Improving the Cooling Performance of Automobile Radiator with TiO_2/Water Nanofluid

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Abstract – In this paper, forced convective heat transfer in a water based nanofluid has experimentally beencompared to that of pure water in an automobile radiator. Five different concentrations of nanofluids inthe range of 0.1-1 vol.% have been prepared by the addition of TiO_2 nanoparticles into the water. Thetest liquid flows through the radiator consisted of 34 vertical tubes with elliptical cross section and airmakes a cross flow inside the tube bank with constant speed. Liquid flow rate has been changed in therange of 90-120 l/min to have the fully turbulent regime. Results demonstrate that increasing the fluid circulating rate canimprove the heat transfer performance. Meanwhile, application of nanofluid with low concentrations can enhance heat transfer efficiency up to45% in comparison with pure water.

Keywords— Nanofluid, Heat transfer coefficient, TiO_2, Radiator, Cooling performance, Experimental

I. INTRODUCTION

A reduction in energy consumption is possible by improving the performance of heat exchange systems and introducing various heat transfer enhancement techniques. Since the middle of the 1950s, some efforts have been done on the variation in geometry of heat exchanger apparatus using different fin types or various tube inserts or rough surface and the like [1-7]. Some of the published investigations have focused on electric or magnetic field application or vibration techniques [8-11]. Even though an improvement in energy efficiency is possible from the topological and configuration points of view, much more is needed from the perspective of the heat transfer fluid. Further enhancement in heat transfer is always in demand, as the operational speed of these devices depends on the cooling rate. New technology and advanced fluids with greater potential to improve the flow and thermal characteristics are two options to enhance the heat transfer rate and the present article deals with the latter option.

Conventional fluids, such as refrigerants, water, engine oil, ethylene glycol, etc. have poor heat transfer performance and therefore high compactness and effectiveness of heat transfer systems are necessary to achieve the required heat transfer. Among the efforts for enhancement of heat transfer the application of additives to liquids is more noticeable. Recent advances in nanotechnology have allowed development of a new category of fluids termed nanofluids. Such fluids are liquid suspensions containing particles that are significantly smaller than 100 nm, and have a bulk solids thermal conductivity higher than the base liquids [12]. Nanofluids are formed by suspending metallic or non-metallic oxide nanoparticles in traditional heat transfer fluids. These so called nanofluids display good thermal properties compared with fluids conventionally used for heat transfer and fluids containing particles on the micrometer scale [13]. Nanofluids are the new window which was opened recently and it was confirmed by several authors that these working fluid can enhance heat transfer performance.

Pak and Cho [14] presented an experimental investigation of the convective turbulent heat transfer characteristics of nanofluids (Al_2O_3-water) with 1-3 vol.%. The Nusselt number for the nanofluids increases with the increase of volume concentration and Reynolds number.

Wen and Ding [12] assessed the convective heat transfer of nanofluids in the entrance region under laminar flow conditions. Aqueous based nanofluids containing Al_2O_3 nanoparticles (27-56 nm; 0.6-1.6 vol.%) with sodium dodecyl benzene sulfonate(SDBS) as the dispersant, were tested under a constant heat flux boundary condition. For nanofluids containing 1.6 vol.%, the local heat transfer coefficient in the entrance region was found to be 41%higher than that of the base fluid at the same flow rate. Heris et al. [15] examined and proved the enhancement of in-tube laminar flow heat transfer of nanofluids (water-Al_2O_3) in a constant wall temperature boundary condition. In other work, Heris et al. [16] presented an investigation of the laminar flow convective heat transfer of Al_2O_3-water under constant wall temperature with 0.22.5 vol.% of nano particle for Reynolds number varying between700 and 2050. The Nusselt number for the nanofluid was found to be greater than that of the base fluid; and the heat transfer coefficient increased with an increase in particle concentration. The ratio of the measured heat transfer coefficients increases with the Peclet number as well as nanoparticle concentrations.

Lai et al. [17] studied the flow behaviour of nanofluids (Al_2O_3-water; 20 nm) in a millimeter-sized stainless steel test tube, subjected to constant wall heat flux and a low Reynolds number ($Re< 270$). The maximum Nusselt number enhancement of the nanofluid of 8% at the concentration of 1 vol.% was recorded. Jung et al. [18] conducted convective heat transfer experiments for a nanofluid (Al_2O_3-water) in a rectangular micro channel under laminar flow conditions. The convective heat transfer coefficient increased by more than 32% for 1.8 vol.% nanoparticle in the base fluids. The Nusselt number increased with an increasing Reynolds number in the laminar flow regime ($5< Re< 300$) and a new convective heat transfer correlation for nanofluids in micro channels was also proposed.

Sharma et al. [19] implemented 12.5 vol.% Al_2O_3 in water in a horizontal tube geometry and concluded that at Pe number of 3500 and 6000 up to 41% promotion in heat transfer coefficient compared to pure water may be occurred. Ho et al. [20] conducted an experiment for cooling in horizontal tube in laminar flow of Al_2O_3-water at 1 and 2 vol.% concentration and concluded the interesting enhancement of 51% in heat transfer coefficient. Nguyenet al. [21] performed their experiments in the radiator type heat exchanger and at 6.8 vol.% Al_2O_3 in water obtained 40% increase in heat transfer coefficient.

He et al. [22] investigated the heat transfer and flow behaviour of TiO_2-distilled water nanofluids were flowing in an upward direction through a vertical pipe in both the laminar and transition flow regimes under a constant heat flux boundary condition. The results indicated that the convective heat transfer coefficient increased with an increase in nanoparticle concentration at a given Reynolds number and particle size. They also found that the pressure drop of the nanofluidswas approximately the same as that of the base fluid.

Duangthongsuk and Wongwises [23, 24] examined the convectiveheat transfer and pressure drop of TiO_2-water nanofluid flowing in ahorizontal double tube counter flow heat exchanger under turbulent flow conditions experimentally. The TiO_2 nanoparticles with diameters of 21 NM dispersed in water with volume concentrations of 0.2–2%.Theirresults showed that the heat transfer coefficient of nanofluid was higher than that of the base liquid and increased with increasing the Reynolds number and particle concentrations. Moreover, their results illustrated that the heat transfer coefficient of the nanofluids a volume concentration of 2.0% was lower than that of base fluid.

In this paper, forced convection heat transfer coefficients are reported for pure water and water/TiO_2 Nano powder mixtures under fully turbulent conditions. The test section is made up with a typical automobile radiator, and the effects of the operating conditions on its heat transfer performance are analyzed.

II. NANOFLUIDS PREPARATION

Preparation of nanoparticles suspension is the first step in applyingnanofluid for heat transfer experiments. In this work theTiO_2-water nanofluid is prepared by a two-step method. The TiO_2nanoparticles with an average size of 15 nm have been provided by NANOSHEL. As the provided nanoparticles had a hydrophobic surface, they agglomerated and precipitated when dispersed in water in the absence of a dispersant/surfactant. Moreover, addition of any agent may change the fluid properties. Thus, the decision is made to functionalize theTiO_2 nanoparticles by a chemical treatment. The TiO_2 nanoparticles were mixed with 1,1,1,3,3,3, hexamethyldisilazane (C6H19NSi2) in a mass fraction ratio of 2:1. The resulting mixture was sonicatedat 30 °C for 1 h using ultrasonic vibration at sound frequency of40 kHz. This process permitted to place the hydrophilic ammonium groups on TiO_2 nanoparticles surface. Then, the soaked nanoparticles were dried with a rotary evaporation apparatus. Eventually, specific quantities of these functionalized nanoparticles are mixed with distilled water as the base fluid to make nanofluids in particular volume fractions. The suspensions were subjected to ultrasonic vibration at 400W and 24 kHz for 3–5 h to obtain uniform suspensions and break down the large agglomerations. Fig. 1 shows the field emission scanning electron microscope (FESEM) image of the nanoparticles after dispersing in water. The prepared nanofluids in the present study remained stable for several days without any observable sedimentation.

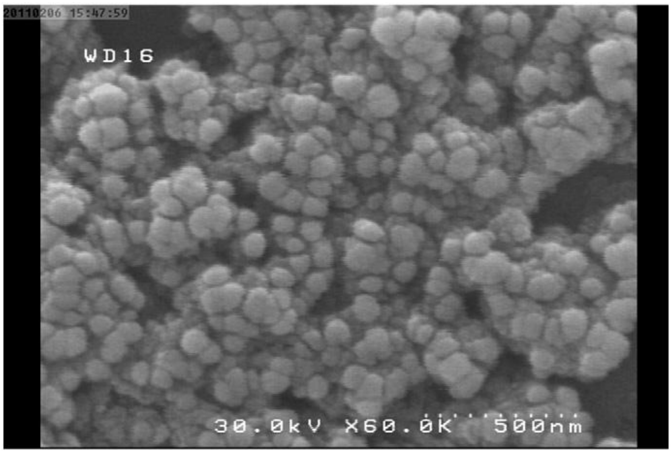

Fig. 1 FESEM image of nanoparticles after dispersion.

III. NANOFLUIDS PROPERTIES

Before the study of the convective heat transfer performance ofthe nanofluids the properties of nanofluid must be known accurately. By assuming that the nanoparticles are well dispersed in the fluid, the concentration of nanoparticles may be considered uniform throughout the tube. Although this assumption may be not true in real systems because of some physical phenomena such as particle migration, it can be a useful tool to evaluate the physical properties of nanofluid.

The density of nanofluid is calculated by the mixing theory as:

$$\rho = \emptyset\, \rho_P + (1-\emptyset)\rho_{bf} \quad (1)$$

The specific heat capacity of nanofluid can be calculated based ofthe thermal equilibrium model as follows

$$C = \frac{\emptyset \rho_P c_P + (1-\emptyset)\rho_{bf} c_{bf}}{\rho} \quad (2)$$

The effective dynamic viscosity of nanofluids can be calculated using Einstein's equation [25] for a viscous fluid containing a dilute suspension (φ⩽2%) of small, rigid, spherical particles [26]. As very dilute suspensions were used in this work the Einstein equation was used to estimate the viscosity of nanofluids. Wen and Ding [12]also used the same equation for calculating the viscosity.

$$\mu = \mu_{bf}\,(1 + 2.5\emptyset) \quad (3)$$

For calculating the effective thermal conductivity of nanofluids the Yu and Choi [27] formula has been used.

$$k = \left[\frac{k_P + 2k_{bf} + 2(k_P - k_{bf})(1+\beta)^3\emptyset}{k_P + 2k_{bf} - (k_P - k_{bf})(1+\beta)^3\emptyset}\right] k_{bf} \quad (4)$$

β is the ratio of the nanolayer thickness to the original particle radius and was set at 0.1 in this study to calculate the effective thermal conductivity of nanofluids [27].

It should be noted that these transport properties are functions of temperature. As a consequence, the properties were calculated by using the mean fluid temperature between the inlet and outlet.

For better understanding, Fig. 2 depicts variations of dimensionless physical properties of nanofluid, i.e. the ratios of physical properties of the nanofluid to those of pure water as a function of nanoparticle concentration. It is obvious that the addition of small amount of titanium oxide nanoparticle can change more or less all the physical properties of the base fluid.

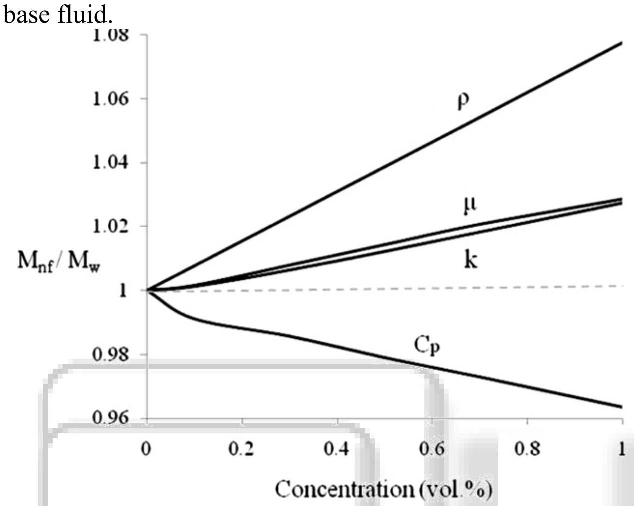

Fig. 2 Dimensionless physical properties of nanofluid in comparison with those of pure water

IV. EXPERIMENTAL RIG

As shown in Fig. 3, the experimental system used in this research includes flow lines, a storage tank, a heater, a centrifugal pump, a flow meter, a forced draft fan and a cross flow heat exchanger (an automobile radiator). The pump gives a variable flow rate of 90-120 l/min; the flow rate to the test section is regulated by appropriate adjusting of globe valve on the recycle line shown in Fig. 3. The working fluid fills 25% of the storage tank whose total volume is 30 l (height of 35 cmand diameter of 30 cm). The total volume of the circulating liquid is constant in all the experiments. Five layer insulated tubes (Isopipe0.75 in diameter) have been used as connecting lines. A flow meter (Technical Group LZM-15Z Type)was used to control andmanipulatethe flow rate with the precision of 0.1 l/min.

For heating the working fluid, an electrical heater and a controller were used to maintain the temperature between 40 and 80 ^{0}C. Two RTDs (Pt-100 U) were implemented on the flow line to record radiator fluid inlet and outlet temperatures. Two thermocouples (J-type and K-type) were used for radiator wall temperature measurement. These thermocouples were installed at the center of the radiator surfaces (both sides). The locations of the surface thermocouples have been chosen so that they give the average wall temperature. For this purpose, 10 thermocouples were attached by silicon paste to various positions of the external walls on each side of the radiator. When the experiment started, the location of the thermocouple presented the average value of the readings was selected as a point of average wall temperature. It would be interesting that these two locations on each side of the radiator did not exactly correspond.

Due to very small thickness and very large thermal conductivity of the tubes, it is reasonable to equate the inside temperature of the tube with the outside one. The temperatures from the thermocouples and RTDs were measured by two digital multimeters,a Nova-P 500-Abtin and Fluke52-USA respectively with an accuracy of 0.1 ^{0}C. All used thermocouples and RTDs were thoroughly calibrated by using a constant temperature water bath, and their accuracy has been estimated to be ±0.2 ^{0}C. Error analysis was carried out by calculating the error of measurements. The uncertainty range of Re comes from the errors in the measurement of volume flow rate and hydraulic diameter of the tubes and the uncertainty of Nu refers to the errors in the measurements of volume flow rate, hydraulic diameter, and all the temperatures. According to uncertainty analysis described by Moffat[28], the measurement error of Re was less than 5.2% and forNuwas less than18%. The repeatability of the experiments was always within 5%.

As shown in Fig. 4, the configuration of the automobile radiator used in this experiment is of the louvered fin-and-tube type, with34 vertical tubes with stadium-shaped cross section (Fig. 5). The fins and the tubes are made with aluminum. For cooling the liquid, a forced fan (Techno Pars 1400 rpm) was installed close and face to face to the radiator and consequently air and water have indirect cross flow contact and there is heat exchange between hot water flowing in the tube-side and air across the tube bundle. Constant velocity and temperature of the air are considered throughout the experiments in order to clearly investigate the internal heat transfer.

V. CALCULATION OF HEAT TRANSFER COEFFICIENT

To obtain heat transfer coefficient and corresponding Nusseltnumber, the following procedure has been performed. According to Newton's cooling law:

$$Q = hA\Delta T = hA(T_b - T_w) \quad (5)$$

Heat transfer rate can be calculated as follows:

$$Q = mC_p\Delta T = mC_p(T_{in} - T_{out}) \quad (6)$$

Regarding the equality of Q in the above equations:

$$Nu = \frac{h_{exp}d_{hy}}{k} = \frac{mC_p(T_{in}-T_{out})}{A(T_b-T_w)} \quad (7)$$

In Eq. (7), Nu is average Nusselt number for the whole radiator, m is mass flow rate which is the product of density and volume flow rate of fluid, C_p is fluid specific heat capacity, A is peripheral area of radiator tubes, T_{in} and T_{out} are inlet and outlet temperatures, T_b is bulk temperature which was assumed to be the average values of inlet and outlet temperature of the fluid moving through the radiator, and T_w is tube wall temperature which is the mean value by two surface thermocouples. In this equation, k is fluid thermal conductivity and d_{hy} is hydraulic diameter of the tube. It should also be mentioned that all the physical properties were calculated at fluid bulk temperature.

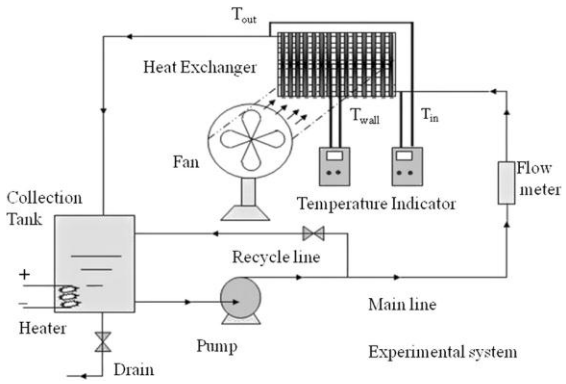

Fig. 3 Schematic of experimental setup

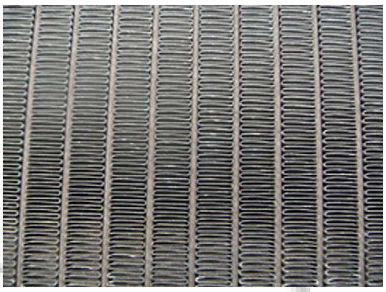

Fig. 4 The applied louvered fin and flat tube of the automobile radiator

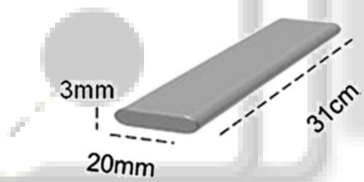

Fig. 5Schematic and dimensions of the radiator flat tube

VI. RESULTS AND DISCUSSIONS

A. *Pure Water*

Before conducting systematic experiments on the application nanofluids in the radiator, some experimental runs with purewater were done in order to check the reliability and accuracy of the experimental setup. Fig. 6shows experimental results for constant inlet temperature of 80^{0}C. As expected, the Nusselt number is seen to increase for increasing the Reynolds number.

Also, comparison was made between the experimental data and two well-known empirical correlations: one of them suggested by Dittuse Boelter [29] and the other developed by Gnielinsky [30]. These two relations were shown in Eqs. (8) and (9) respectively. InEq. (9), f is friction factor and was calculated using Eq. (10) suggested by Filonenko [31].

$$Nu = 0.0236\, Re^{0.8}\, Pr^{0.3} \qquad (8)$$

$$Nu = \frac{\left(\frac{f}{8}\right)(Re-1000)\,Pr}{1+12.7\left(\frac{f}{8}\right)^{0.5}\left(Pr^{2/3}-1\right)} \qquad (9)$$

$$f = (0.79\,\ln Re - 1.69)^{-2} \qquad (10)$$

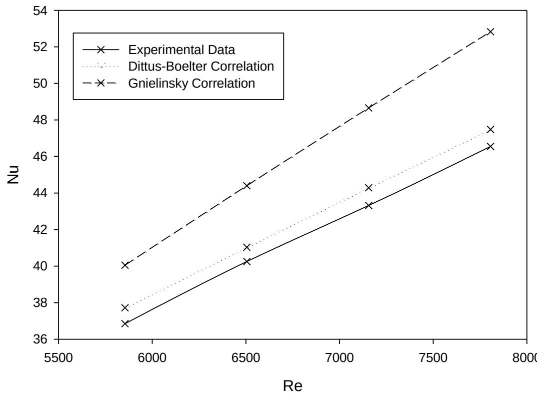

Fig. 6 Experimental results for pure water in comparison with the existing correlations.

In Fig. 6reasonably good agreement can be seen between Dittuse Boelter equation and the measurements over the Reynolds number range used in this study. The results show the correlation presented by Gnielinsky did not agree with the present experimental values for water flow in flat tubes.

B. *Nanofluid*

The nanofluid is implemented in different TiO_2 concentrations, i.e. 0, 0.1, 0.3, 0.5, 0.7, and 1 vol.% and at different flow rates of 90, 100, 110 and 120 l/min were implemented as the working fluids. It is important to mention that from a practical viewpoint for every cooling system, at equal mass flow rate the more reduction in working fluid temperature indicates a better thermal performance of the cooling system.Fig. 7shows the radiator outlet temperature, T_{out}, as a function of fluid volume flow rate circulating in the radiator. Six series of data shown in this figure belong to purewater and also five different concentrations of nanofluids. It should be noted that all the data in Fig. 7obtained when the fluid inlet temperature to the radiator was 80^{0}C. One can clearly observe that fluid outlet temperature has decreased with the augmentation of nanoparticle volume concentration.

Fig. 8shows the heat transfer enhancement obtained due to thereplacement of water with nanofluids in the automobile radiator.As can be seen in Fig. 8, Nu number in all the concentrations hasincreased by increase in the flow rate of the fluid and consequentlyRe number.

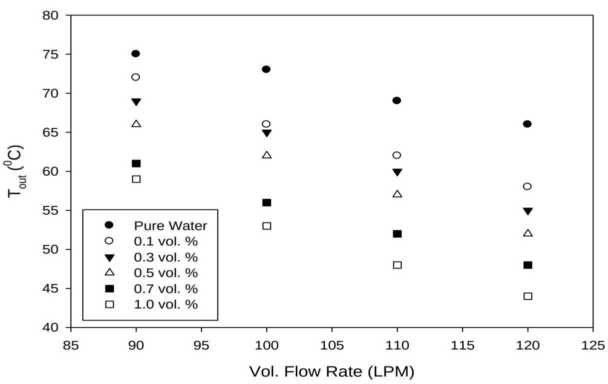

Fig. 7 Comparison of the radiator cooling performance when using nanofluid and pure water

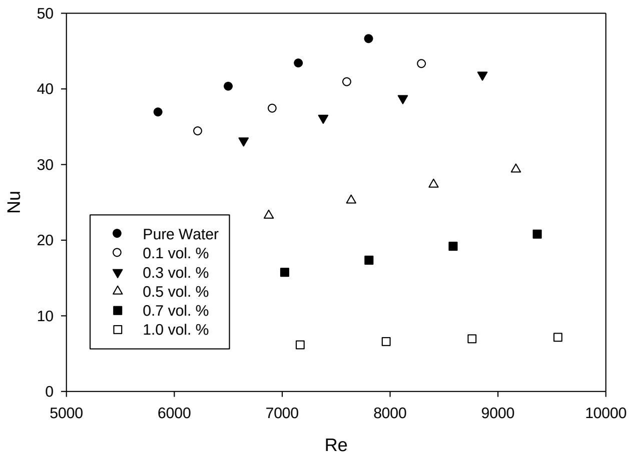

Fig. 8 Nu number variations of nanofluids at different concentrations as a function of Re number ($T_{in} = 80\ ^0C$)

Additionally, the concentration of nanoparticle plays an important role in the heat transfer efficiency. It can be shown that whenever the concentration becomes greater, heat transfer coefficient becomes larger. By the addition of only 1 vol.% of TiO_2 nanoparticle into the pure water, an increase of about 30-45% in comparison with the pure water heat transfer coefficient was recorded.

As shown in Fig. 2, the physical properties of nanofluids are slightly different than the base fluid. Density and thermal conductivity increased and specific heat decreased slightly in compare to base fluid. Viscosity increases more markedly, which is unfavorable in heat transfer.

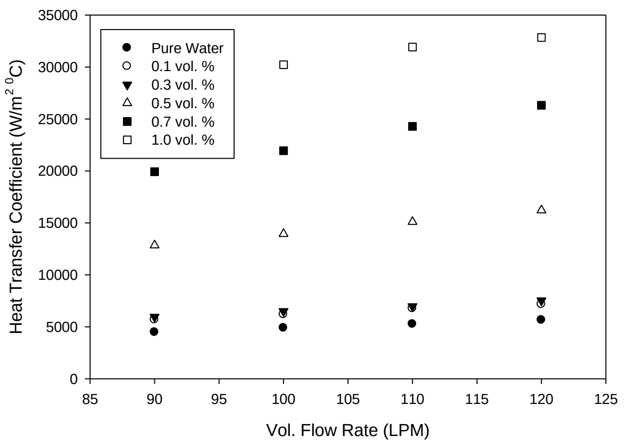

Fig. 9 Experimental heat transfer coefficient of nanofluid at different Vol. Flow Rate

The average heat transfer coefficient of nanofluids as a function of volume flow rate for different nanoparticle concentrations is presented in Fig. 9. It is observed that the heat transfer coefficient of all nanofluids is significantly higher than that of the base fluid.

These higher heat transfer coefficients obtained by using nanofluid instead of water allow the working fluid in the automobile radiator to be cooler. The addition of nanoparticles to the water has the potential to improve automotive and heavy-duty engine cooling rates or equally causes to remove the engine heat with a reduced-size coolant system. Smaller coolant systems result in smaller and lighter radiators, which in turn benefit almost every aspect of car and truck performance and lead to increased fuel economy.

VII. CONCLUSION

In this article, experimental heat transfer coefficients in the automobile radiator have been measured with two distinct working liquids: pure water and water based nanofluid (small amount of TiO_2 nanoparticle in water) at different concentration sand temperatures and the following conclusions were made.

1. The presence of TiO_2 nanoparticle in water can enhance the heat transfer rate of the automobile radiator. The degree of the heat transfer enhancement depends on the amount of nanoparticle added to pure water. Ultimately, at the concentration of 1 vol.%, the heat transfer enhancement of 45% compared to pure water was recorded.
2. Increasing the flow rate of working fluid (or equally Re)enhances the heat transfer coefficient for both pure water and nanofluid considerably.
3. It seems that the increase in the effective thermal conductivity and the variations of the other physical properties are not responsible for the large heat transfer enhancement. Brownian motion of nanoparticles maybe one of the factors in the enhancement of heat transfer. Although there are recent advances in the study of heat transfer with nanofluids, more experimental results and theoretical understanding of the mechanisms of the particle movements are needed to explain heat transfer behavior of nanofluids.

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Nomenclature

k: thermal conductivity, W/m ^0C
Nu: Nusselt number
Pr: Prandtl number
Re: Reynolds number
T: Temperature, ^0C
A: peripheral area (m^2)
C_p: specific heat (J/kg ^0C)
P: tube periphery (m)
d_{hy}: hydraulic diameter (m)¼ (4S/P)
h: heat transfer coefficient ($W/m^{20}C$)
m: mass flow rate (kg/s)
Q: heat transfer rate (W)

Greek letters
ρ: density (kg/m^3)
μ: viscosity (kg/m s)
Φ: volume fraction

Subscripts
b: bulk
exp.: experimental
in: input
nf: nanofluid
out: output
p: particle
w: wall

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