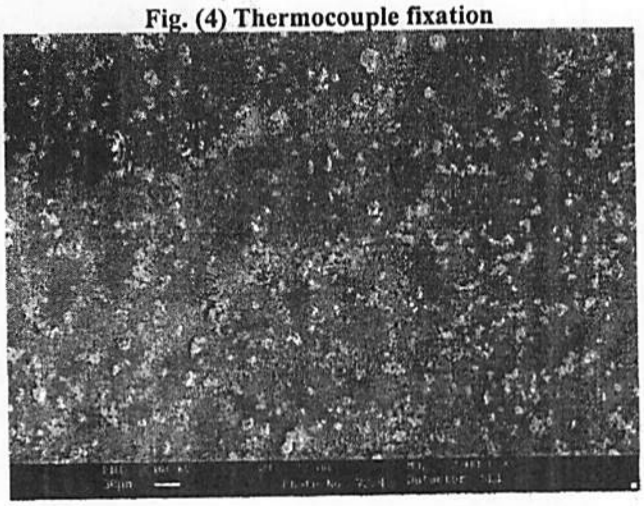

Fig. (5) SEM photograph of Al2O3 particle

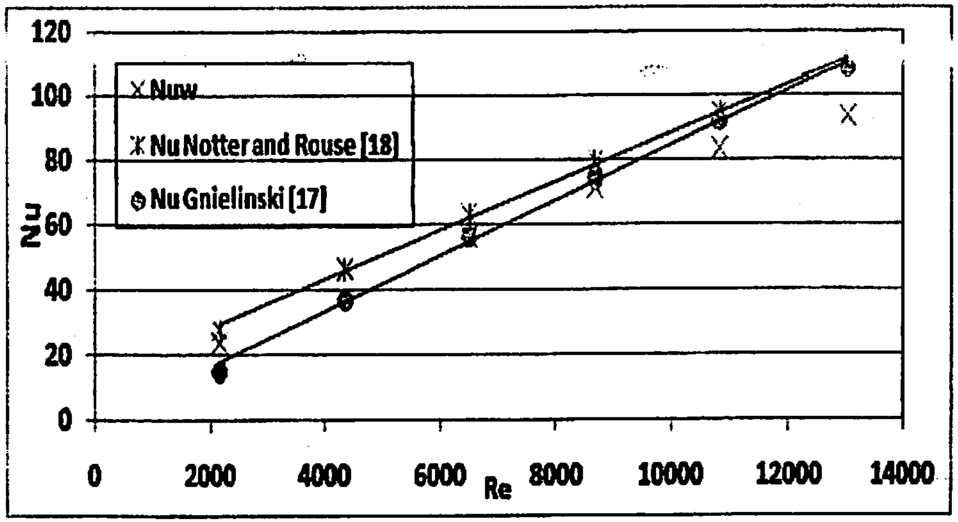

Fig.6 Comparison of measured Nusselt number, Reynolds number with published data

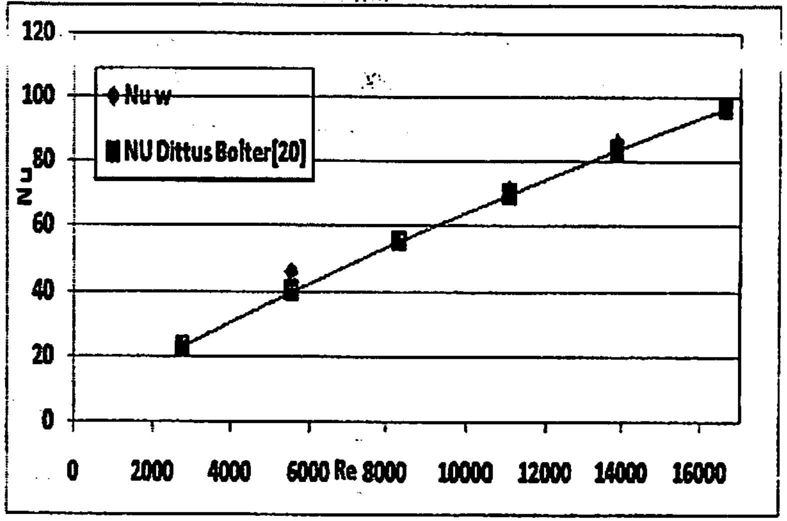

Fig. 7 Comparison between the present work and correlation of Dittus-Boelter [20] correlation

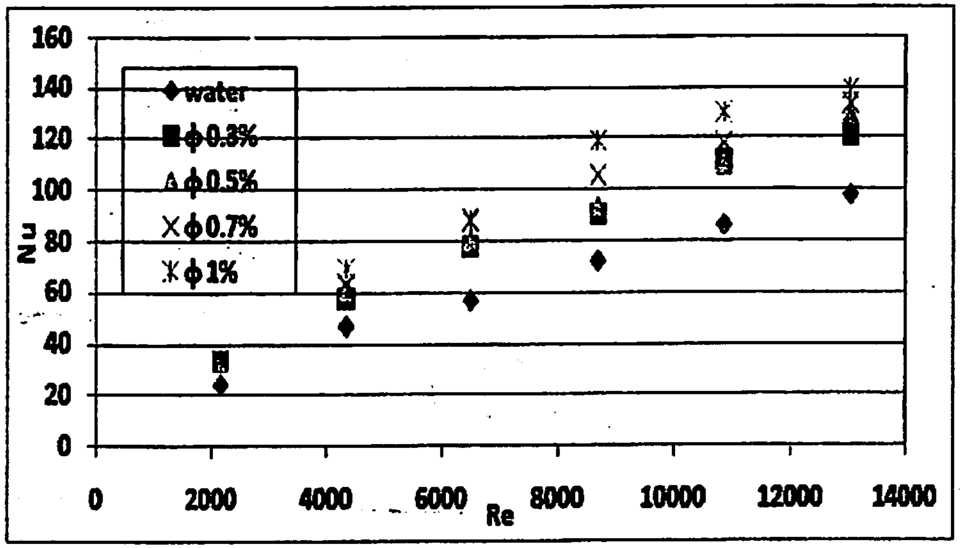

Fig. 8 Experimental Nusselt number of Al_2O_3 nanofluid with that of pure water

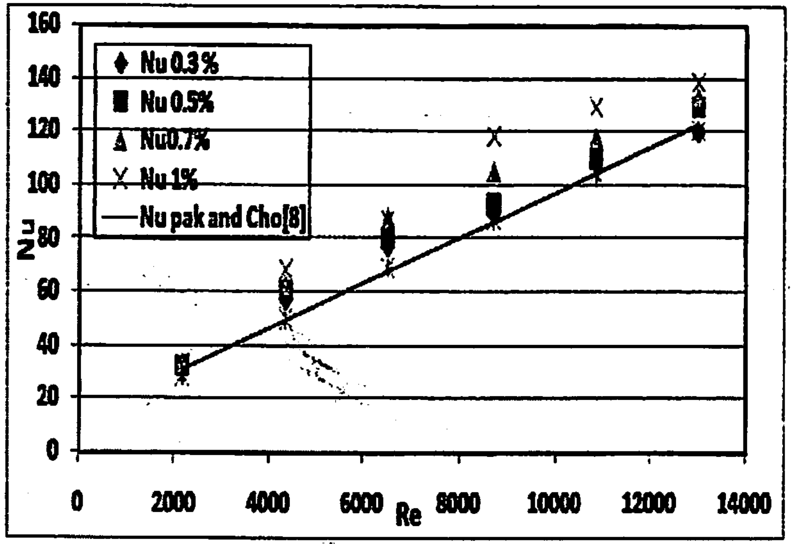

Fig. 9 Comparison of experimental Nusselt number of Al_2O_3 nanofluid with the data obtained from Pak and Cho [8] equation

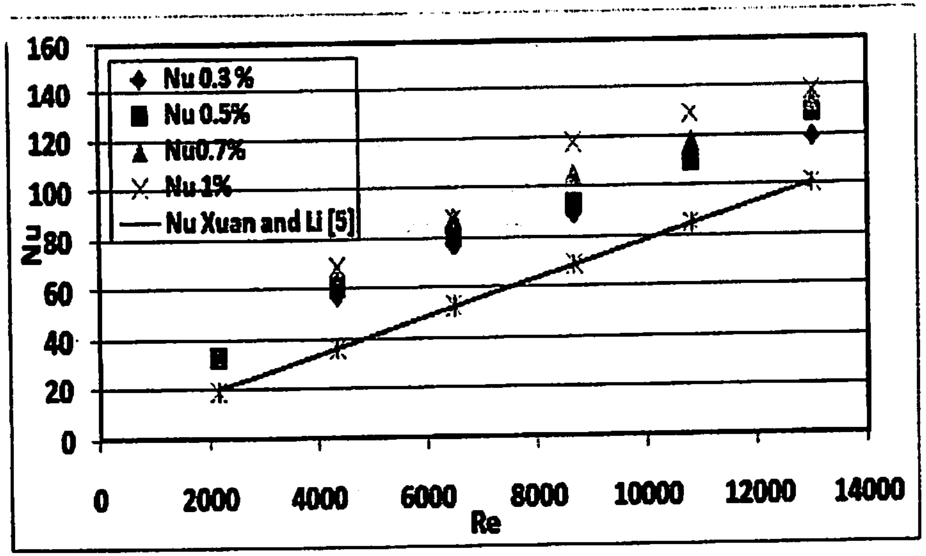

Fig. 10 Experimental Nusselt number of Al_2O_3 nanofluid is in comparison with data obtained from Xuan and Li [5] equation.

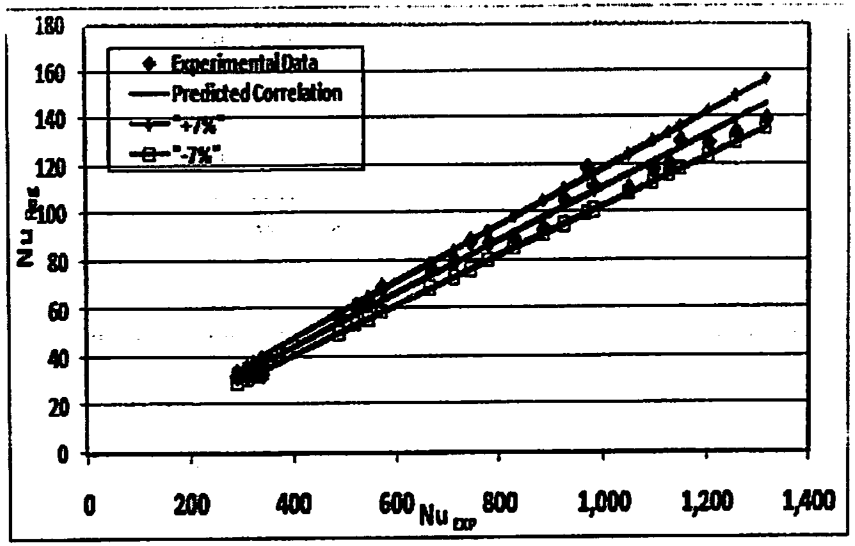

Fig. 11 Comparison of experimental Nusselt number with the developed regression Eq. (18).

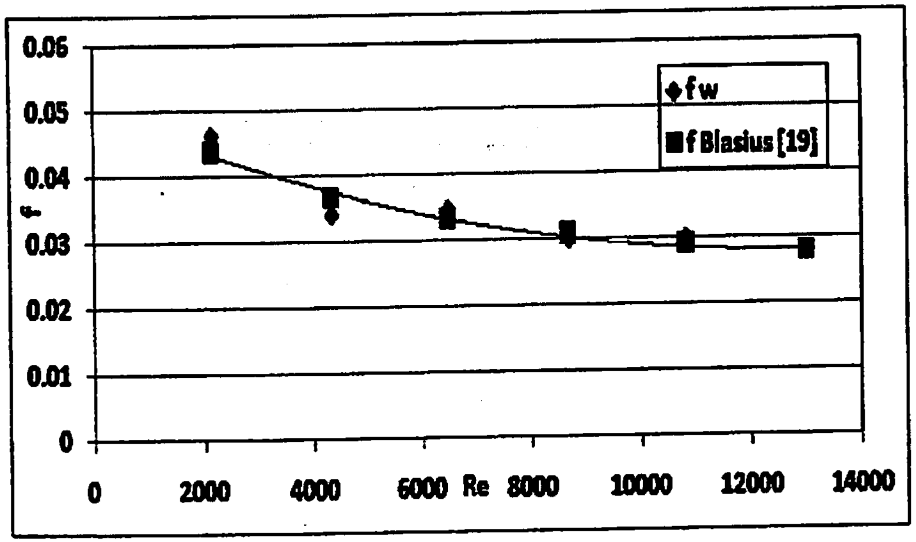

Fig. 12 Comparison between the present work and correlation of Blasius [19]

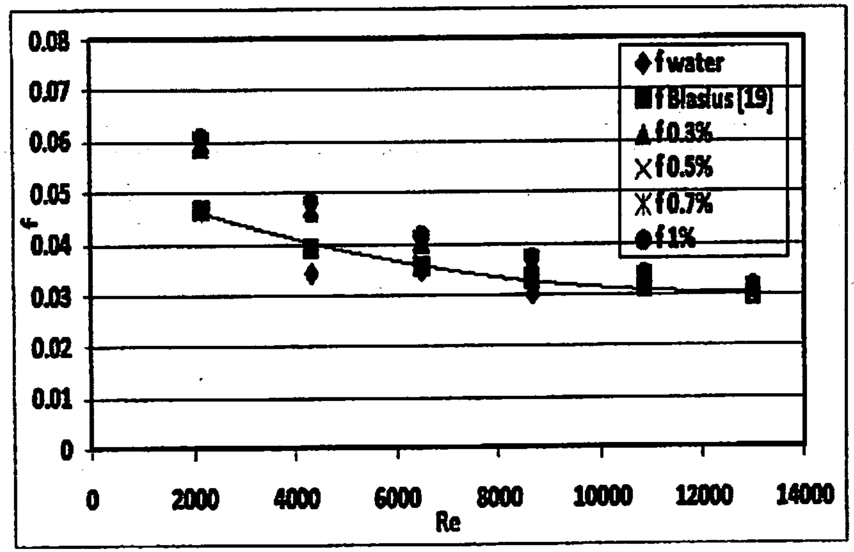

Fig. 13 Experimental friction factor of different volume concentrations of Al_2O_3 nanofluid shown in comparison with water

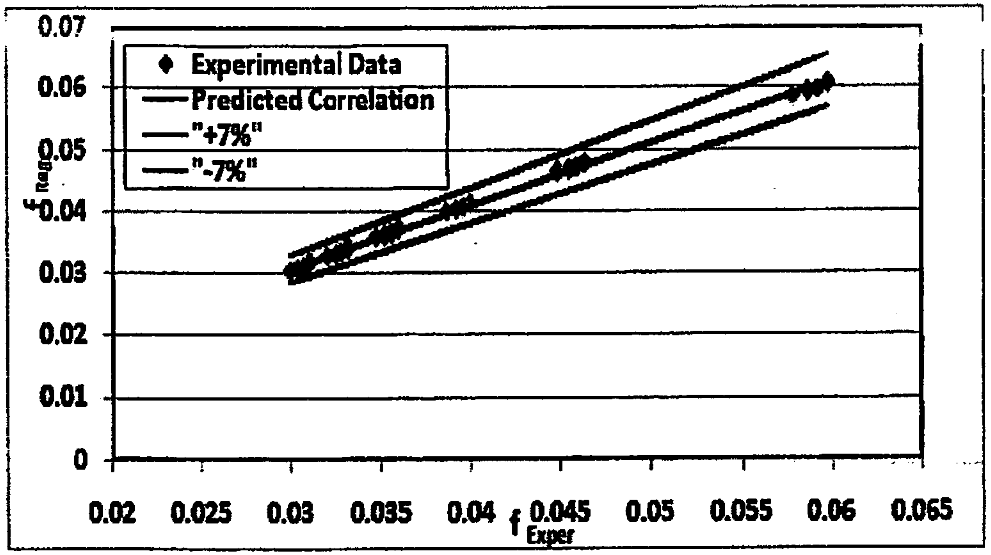

Fig. 14 Comparison of experimental friction factor with the developed regression Eq. (19).

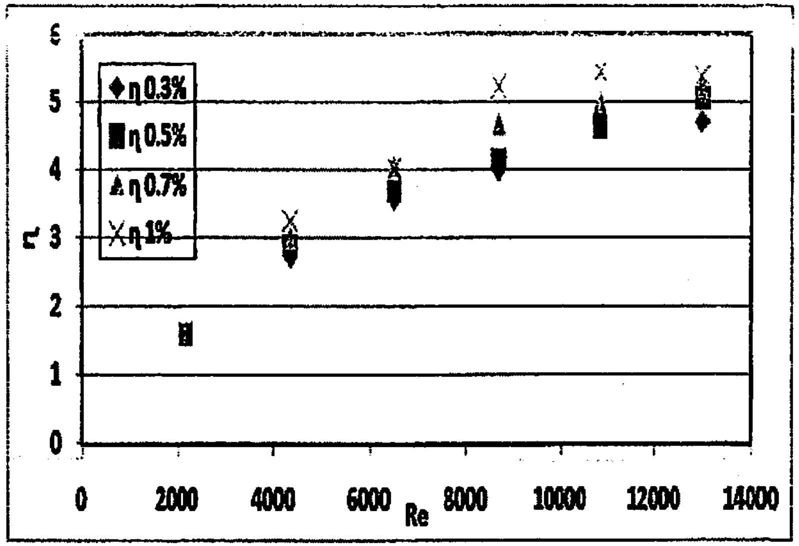

Fig. 15 variation of thermal performance factor with Reynolds number

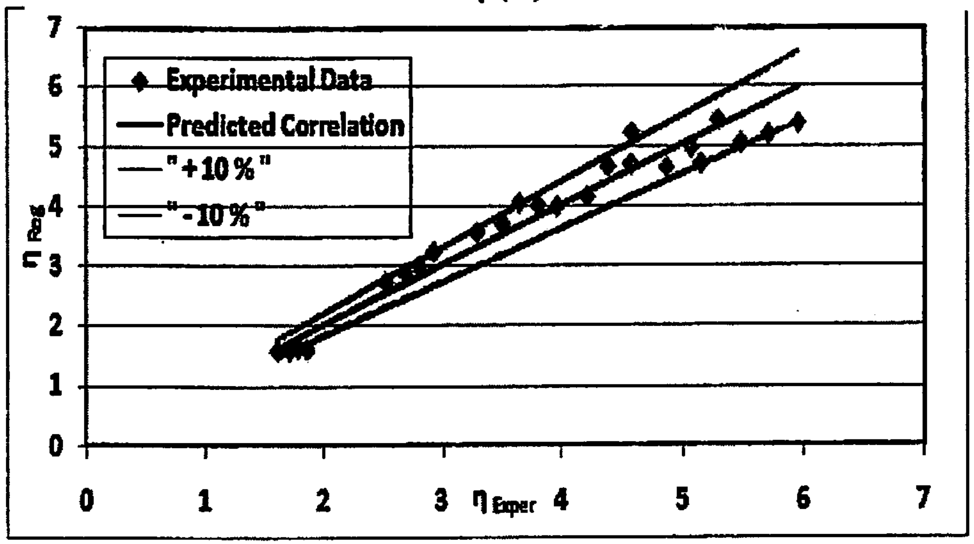

Fig. 16 comparison of experimental thermal performance with the developed regression Eq. (20).

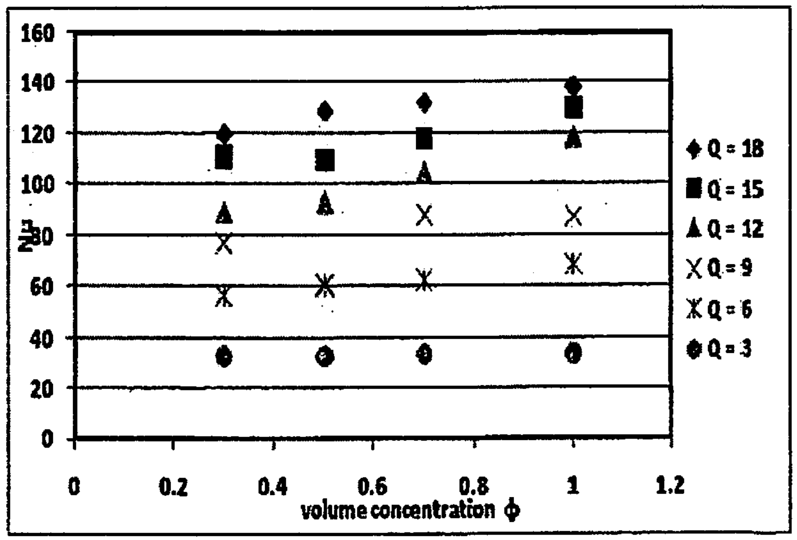

Fig. 17 variation of Nusselt Number with volume concentration at various flow rates