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Experimental Investigation of Laminar Convection Heat Transfer of Al_2O_3-Ethylene Glycol-Water Nanofluid as a Coolant in a Car Radiator

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ABSTRACT

In this experimental study, heat transfer of a coolant nanofluid, obtained by adding alumina nanoparticles to Ethylene Glycol-water (60:40 by mass), in a car radiator has been investigated. For this purpose, an experimental setup has been designed and constructed. Firstly, to investigate the accuracy of the results, the experiments have been done for the base fluid. Then the experiments have been performed for the nanofluid with different nanoparticles volume fractions of 0.003, 0.006, 0.009 and 0.012. To ensure laminar flow regime three coolant flow rates of 9, 11 and 13 lit/min have been tested. The thermophysical properties have been calculated using the recently presented temperature dependent models in the literature. According to the results, both the convective heat transfer coefficient and Nusselt number increase (about 9%) with increasing the coolant flow rate. Also, convective heat transfer coefficient increases with increasing the nanoparticles volume fraction. Although Nusselt number decreases when nanofluid is utilized, it enhances as the nanoparticles volume fraction increases. Based on the experimental results obtained, A new empirical correlation has been developed for average Nusselt number of Al_2O_3-EG-water nanofluid in developing region of flat tubes of car radiator for laminar flow and its maximum error is 3%.

Key words: Experimental study; Car radiator; Al_2O_3-Ethylene Glycol-water nanofluid; Nusselt number.

NOMENCLATURE

A	tube peripheral area
c_p	specific heat at constant pressure
d_h	hydraulic diameter of the radiator tube
d_p	nanoparticle diameter
EG	Ethylene Glycol
h	convection heat transfer coefficient
k	thermal conductivity
L	tube length
m	mass flow
Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
q	heat transfer rate
Q	volumetric flow rate
Re	Reynolds number
T	temperature
T_o	Reference temperature, 300 K
κ	Boltzmann constant, 1.3806503×10^{-23}
μ	viscosity
μ_s	viscosity at wall temperature
ρ	density
φ	volume fraction of nanoparticles

Subscripts

a	air
c	coolant
f	base fluid
fd	fully developed
i	inlet
lam	laminar
nf	nanofluid
o	outlet
p	nanoparticles

1. INTRODUCTION

Introduction of "nanofluids" over the past two decades owes to the heat transfer limitations of common working fluid. Water and water-Ethylene Glycol (EG) mixtures are fluids used in automobiles cooling systems. These fluids have low thermal conductivity compared to metals and even metal oxides. By using the nanofluid instead of the conventional fluid, heat transfer can be increased while smaller dimensions for the cooling system and less fluid flow rate are utilized. On the other hand, heat transfer enhancement from engine to the ambient increases the engine efficiency which consequently causes reduction of fuel consumption. Also, more powerful engines could be designed for different climatic conditions.

To the best of our knowledge, Singh *et al.* (2006) were among the first investigators who conducted a study on car radiator utilizing several nanofluids. They determined that use of nanofluids in car radiators can lead to a reduction up to 10% in its frontal area; thus the consequent reduction in aerodynamic drag can lead to up to 5% in fuel saving. For the same heat transfer rate, application of nanofluid contributed to reduction of friction, wear and parasitic losses and subsequently led to more than 6% fuel saving. However, they determined that use of nanofluids degrades the radiator material. In fact they observed no erosion using the coolants made of pure fluids, but there was erosion when nanofluids were used.

Saripella *et al.* (2007) modeled the cooling system of a Class 8 truck engine by Flowmaster computer code. Their numerical simulations were performed by replacing EG-water mixture (50:50) with a nanofluid comprised of CuO nanoparticles with volume fractions (φ) of 0.02 and 0.04 suspended in the same base fluid. They reported that using the nanofluid with $\varphi=0.04$ results in: (a) 5% increase in engine power, (b) 5% reduction in radiator air-side area, (c) 2.5% reduction in fuel consumption.

Putra and Maulana (2008) investigated the effect of using water-based nanofluid containing Al_2O_3 nanoparticles with $\varphi=0.01$ and 0.04 on car radiator performance. Their result showed 52-79% enhancement of convective heat transfer coefficient for $\varphi=0.04$ compared to that of the base fluid.

Vasu *et al.* (2008) studied the thermal design of a flat tube plain finned compact heat exchanger with the ε-NTU rating method using Al_2O_3-water nanofluid as the coolant. Their results showed that the pressure drop of the nanofluid with $\varphi=0.04$ was almost two times of that of the base fluid.

Bai *et al.* (2008) investigated the cooling performance of different nanofluids in engine cooling system, numerically. Their results indicated that nanofluids could enhance engine heat dissipating capacity in general and that Cu-water nanofluid had better heat transfer capability. They also found that with volume fraction increase, the engine heat dissipating capacity enhanced more such that for $\varphi=0.05$ it increased 44.1%. With remarkable enhancement of heat transfer capability, the workload of the pump of engine cooling system only increased 6%, which in their view seemed acceptable.

A three-dimensional numerical study of laminar flow and heat transfer of two nanofluids comprised of Al_2O_3 and CuO nanoparticles suspended in EG-water mixture (60:40 by mass) in an automobile radiator was performed by Vajjha *et al.* (2010a). The important point of their study was use of new temperature dependent correlations for viscosity and thermal conductivity of nanofluids. According to their results, at Reynolds number of 2000, the percentage enhancements in the average heat transfer coefficient for Al_2O_3 nanofluid with $\varphi=0.1$ and CuO nanofluid with $\varphi=0.06$ over the base fluid were 94% and 89%, respectively. Also the average heat transfer coefficient increased with the Reynolds number and the influence of the Re on heat transfer was stronger than φ. Also, they reported that the average skin friction coefficient for 0.06 CuO nanofluid in the fully developed region was about 2.75 times greater compared with that of the base fluid at a constant inlet velocity. For the same amount of heat transfer, the pumping power requirement was 82% lower for Al_2O_3 nanofluid with $\varphi=0.1$ and 77% lower for CuO nanofluid with $\varphi=0.06$ when compared to the base fluid.

The effect of using the Cu-EG nanofluid as a coolant in an automotive car radiator was investigated by Leong *et al.* (2010). They observed that the heat transfer enhancement of almost 3.8% could be achieved with adding 0.02 in volume copper nanoparticles to the base fluid; hence, they estimated 18.7% reduction of air frontal area of the radiator. However, additional 12.13% pumping power was needed for a radiator using the nanofluid with $\varphi=0.02$ compared to that of the same radiator using only pure ethylene glycol coolant at the same coolant volumetric flow rate.

Sheikhzadeh and Fakhari (2011) investigated application of Al_2O_3-EG-water nanofluid in an automobile cooling system for various conditions. They used the temperature dependent correlations to evaluate thermophysical properties for both the nanofluid and the base fluid. They compared the predictions of several Nu correlations existing in the literatures for the nanofluid under laminar flow regime and reported that the predictions of Nu correlation given by Li and Xuan (2002) showed the best agreement with their results.

Peyghambarzadeh *et al.* (2011) studied experimentally forced convective heat transfer of Al_2O_3-water nanofluid in an automobile radiator under turbulent flow regime. Their experimental results showed that increasing the fluid circulating rate can improve the heat transfer performance and the fluid inlet temperature to the radiator has trivial effect. They showed that using the nanofluid with $\varphi=0.01$ can enhance heat transfer efficiency up to 45% in comparison with pure water. Peyghambarzadeh *et al.* (2011) conducted another experimental study to investigate the forced convective heat transfer enhancement by employing the nanofluids in an automobile radiator. They added Al_2O_3 nanoparticles to water, Ethylene Glycol and EG-w mixtures with different volumetric concentrations and observed significant increase for total heat transfer rates. They also observed the Nusselt enhancement up to 40% for the nanofluid with φ=0.01 at the best condition. Furthermore, their experimental results demonstrated that the nanofluids heat transfer behavior are highly dependent on the nanoparticles volume fraction and the flow conditions, but are weakly dependent on the temperature. They also reported that predictions of Nu correlations given by Li and Xuan (2002) and Xuan and Li (2003) had good agreement with their experimental results for both the laminar and the turbulent regimes. Some mistakes in calculation of hydraulic diameter, Reynolds number and subsequently Nusselt number in Peyghambarzadeh *et al.* (2011a, 2011b) make their results doubtful.

The cooling performance of an automobile radiator under laminar flow utilizing nanofluids was numerically investigated by Huminic and Huminic (2012). Their results showed that at Reynolds number of 10, the cooling performance of the radiator utilizing Cu-EG nanofluid with φ=0.02 was enhanced about 8% compared to that of utilizing Ethylene Glycol alone.

Fakhari and Sheikhzadeh (2012) reviewed previous studies (Leong *et al.*, 2010; Peyghambarzadeh *et al.,* 2011a; Peyghambarzadeh *et al.,* 2011b; Saripella *et al.,* 2007; Sheikhzadeh & Fakhari, 2011; Singh *et al.*, 2006; Vajjha *et al*., 2010a; Vasu *et al.*, 2008) which have been carried out on the subject of nanofluids application in the car radiator. They compared the results reported by the previous researchers. Also, by curve fitting the experimental results of Peyghambarzadeh *et al.* (2011a; 2011b) they presented two new Nu correlations for both laminar and turbulent flows.

Naraki *et al.* (2013) investigated experimentally the overall heat transfer coefficient of CuO/water nanofluids under laminar flow regime ($100 \leq Re \leq 1000$) in a car radiator. Their results show that implementation of nanofluid increases the overall heat transfer coefficient up to 8% at nanofluid concentration of 0.4 vol % in comparison with the base fluid.

Chougule and Sahu (2014a) conducted an experimental study to investigate the forced convective heat transfer performance of Al_2O_3-water and CNT-water nanofluids in an automobile radiator. According to their results, nanocoolants exhibit enormous change in the heat transfer compared with the pure water. They showed that the heat transfer performance of CNT-water nanofluid is better than it of Al_2O_3-water nanocoolant. Furthermore, the Nusselt number increase with the increase in the nanoparticle concentration and nanofluid velocity.

Heat transfer and friction factor of a car radiator by using TiO_2 and SiO_2 nanoparticles dispersed in water as a base fluid was studied experimentally by Hussein *et al.* (2014). The range of Reynolds number and volume fraction are (250–1,750) and (1.0–2.5 %), respectively. According to their results, a highest Nusselt number have been recorded up to 16.4 and 17.85 for TiO_2-W and SiO_2-W respectively.

The convective heat transfer enhancement of CNT-water nanofluid studied experimentally inside an automobile radiator by Chougule and Sahu (2014b). Their CNT nanocoolants was synthesized by functionalization (FCNT) and surface treatment (SCNT) method. According to their results, both nanocoolants exhibit enormous change Nusselt number compared with water. Their results of functionalized CNT nanocoolant with 5.5 pH exhibited better performance compared to the nanocoolant with pH value of 6.5 and 9. Also, the surface treated CNT nanocoolant exhibits the deterioration in heat transfer performance. They showed that Nusselt number increase with the increase in the nanoparticle concentration and nanofluid velocity.

Sheikhzadeh *et al.* (2014) numerically investigated use of Cu-EG nanofluid with φ=0-0.05 in a typical car radiator. They reported that using the nanofluid with φ=0.05 results in: (a) 64.3% increase in overall heat transfer coefficient based on the air side, (b) 26.9% enhancement in heat transfer rate, (c) reduction of outlet coolant temperature.

Srinivas *et al.* (2015) studied anticorrosive and enhanced heat transfer properties of carboxylated water based nanofluids in the developing flow regime. In their work, DI water mixed with Sebacic acid as carboxylate additive was dispersed with very low mass concentration of multi walled carbon nano tubes. Their experimental results determined that carboxylate water dispersed with MWCNTs is resistant to corrosion and hence suitable for automotive environment. In addition to MWCNTs, carboxylated water dispersed with nano sized silver, copper and Aluminum oxide were also tested for corrosion performance but found to be giving considerable corrosion in automotive environment. According to their results, the coolant side overall heat transfer coefficient and overall heat transfer coefficient have improved markedly by using the nanofluid. Also, they developed new correlation of Stanton number.

So far, several theoretical and experimental studies on nanofluids heat transfer in car radiators have been performed. The theoretical ones (except (Vajjha *et al.,* 2010a)) used the temperature independent models for thermophysical properties of nanofluids which brought in many errors in the calculations. The experimental ones (Hussein *et al.,* 2014; Peyghambarzadeh *et al.,* 2011a; Peyghambarzadeh *et al.,* 2011b) not only had the problem of using temperature independent models, but also had several mistakes in calculation of hydraulic diameter, Reynolds number and subsequently Nusselt number. However, the question of what is the real effect of nanofluids application on heat transfer in the car radiators is still unanswered. This study attempts to investigate experimentally the effect of using a nanofluid comprised of alumina nanoparticles suspended in EG-water (60:40 by mass) on thermal performance of a car radiator.

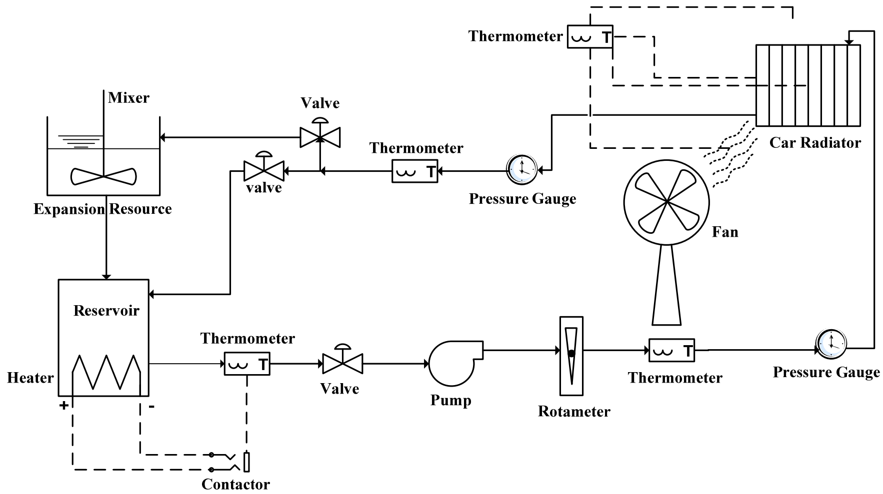


Fig. 1. A schematic diagram of the experimental set up.

2. EXPERIMENTAL SETUP

The schematic diagram of the experimental set up is shown in Fig. 1. The experimental setup includes a car radiator with a fan, reservoir, expansion resource, heater, thermometers, pump, rotameter, miniature fuses, tubes and connectors. Some characteristics of the used car radiator are mentioned in Tables 1-3 and Fig. 2.

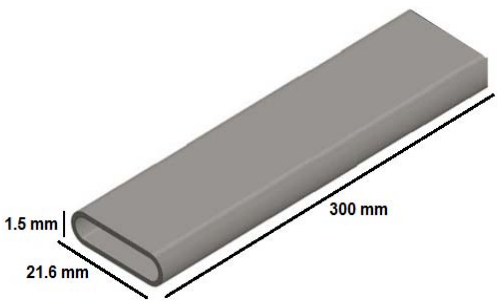


Fig. 2. Schematic and dimensions of the radiator flat tube.

Table 1 Dimensions of the typical car radiator

Height	Length	Width
330 mm	364 mm	27 mm

Table 2 Characteristics of the car radiator fins

Type	Material	Thickness	Spacing
helical	Aluminium	0.11 mm	1.84 mm

Three thermometers (Model: SAMWON ENG) have been used to measure the coolant temperature. Also an Autonics thermometer (model: T4WM) has been applied to measure air input and output temperatures as well as temperatures of front and rear surfaces of the radiator. The thermometers have the measurement accuracy of 0.1 ^{o}C and their temperature sensors are RTD (PT100 type).

A centrifugal pump of DAB ™ A50-180M model has been used. The reason for this selection is threefold. First, car water pumps are generally centrifugal type. Second, when the output of pump has been closed to regulate flow, pump is not damaged and at the last, the pump has the capacity to tolerate high temperatures up to 110 °C.

To measure the volumetric flow rate of the circulating base fluid, a rotameter has been used which has the measurement range of 2-8 lit/min and the accuracy of 1 lit/min. After being installed as part of the experimental setup, the rotameter was calibrated for various temperatures. Since the nanofluid had milky color and consequently the central trackball of rotameter was not visible, direct measurement method was used to measure the nanofluid flow rate. For this purpose and for every case, the experiment was conducted first for the approximate desired flow rate until the steady state statues reached. Then, the nanofluid flow rate was measured by measuring the passage time of a certain volume of the nanofluid in the steady state condition. For every flow rate, the measurement was repeated three or four times and the mean of measured values accepted as the volumetric flow rate.

Because of the method used to measure the nanofluid flow rate, a limitation was made by shrinking the size of reservoir, that is, reduction in the passage time of the nanofluid and consequently reduction in measurement accuracy of its flow rate. However, due to the high cost of the nanofluid preparation, a high volume reservoir could not be used. Therefore, a cylindrical reservoir with 12cm diameter and 20cm height has been constructed and embedded in the experimental setup. In order to fill and bleed the setup and also to create the necessary space for coolant expansion, an expansion resource has been perched over the reservoir and at the highest point of the setup. A 4000 W electrical element to heat the working has been placed in the reservoir.

Table 3 Characteristics of the car radiator tubes

Inner dimension	Number of tubes	Material	Spacing	Thickness	d_h
1.5×21.6×300 (mm)	34	Aluminium	8.02 mm	0.18 mm	2.84 mm

3. NANOFLUID PREPARATION

The commercial nanoparticles of γ-Al_2O_3 with diameter of approximately 20 nm (Nano Pars Lima Co., Ltd.) as dry powder were purchased. TEM image of the Al_2O_3 nanoparticles, shown in Fig. 3, indicate that the primary shape of nanoparticles is approximately spherical. The characteristics of alumina nanoparticles are shown in Table 4. In order to prepare a nanofluid by dispersing the nanoparticles in a base fluid proper mixing and stabilization of them is required. In the present work, a two-step method was used to prepare uniform and stable nanofluid. The nanofluid was prepared by dispersing the nanoparticles in the base fluid consisting of water and Ethylene Glycol (60:40 by mass). The solution was mixed thoroughly with a mechanical mixer to achieve uniform mixture. Then it was subjected to ultrasonic vibration continuously for approximately 4 h to break down agglomeration of the nanoparticles and to achieve homogeneous nanofluid.

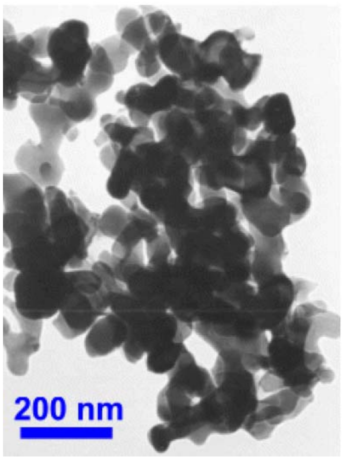


Fig. 3. TEM image of the Al_2O_3 nanoparticles.

Table 4 Some characteristics of γ-alumina nanoparticles

properties	value
Appearance	White powder
Purity	+99%
Grain size	20 nm
Specific surface area	>138 m^2/g
Ca	≤ 25 ppm
V	≤ 7 ppm
Cl	≤ 315 ppm
Na	≤ 70 ppm
Mn	≤ 3 ppm
Co	≤ 2 ppm

It is worth mentioning that the nanoparticles deposition is not observed until the nanofluid becomes stagnant or quasi-stagnant (with low speed). As the nanofluid passes through the pump, it undergoes high shear and stress tensions; thus clustering is removed (Nasiri *et al.*, 2011). Additionally, high flow rate in the radiator tubes and connecting pipes improves the stabilization of the nanofluid. In this study, there was no dispersant or stabilizer added to the nanofluid. This is due to the fact that the addition of any agent may change the nanofluid properties (especially at temperatures above 60 °C) (Pak & Cho, 1999; Wang and Mujumdar, 2008; Wen *et al.*, 2009; Wu *et al.*, 2009) and the wish was to simulate the actual condition encountered in a car radiator.

4. THERMOPHYSICAL PROPERTIES

In this study the nanofluid properties are calculated using the temperature-dependent correlations which have been published recently in the literature. According to wide temperature variations during each test, using temperature-dependent correlations makes the results free of errors and assumptions observed in previous similar studies.

As described by Vajjha *et al.* (2009), the best correlation for density of Al_2O_3 nanoparticles dispersion in 60:40 EG-water is the theoretical Eq. given by Pak and Cho (1999). Therefore, this density Eq. is adopted in the present work as:

$$\rho_{nf} = \varphi\rho_p + (1-\varphi)\rho_f \tag{1}$$

The specific heat of Al_2O_3-EG-w nanofluid given by Vajjha and Das (2009a), within the temperature range of 315 to 363 K, is:

$$\frac{c_{p,nf}}{c_{p,f}} = \frac{0.0008911\,T + 0.5179\dfrac{c_{p,p}}{c_{p,f}}}{0.425+\varphi} \tag{2}$$

Sahoo *et al.* (2009) measured the viscosity of Al_2O_3-EG-w nanofluid for volume fractions up to 0.1. For the temperature range of 273 to 363 K, they proposed:

$$\mu_{nf} = 2.392\times10^{-7}\exp\left(\frac{2903}{T}+12.65\varphi\right) \tag{3}$$

Vajjha and Das (2009b) measured the thermal conductivity of Al_2O_3-EG-w nanofluid for 60:40 EG-w. They developed a thermal conductivity model as a two-term function in the temperature range of 298 to 363 K as:

$$k_{nf} = \frac{k_p + 2k_f - 2(k_f - k_p)\varphi}{k_p + 2k_f + 2(k_f - k_p)\varphi}k_f + 5\times10^4 B\varphi\rho_f c_{p,f}\sqrt{\frac{\kappa T}{\rho_p d_p}}f(T,\varphi) \tag{4-a}$$

where $f(T, \varphi)$ is:

$$f(T,\varphi) = \left(2.8217\times10^{-2}\varphi + 3.917\times10^{-3}\right)\frac{T}{T_o} + \left(-3.0669\times10^{-2}\varphi - 3.91123\times10^{-3}\right) \quad (4\text{-b})$$

and $B = 8.4407(100\varphi)^{-1.07304}$

The above relations for nanofluid thermophysical properties require the properties of both the nanoparticles and the base fluid. The properties of the nanoparticles are presented in Table 5.

Table 5 Properties of nanoparticles at temperature of 300 K (Incropera and DeWitt, 1996)

Nanoparticles	ρ (kg m^{-3})	c_p (J kg^{-1} K^{-1})	k (W m^{-1} K^{-1})
Al_2O_3 (20 nm)	3970	765	765

The properties of the 60:40 EG-water as the base fluid considered as functions of temperature (Vajjha *et al.*, 2010b) are:

$$\rho_f = -2.43\times10^{-3}T^2 + 0.96216T - 1009.9261 \quad (5)$$

$$c_{p,f} = 4.2483T + 1882.4 \quad (6)$$

$$\mu_f = 5.55\times10^{-7}\exp\left(\frac{2664}{T}\right) \quad (7)$$

$$k_f = -3.196\times10^{-6}T^2 + 2.51188\times10^{-3}T - 0.105411 \quad (8)$$

The Eqs. (5-8) are valid within the temperature range of 293 K<T<363 K.

To evaluate heat transfer rate at the air side, thermophysical properties of air are needed. For this purpose, the density and specific heat of air are calculated as a function of temperature, within the temperature range of 20-60 ºC, from:

$$\rho_a = 1.294824 - 0.00788T + 0.00092T\times Ln(T) + 1.389225\times10^{-30}\exp(T) - 11473194\exp(-T) \quad R^2 = 0.999 \quad (9)$$

$$\frac{1}{c_{P,a}} = 0.994775 - 1.394156\times10^{-7}T^2\times Ln(T) + \frac{0.04615}{T^{3/2}} \quad R^2 = 0.999 \quad (10)$$

The Eqs. (9) and (10) were obtained by curve fitting using the data presented in (Incropera & DeWitt, 1996).

5. MATHEMATICAL FORMULATION

5.1 Heat Transfer

In order to calculate the heat transfer, some assumptions have been made as followings:

A) Radiator tubes wall temperature has been assumed to be constant. Two thermometers were mounted on front and back sides of the external surface of one of the tubes and the average of these measured temperatures was considered as the tubes wall temperature.

B) The contact thermal resistance of the thermometers was ignored.

C) Due to the very small thickness and high thermal conductivity of the radiator tubes walls, it seemed reasonable to equate the inside temperature of the tubes with the outside ones. Thus, the conduction thermal resistance of the radiator wall and deposition resistance were neglected.

The heat transfer rate from the coolant flow is:

$$q_c = \dot{m}_c c_{p,c}\left(T_{c,i} - T_{c,o}\right) \quad (11)$$

The heat transfer rate to the air flow is:

$$q_a = \dot{m}_a c_{p,a}\left(T_{a,i} - T_{a,o}\right) \quad (12)$$

The values of q_a and q_c should be theoretically equal, but slight differences between them were observed in the experiments. So the average value of them was used in further calculations.

$$q = \frac{q_c + q_a}{2} \quad (13)$$

Also, according to Newton's cooling law, the heat transfer rate from hot fluid to radiator tube wall is:

$$q = hA\left(T_b - T_{wall}\right) \quad (14)$$

Therefore the convective heat transfer coefficient is:

$$h = \frac{q}{A\left(T_b - T_{wall}\right)} \quad (15)$$

And the average Nusselt number is defined as:

$$Nu = \frac{hd_h}{k} = \frac{qd_h}{Ak\left(T_b - T_{wall}\right)} \quad (16)$$

where A is peripheral area of radiator tubes, T_b is bulk temperature which has been assumed to be the average values of inlet and outlet temperature of the fluid moving through the radiator and T_{wall} is tube wall temperature. It should also be mentioned that all of thermophysical properties have been calculated at fluid bulk temperature.

5.2 Uncertainty

In experimental works, data uncertainty must be calculated. Uncertainty shows to what extent a researcher is certain about the results. In general, it is assumed that uncertainty, Z, is a function of n variables from x_1 to x_n with known uncertainties. Therefore, the following Equation. can be used to calculate uncertainty (Coleman and Steele, 2009) of the results.

$$Z = Z(x_1, x_2, x_3,...) \Rightarrow (\delta Z)^2 = \left(\frac{\partial Z}{\partial x_1}\delta x_1\right)^2 + \left(\frac{\partial Z}{\partial x_2}\delta x_2\right)^2 + \left(\frac{\partial Z}{\partial x_3}\delta x_3\right)^2 + ... \tag{17}$$

By using Eq. (17), the data uncertainty has been calculated for all quantities (Fakhari, 2012). However at results section, it is displayed only for some quantities.

6. BENCHMARK TEST CASE

Since for the thermal and hydrodynamic entrance length depend on $\left(x_{fd,t}/d_h\right)_{lam} \approx 0.05 \mathrm{Re}\,\mathrm{Pr}$ and $\left(x_{fd,h}/d_h\right)_{lam} \approx 0.05 \mathrm{Re}$ (Incropera and DeWitt, 1996), respectively, the minimum values for thermal and hydrodynamic entrance lengths respectively are 0.518 and 0.034 m, respectively. In this study the length of each radiator tube is 0.3 m, therefore it is certain that all of the experiments have been performed in thermally developing region.

Before conducting systematic experiments utilizing nanofluid, the reliability and accuracy of the experimental results were checked using 60:40 EG-w mixture as the working fluid in several experiments. In the test loop shown in Fig. 1, the base fluid was circulated at various volumetric flow rates. Then using the measured data in Eq. (15), the heat transfer coefficient was calculated. Applying Eq. (8), thermal conductivity of the base fluid has been calculated and finally the corresponding Nusselt number for each volumetric flow rate has been obtained. Fig. 4 presents the results of the benchmark test cases performed to verify the procedure adopted for the convective heat transfer coefficient measurement and the results obtained. Also in Fig. 4 comparison between the experimental data and the predictions of correlations presented by Sieder and Tate (Incropera and DeWitt, 1996) and Bejan (2004) are shown. The Sieder and Tate relation for the heated fluid in a round tube with isothermal wall is given as:

$$\overline{\mathrm{Nu}} = 1.86\left(\frac{\mathrm{Re}\,\mathrm{Pr}}{L/d_h}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \tag{18}$$

$$\text{where} \begin{cases} 0.48 \le \mathrm{Pr} \le 16700 \\ 0.0044 \le \left(\dfrac{\mu}{\mu_s}\right) \le 9.75 \\ \left(\dfrac{\mathrm{Re}\,\mathrm{Pr}}{L/d_h}\right)^{1/3}\left(\dfrac{\mu}{\mu_s}\right)^{0.14} \ge 2 \end{cases}.$$

The relation given by Bejan for the thermally developing Hagen-Poiseuille flow in a round tube with uniform heat flux is:

$$\overline{\mathrm{Nu}} = 1.953\left(\frac{L/d_h}{\mathrm{Re}\,\mathrm{Pr}}\right)^{-1/3} \tag{19}$$

where $\left(\dfrac{L/d_h}{\mathrm{Re}\,\mathrm{Pr}}\right) \le 0.03$.

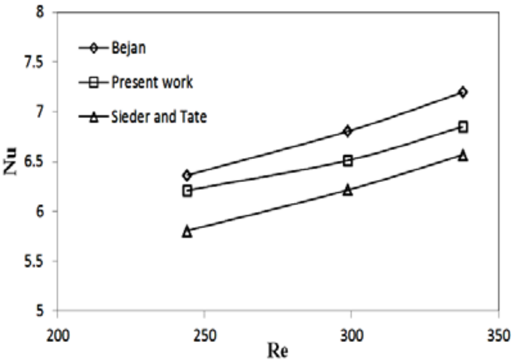


Fig. 4. Comparison between the experimental results and the theoretical values obtained from the relations of Sieder and Tate (Eq. 18) and Bejan (Eq. 19).

As seen from Fig. 4, although the trends are very similar, the Nu obtained in the present work disagrees up to 6.8% and 4.7% with those predicted by Sieder-Tate and Bejan relations, respectively. To explain the differences it should be noted that Sieder-Tate and Bejan relations have been presented for the circular tube geometry, whereas the car radiator tubes of this study are flat. As the cross section of a flat tube comprises from 4 sharp corners, for laminar flow of this study use of circular tube correlation is less accurate. On the other hand, the Sieder-Tate and Bejan correlations are based on constant axial wall temperature and constant axial wall heat flux boundary conditions, respectively. But the actual boundary condition of the car radiator of this study differs. As is expected, the results of the present study are between the results related to isothermal wall and uniform heat flux boundary conditions. Therefore, the accuracy of experimental data is confirmed.

7. RESULTS

The experimental tests were performed utilizing nanofluids with four nanoparticles volume fractions of 0.003, 0.006, 0.009 and 0.012. To investigate the effect of coolant flow rate, the experiments were conducted for three coolant flow rates of 9, 11 and 13 lit/min.

According to the present measurements and literatures (Anoop *et al.*, 2009; Asirvatham *et al.*, 2009; Wen & Ding, 2004), when nanofluid is used the thermal entrance region of tubes is longer than that of the base fluid and it increases further as nanoparticles volume fraction increases. In the following subsections the results are presented first on the effect of nanoparticles volume fraction, coolant flow rate and the Peclet number on heat transfer, then establishment of a new correlation for Nu is discussed. Also, uncertainties of Nu and h presented only at Fig. 7.

7.1 Effect of Nanoparticles Volume Fraction

In this section, the results for different nanoparticles volume fractions at constant coolant flow rate of 13 lit/min are presented and discussed. The variation of convective heat transfer coefficient with nanoparticles volume fraction is shown in Fig. 5. As the nanoparticles volume fraction increases, the convective heat transfer coefficient enhances such that for the nanofluid with φ =0.012 the relative enhancement compared to that of the base fluid is 10%.

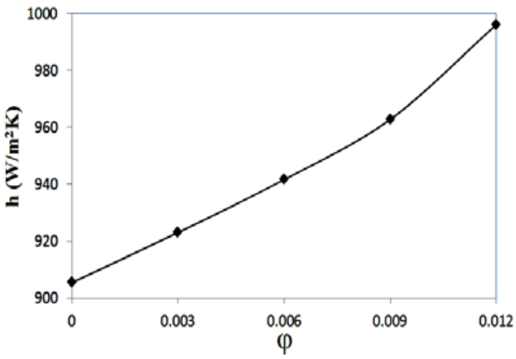


Fig. 5. Variations of convective heat transfer coefficient with φ at coolant flow rate of 13 lit/min.

Fig. 6 illustrates that when nanofluid with φ =0.003 is used, the Nusselt number decreases compared with that of base fluid, but by increasing it further the Nusselt number increases. It should be noted that although adding the nanoparticles results in an increase in the heat transfer as well as in the convective heat transfer coefficient, the thermal conductivity enhances too. In the other words, the maximum values of k and h occur for nanofluid with φ =0.012 at coolant flow rate of 13 lit/min (See Eq. 4 and Fig. 7).

Our quantitative analysis presents that the thermal conductivity of nanofluid with φ =0.012 is 16.33% higher than that of the base fluid. This increase in thermal conductivity is higher than the enhancement of convective heat transfer coefficient (=10% for nanofluid with φ =0.012 at Q=13 lit/min). As a result for the whole range of volume fractions of this study the Nusselt number of the nanofluid is less than that of the base fluid.

The same behavior was observed in the numerical work of Vajjha *et al.* (2010a), but Peyghambarzadeh *et al.* (2011b) and Hussein *et al.* (2014) reported that the nanofluid Nusselt number was always higher than that of the base fluid for laminar regime. The difference seems to be related to the nanofluid thermal conductivity model utilized by them. Peyghambarzadeh *et al.* (2011b) employed the mathematical model of Hamilton-Crosser, which underestimates the thermal conductivity of nanofluid (Vajjha *et al.*, 2009b). Also Hussein *et al.* (2014) used constant value for the nanofluids thermophysical properties which brought in many errors in the calculations.

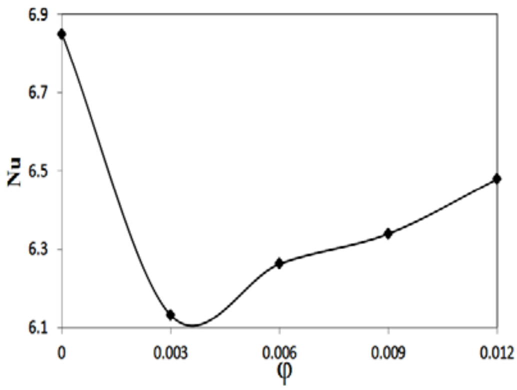
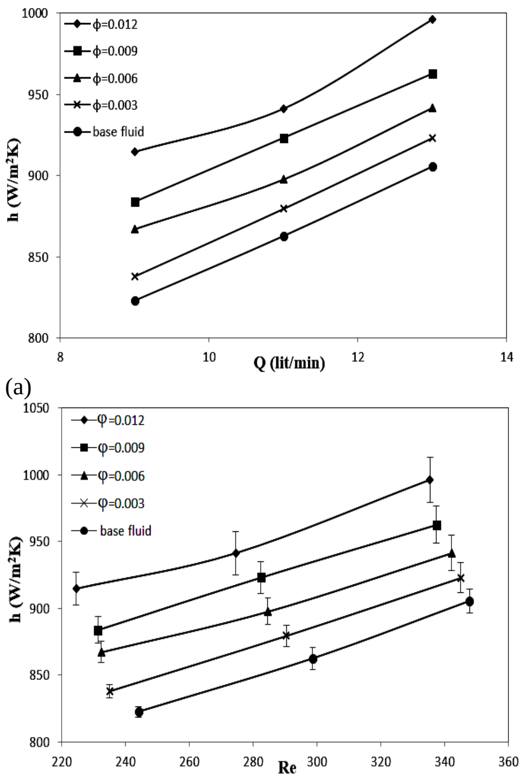


Fig. 6. Variations of Nu with φ at constant coolant flow rate of 13lit/min.



(a)

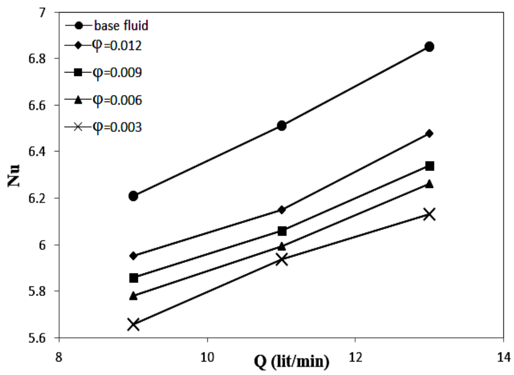
(b)

Fig. 7. Variations of h with a) coolant volumetric flow rate, b) Re number.

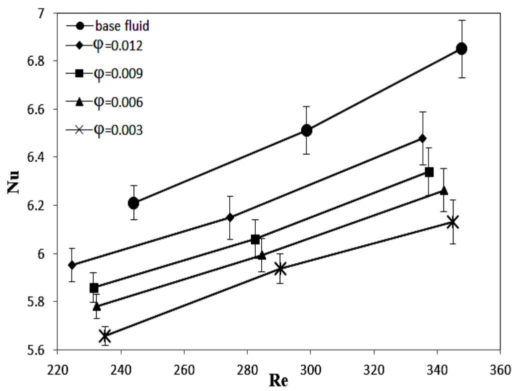
7.2 Effect of Coolant Flow Rate

Coolant flow rate plays a vital role in determining the radiator's thermal performance. If the coolant flow rate is not properly controlled, the engine becomes either overcooled or overheated. The main task is to ensure that engine is operating at optimum temperature by controlling the coolant flow rate. Although the coolant pump is usually driven by the engine, thermostat is also playing an important role to control the coolant flow rate. Thus, all of the tests in this study have been begun from temperature of 90 ºC, which is almost the opening temperature of thermostats.

As it shown in Figs. 7 and 8, as coolant flow rate (Reynolds number) increases, h and Nu enhance. For example, convective heat transfer coefficient of the nanofluid with φ =0.012 and the base fluid both for Q=13 lit/min are 8.9% and 10% higher than those for Q=9 lit/min, respectively. Also, Nu number of the nanofluid (φ =0.012) and base fluid at Q=13 lit/min increase by 8.9% and 10.3% compared to those of them at Q=9 lit/min, respectively.



(a)



(b)

Fig. 8. Variations of Nu with a) coolant volumetric flow rate, b) Re number.

7.3 Effect of Peclet Number Variations

Nusselt number has a direct relation with Reynolds and Prandtl numbers. By increasing the nanoparticles volume fraction the Prandtl number increases, whereas the Reynolds number decreases. Therefore, to investigate the overall effect of both of these dimensionless number, varitions of Nu and h with respect to the Peclet number, which is the product of Reynolds and Prandtl numbers, have been investigated. As it can be observed from Fig. 9, incresing the Peclet number causes both Nusselt number and convective heat transfer coefficient to increase.

7.4 New Nusselt Number Correlation

So far, several correlations have been developed to predict the Nusselt number of nanofluids in thermally developing region under laminar regime (Anoop *et al.*, 2009; Asirvatham *et al.*, 2009; Esmaeilzadeh, Almohammadi, Nasiri Vatan, and Omrani, 2013; Moraveji, Darabi, Haddad, and Davarnejad, 2011; Ravikanth S. Vajjha, Das, and Namburu, 2010). According to them for specified geometry, Nu is function of Reynolds number, Prandtl number and nanoparticles volume fraction.

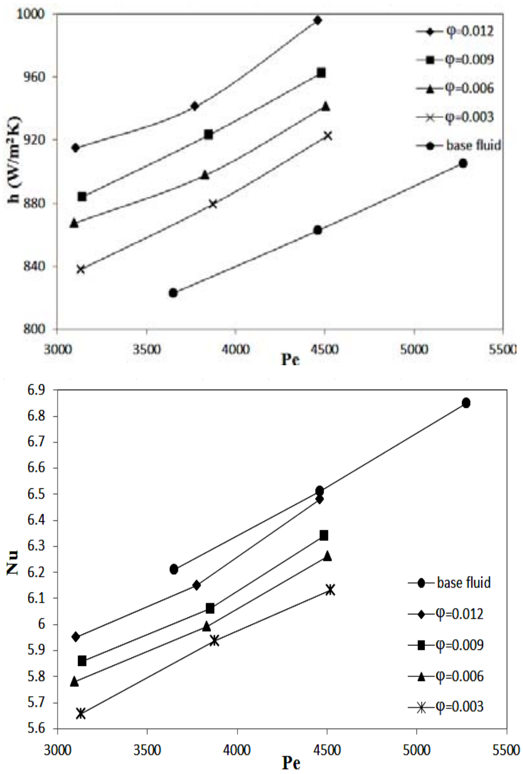


Fig. 9. Variations of h and Nu with the Peclet number.

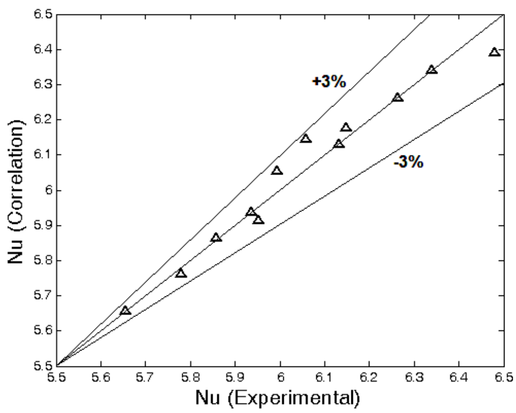


Fig. 10. Comparison of the correlation and experimental results.

After conducting experiments with nanofluids, a correlation is tried to be proposed. The correlation which is valid for Al_2O_3-EG-w nanofluid with

$0.003 \le \varphi \le 0.012$ in the developing region of flat tubes of car radiator for laminar flow with $200 \le Re \le 400$ is derived using non-linear regression analysis and is presented below.

$$Nu = 0.8038\varphi^{0.0305} Re^{0.2252} Pr^{0.3472} \quad R^2=0.97 \quad (20)$$

Figure 10 shows the correlated Nusselt number data obtained from Eq. (20) for the nanofluid. As shown in this figure, the correlated Nu data are in good agreement with the experimental ones. The maximum error is 3%.

8. CONCLUSION

In this study, heat transfer of Al_2O_3-EG-water nanofluid under laminar regime in a car radiator has been investigated. For this purpose an experimental setup has been designed and constructed. The experiments have been performed for the base fluid and nanofluid with different volume fractions and for various coolant flow rates. The thermophysical properties have been calculated using the recently presented temperature dependent models in the literature. According to the results:

a) Convective heat transfer coefficient of the nanofluid (φ =0.012) increase 10% compared to the base fluid at constant flow rate of 13 lit/min.

b) By adding the nanoparticles, Nusselt number first decreases, and then it increases by increasing nanoparticles volume fraction.

c) By increasing the nanofluid flow rate from 9 to 13 lit/min, convective heat transfer coefficient of the nanofluid (φ =0.012) increases 8.9% compared to the base fluid.

d) By increasing the nanofluid flow rate from 9 to 13 lit/min, Nusselt number of the nanofluid (φ =0.012) increases 8.9% compared to the base fluid.

e) Incresing the Peclet number causes both Nu and h to increase.

f) A new empirical correlation has been developed for average Nusselt number of Al_2O_3-EG-water nanofluid in developing region of flat tubes of car radiator for laminar flow and its maximum error is 3%.

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