

# DESIGNING A MORE EFFECTIVE FINNED AUTOMOTIVE RADIATOR COOLING PROCESS USING CARBON NANOTUBE-WATER NANOFLUID TURBULENT FLOW

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## Abstract

The Heat removal from coolant flowing through the automobile radiator is important for optimization of fuel consumption. Increase in engine cooling, reduction of area in the automotive frontal zone and reduction of the pumping power need are computed. Results for water have been benchmarked using recent experimental data. For Carbon Nanotube-water nanofluid, an increase in Nusselt number is shown with increasing nanoparticle volumetric concentration (0-1%), inlet radiator temperature and coolant flow rate within Re 5,000 to 25,000. The highest cooling performance is 116% with performance indices always of one or higher. The engine heat drops by 61.2% when a 75% reduction in radiator length is arrived at by ANSYS FLUENT 2D analysis under turbulent flow conditions. By changing the number of fins per row, fin spacing and fin height, the heat transfer performance of the proposed compact designs was improved to the level of current design. An increase in particle volumetric concentration of the nanofluid and flow rate results in radiator pressure loss increase. The required pumping power weakens by a decrease in volumetric flow needed to maintain the heat transfer. This allows a reduction in size and weight of the automotive radiator without altering the heat transfer performance. The contour plots for the velocity, temperature and pressure distributions in the flow domain are represented graphically. The additional heat removing rate obtained for the best cooling performance is presented. The results obtained in this study suggest optimum data specifications for the tubes and fins of the radiator design.

**Keywords:** Radiator cooling, carbon nanotube, fin, nanofluid

## Introduction

Over one-third of the energy brought about by the engine through combustion is lost as heat. If the heat due to combustion is not disposed of, it will result in an excessively hot engine which in turn causes a the engine parts to be overheated, stress develops in between and the lubricating oil gets viscosity breakdown, culminating in faster wear. With a demand for engines that are more powerful and to

hold down the stress on the engine, we should get economical and compact radiators with the highest thermal performance. The size of most of the radiators differ from  $0.4826 \times 0.2921 \times 0.01778$  m to  $0.6858 \times 0.4318 \times 0.02286$  m based on engine types.

The manufacturing cost of the radiator is 20% of the total cost of the engine. By using nanofluids as coolant, radiator can be positioned in a better way

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with a compact size. There are factors other than the increase in thermal conductivity, reported for the enhanced heat transfer rates of nanofluids in the literature. Kang *et al.* (2006) experimentally investigated on the thermal performance of heat pipe with silver nanofluid (Kang *et al.*, 2006).

The diffusion velocity method is a good numerical tool for fluid flow modelling and heat transfer problems. Increasing the porous parameter has an effect of decreasing the temperature and primary and secondary velocities of the two-fluid convective flow between two parallel plates (Murty and Prakash, 2016). Applying a magnetic field on an iron oxide nanofluid-charged pulsating heat pipe enhances thermal performance (Kang *et al.*, 2017). The fluid velocity, temperature and concentration increase with increasing values of Soret number over an oscillating vertical plate under a magnetic field (Hussanan *et al.*, 2015).

The cooling rate is generally enhanced with increasing the surface area to exchange more heat with the fluid. But, this will increase the size of the radiator and at the same time, poor thermal characteristics of conventional fluids fail to exhibit higher cooling capacity. Advanced technologies and fluids with characteristics to improve thermal performance are thus inevitable. At room temperature, metallic solids possess an order-of-magnitude higher thermal conductivity than fluids. This new class of heat transfer fluids (nanofluids) is engineered by dispersing nanometer-sized solid particles, rods or tubes in traditional heat transfer fluids. Thus, nanofluids have attracted great interest from the research community due to their potential benefits and applications in numerous important fields such as microelectronics, transportation, manufacturing, medical and HVAC.

The researchers reported only a few investigations so far, by using the nanofluids in the automobile radiator. Using the CuO nanoparticles suspended in a base fluid of a 50/50 mixture of ethylene glycol and water, the cooling method of a Class 8 truck engine was designed (Saripella *et al.*, 2007). The radiator flat tubes with CuO and Al<sub>2</sub>O<sub>3</sub> nanofluids at laminar regime has been numerically studied and the heat transfer enhancement was significant (Vajjha *et al.*, 2010). Using the Al<sub>2</sub>O<sub>3</sub>-water nanofluids of various volume concentrations (0.1-1%) in a car radiator, the mass flow rate (2-5 l/min) and the inlet temperature (37-49°C) under the highly turbulent flow condition, having 45% higher tube side heat transfer coefficient was reported (Peyghambarzadeh *et al.*, 2011a). Using the ethylene glycol based 2% copper nanofluid in an automobile cooling system, resulting in a 3.8% enhancement of the heat transfer in the automotive car radiator (Leong *et al.*, 2010). Cooling performance of the

radiator is the fraction of the engine energy converted as net heat output, removed by the coolant. It refers to how well a radiator cools to changes in external temperature. Adding the Al<sub>2</sub>O<sub>3</sub> nanoparticles in the fluids such as the pure ethylene glycol, the pure water and their binary mixtures, enhanced the cooling performance of the car radiator by 40% (Peyghambarzadeh *et al.*, 2011b). The experimental analysis (Naraki *et al.*, 2013) of a car radiator using the CuO-water nanofluid, with an inlet temperature ranging from 50 to 80°C, resulted in the decrease of the overall heat transfer coefficient.

Chaurasia *et al.* (2019) reported that addition of nanoparticles, higher Re and inlet temperature enhances the rate of heat transfer using Al<sub>2</sub>O<sub>3</sub>/Water nanofluid. Radiator efficiency also increase similarly upto 65°C, after which the efficiency decreases. Contreras *et al.* (2019) through experimental study at 55-85°C obtained maximum heat transfer augmentation of 4.7% and 3.3% for 0.05% Ag-water and 0.1% graphene-water occurred as compared to base fluid in automotive cooling system. Using correlations and modelling, heat transfer coefficient of 2% Cu-EG nanofluid at Re 5,000 was 3.8% higher to its base fluid in a radiator, reducing 18.7% area with 12.1% pumping power penalty (Leong *et al.*, 2010). Peyghambarzadeh *et al.* (2013) experimentally found the enhancement of overall heat transfer coefficient of 0.65 % CuO and Fe<sub>2</sub>O<sub>3</sub> nanofluids in car radiators at 0.11 lps and inlet temperature of 80°C as 7% and 9% respectively. The 0.15% SiO<sub>2</sub> nanofluid flow in a shell and tube exchanger recorded a higher heat transfer coefficient and friction factor than its base fluid as well as upon addition of further nanoparticles (Senthilkumar *et al.*, 2018).

Abdulwahab (2014) and Jamil (2016) reported that the carbon nanotube nanofluids have higher area, aspect ratio and thermal conductivity with lower thermal resistance and specific gravity than many conventional nanofluids in use and exhibits higher thermal performance. Ding *et al.* (2006) obtained thermal enhancement of 350% using 0.5% MWCNT-water nanofluid in a horizontal tube. 13.2% thermal conductivity enhancement was obtained by dispersing 0.1% MWCNT in engine oil (Etefaghi *et al.*, 2013). Heat transfer to pumping power efficiency was 14.1% by using low volume fraction of MWCNT-EG nanofluid (Teng and Yu, 2013).

The increase in the concentration of the nanoparticle and the speed of the Carbon Nanotube-water nanofluid, experimentally enhanced the convective heat transfer inside an automobile radiator (Chougule *et al.*, 2014). Compared to the CuO-water and the Al<sub>2</sub>O<sub>3</sub>-water nanofluid, the

CNT-water nanofluid has higher thermal conductivity and aspect ratio, lower specific gravity and thermal resistance and larger specific surface area leading to the enhanced heat transfer (Chougule *et al.*, 2012). 350% higher heat transfer performance is reported by using 0.5% MWCNT-water nanofluid through a horizontal tube at a Re of 800 (Ding *et al.*, 2006). The reduced or compact shape of the radiator may improve the fuel economy, cut the vehicle weight and the drag. The rise in thermal conductivity of 20% could enhance the heat transfer by 32%, using MWCNT-water nanofluid experimentally (Garg *et al.*, 2009). In the turbulent flows, the convective heat transfer enhancement of the MWCNT/water nanofluids was found to decrease with the increasing temperatures than the laminar flows (Amrollahi *et al.*, 2010). The CNT dispersions could result enhanced heat transfer in cetyl trimethyl ammonium chloride-water solution at higher temperatures (Liu and Liao, 2010).

The experimental values of thermal conductivity of CNT/water nanofluid having aspect ratio 3000 was 16.2% higher at 60°C and increases further with the addition of nanoparticles and temperature (Wang *et al.*, 2015). The maximum heat transfer enhancements upon addition of nanoparticles were 21% for MWCNT, 17% for Cu and 12% for Al<sub>2</sub>O<sub>3</sub> nanofluids at 480 kg/h (Akash *et al.*, 2018). 78.08% thermal enhancement is noticed with 0.5% Graphene-water nanofluid at Re 1,000 (Toh and Ting, 2019). Ghasemi *et al.* (2019) reported that the type of base fluid used with MWCNT nanoparticles decides the enhancement of the thermal conductivity.

Park and Pak, (2002) reported that the heat transfer rate of a straight slab was 20% lower than in a meso-channel heat exchanger of similar length and hence the latter can be used as a car radiator. The researchers experimentally evaluated, the friction factor in a circular tube (Dehghandokht *et al.*, 2011). The forced convection heat transfer of both the CuO and the Fe<sub>2</sub>O<sub>3</sub> nanofluid flow, was experimentally studied through a car radiator (Chandrasekar *et al.*, 2010). The heat transfer coefficient was found experimentally Peyghambarzadeh *et al.* (2013) at the particle volume concentration of 2.3% was 10% higher and at particle volume concentration of 19% was 50% higher than water using SiO<sub>2</sub> nanofluid with 22 nm particle diameter at inlet temperature 20-70°C and Re 200-10,000. Nanofluid is lucrative due to its exorbitant thermal conductivity and has high potential in thermal management. However, pressure drop, erosion, clogging and sedimentation are the trivial issues due to the suspension period problems of nanofluids.

Frontal area of the radiator can be reduced by up to 10% and a fuel savings of up to 5% can be

obtained with the use of nanofluids having high thermal conductivity (Ferrouillat *et al.*, 2011). At both the high and low rotating speeds, the CuO nanofluid dispersed into the engine transmission oil and produced the lowest transmission temperature than the Al<sub>2</sub>O<sub>3</sub> nanofluid (Tzeng *et al.*, 2005). The cooling is one of the highest technical challenges faced by the automotive industry as the trend toward a higher engine power and exhaust gas recirculation necessarily leads to an increased frontal area and a larger radiator, resulting in an added aerodynamic drag and increased fuel consumption (Steve, 2006). With the addition of nanoparticles, the frictional pressure drop in a smooth flat tube was found by Peng *et al.* (2009) using a nanofluid with refrigerant as the base fluid up to 20.8% higher than its base fluid. The shear stress and heat transfer coefficient using a nanofluid exhibited a notable improvement than the base fluid in a flat tube of the car radiator (Trivedi and Vasava, 2012). Pulsating heat pipe filled 60% with 100 ppm silver nano-fluid water solution has more liquid slugs that will turn and carry more sensible heat and hence heat dissipation result is analogous and the best (Lin *et al.*, 2008). When applied to the internal combustion engine cooling passages, the under-prediction of the single-phase convective heat transfer coefficient is known to occur, while using the simple heat transfer correlations (Robinson *et al.*, 2003).

Even though, the flow pattern in the automotive radiator is extremely turbulent (airflow is laminar only), studies on the turbulent flows in the automotive radiators are very limited. The frontal positioning of a larger radiator is to get advantage of maximum air flow for better cooling. Increase of to over one-third of the engine energy removal as heat from the CNT-water nanofluid coolant is important for optimization of fuel consumption, viscosity maintenance of lubricating oil, the strength of metal parts of the engine, reduction in stress between engine parts and wear. Reduction of the pumping power need, area in the automotive frontal zone by raising the surface area to the coolant and the air ratio, reducing the radiator length and weight of radiator can place it in a better position, improve in the fuel economy, cut the vehicle weight and the drag. With a demand for more powerful engines, exhaust gas recirculation and to hold down the stress on the engine, compactness and economy of radiator with higher thermal efficiency is of paramount importance, as the manufacturing cost of the radiator is 20% of the total cost of the engine. With the anticipated results of the quantifications of an increase in the engine heat rejection, reduction in the automotive frontal area and the pumping power for the radiator, smaller radiator designing is proposed with same or higher heat transfer efficiency in this

study. The present optimisation study chooses an altering of the fin design and raising the surface area to the coolant and the air ratio. The results could also be applicable to any other system like the nuclear steam supply system.

### Geometry, Governing Equations and Simulation

As mentioned earlier and to summarize, most radiators have size variations of  $19'' \times 11.5'' \times 0.7''$  to  $27'' \times 17'' \times 0.9''$  with  $24'' \times 17'' \times 0.9''$  size being the most common (Maplesoft, 2008).

The heat transfer performance of an existing internal radiator system of  $609.6 \times 431.8 \times 22.86$  mm is optimised for a better, smaller and efficient radiator design without a serious design change. The reference radiator is having a length of 609.6 mm, a height of 431.8 mm, a width of 22.86 mm, a fin width of 34.343 mm, a fin height of 1.638 mm, a tube thickness or a cross-sectional length of 1.6 mm, the distance between tubes of 16.4 mm, a fin length of 39 mm, a gap between fin layers of 1.507 mm and a fin thickness of 0.0254 mm. The aluminium tube passes up through one side of the radiator height, traverse parallel to the radiator length just above the radiator height at a distance equal to the distance between tubes. Then it goes down the radiator height, traverse parallel to the radiator length just below the radiator height, equal to the distance between tubes and repeats the cycle until it completes the number of tubes on the other side of the radiator height. To increase the anchorage to the tube, the aluminium fins are attached to the left-hand side of the tube with a cone shape at the base. The cone shape at the base of the aluminium fins has a height of 4.8 mm.

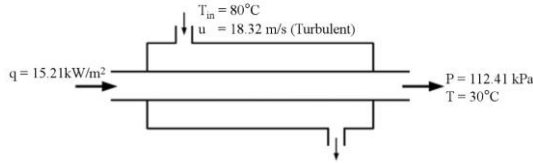
Different aspects and minute details that are expensive and difficult to find by experimental works can be enabled numerically and radiator performance and effective heat transfer prediction is possible by CFD simulation (Fan *et al.*, 2012). FLUENT 14.5 onwards is enabled with the capability modelling of two-temperature and hence,  $k-\omega$  turbulence model is a crucial and viable solution for an accurate thermal and structural analysis prediction especially for the vehicle radiators (Hayes *et al.*, 2008). Mono and hybrid nanofluids with enhanced thermophysical characteristics have several applications in heat exchangers (Bahiraei *et al.*, 2019). The  $k-\varepsilon$  model is only valid for fully turbulent flows due to assumptions in the derivation of the equations and that the effects of the molecular viscosity are negligible. In the near-wall regions, the turbulent flow is heavily disturbed by the wall and the Reynolds number becomes small and the viscous effects dominates the flow in the radiator.  $k-\varepsilon$  models have difficulties handling large gradients in the near-wall region.

For simulating the governing equations numerically, using the  $k-\omega$  turbulence model with the help of the ANSYS FLUENT 15.0, the control volume approach was used as a fully developed, steady state, 2-D, axisymmetric flow. The  $k-\omega$  model solves for two variables:  $k$ , the turbulence kinetic energy; and  $\omega$  - the specific rate of dissipation of kinetic energy (Wilcox and David, 2006). The  $k-\omega$  model is useful in many cases where the  $k-\varepsilon$  model is not accurate, such as internal flows, flows that exhibit strong curvature, etc. The  $k-\omega$  turbulence model was therefore adopted since it gives accurate results for wall-bounded flows. The algorithm used for the pressure-velocity coupling is SIMPLE and the Boussinesq approximation is used for turbulent viscosity. The computational mesh with a constant heat flux at the wall, utilizing the geometry of the problem as a flat tube was developed.

Good agreement was observed between experimental data and 2D simulation results for the resulting strains and hence provides an economic, robust and reasonably accurate way for industrial applications (Mao *et al.*, 2010). Fin pitch to tube cross-section with few millimetres to radiator size with meters, poses much difficulty for the turbulent flow and heat transfer 3D simulation. Cetin *et al.* (2017) reported that the time consumption and expenses makes the turbulent flow and heat transfer simulation, non-viable between the tube and fins. Guler (2014) performed radiator cooling capacity with two-temperature model and emerged as an efficient rating and design tool, without the requirement of experimental or empirical data. The physics in one direction is not seeing any major changes as the size along any dimension is finite. The cross-section of both the channels are rectangular, and a planar mesh solution gives similar result to a 3D mesh, and hence a 2D simulation will suffice.

The rate of flow of the heat per unit area through the wall surface is not varying and the steady flow velocity does not depend on the time. The unsteady nature of the flow and the variable heat flux does not arise in the present study. The assessment of all the scalar and the velocity values with the extensive employment of the grid schemes are made at the interface centres of the control volume. Saripella *et al.* (2007) reported that the solutions were converged, as the residuals become less than  $10^{-6}$ , with respect to the pertaining equations in whole. Figure 1 explains the computational domain with the boundary conditions specified. The convergence criteria used to control the numerical solution for continuity and momentum equations were  $10^{-6}$  and energy equation was  $10^{-7}$  respectively. The numerical simulations were finished after about 1,300 steps iteration and converge. An investigation

of a spatial resolution with a polynomial degree setting of 6-7 was performed for monitoring the Velocity, pressure and temperature. With the computational domain size having polynomial degree 6, the variation in convergence and uncertainty are less than 0.5% and 0.3% respectively for spatial resolution, in comparison with its smaller polynomial degree.



**Figure 1. Nanofluid flow through a radiator**

The researchers used CNT nanoparticles of size lower than 100 nm and adopted a single-phase method. The thermo physical properties of 1% CNT-water nanofluid at 80°C used are  $5.1 \times 10^{-4}$  kg/m.s viscosity, 1.176 W/m.K thermal conductivity, 975 kg/m<sup>3</sup> density and 4.105 kJ/kg. K specific heat (Chougule *et al.*, 2012). The average diameter of the nanoparticles used is 25 nm.

For the steady state, the continuity, the momentum and the energy dimensional conservation equations are (Wilcox, 2008):

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0 \quad (1)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_j u_i) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}(t_{ji} + \rho \tau_{ji}) \quad (2)$$

$$\begin{aligned} \frac{\partial}{\partial t} \left[ \rho \left( e + \frac{1}{2} u_i u_i + k \right) \right] + \frac{\partial}{\partial x_j} \left[ \rho u_j \left( h + \frac{1}{2} u_i u_i + k \right) \right] = \frac{\partial}{\partial x_j} \left[ u_i (t_{ij} + \rho \tau_{ij}) \right] + \left( \frac{\mu}{Pr_L} + \frac{\mu_T}{Pr_T} \right) \frac{\partial h}{\partial x_j} + \left( \mu + \sigma^* \frac{\rho k}{\omega} \right) \frac{\partial k}{\partial x_j} \end{aligned} \quad (3)$$

Convergence of the entire pertinent solutions are achieved with residuals less than  $10^{-6}$ . Compared the simulation results, with the equation for the friction factor and the Nu for the turbulent flow as (Vajjha *et al.*, 2010):

$$f = \frac{0.079}{Re^{\frac{1}{4}}} \quad (4)$$

$$Nu_{av} = 1.9421 \left( Re_{D_h} Pr \frac{D_h}{L} \right)^{\left( \frac{1}{3} \right)} \text{ for } Re Pr \frac{D_h}{L} \geq 33.3 \quad (5)$$

$$Nu_{av} = 6.1 + 0.003675 \left( Re_{D_h} Pr \frac{D_h}{L} \right) \text{ for } Re Pr \frac{D_h}{L} < 33.3 \quad (6)$$

Assuming the walls of the radiator tubes perfectly smooth, using 0 to 1% nanofluid volume concentrations and varying the Re from 5,000 to 25,000, the pressure drop was found using the standard equations.

$$Re = \frac{\rho_{nf} D_h u}{\mu_{nf}} \quad (7)$$

$$\Delta P = \frac{f \rho L u^2}{2D} \quad (8)$$

The rate of heat dissipation,  $Q_t$ :

$$Q_t = h * A * (T_s - T_f) + Q_r + Q_c \quad (9)$$

where  $h$  is the convective heat transfer coefficient,  $A$  is the surface area,  $T_s$  is the surface temperature of the object cooling, and  $T_f$  is the temperature of the fluid or air which is cooling the object.

The pressure drop is determined with the CFD software in this study. The pressure drop value obtained using Equation (8) is within 2% of former value. The typical inlet temperature for the automotive radiator was 80°C (Vajjha *et al.*, 2010). As the pertinent coolant temperatures are possible during the severe driving conditions of the manufacturers of the vehicles alone, the nucleate boiling was not taken into account. In reality, when the nanofluid enter from the radiator top, it is inevitable that the flow will not be uniform. The inlet axial velocity in the present investigation was fixed as constant, and hence remained as an ideal real flow behaviour. The boundary conditions adopted were the pressure outlet and the no-slip for the velocity along the tube wall (Table 1). On the radiator outside wall, a fixed convective heat transfer coefficient of 50 W/m<sup>2</sup>K is realistic, being

**Table 1. Boundary conditions for the working fluids**

- The inlet nanofluid uniform axial velocity is taken as 18.32 m/s.
- The inlet air velocity is taken as 16 m/s.
- At walls, no slip condition is given.
- The outlet pressure given is 112.41 kPa.
- A constant heat flux of 15.21 kW/m<sup>2</sup> at the wall is given.
- The inlet nanofluid temperature is given separately from 80 to 110°C.
- A fixed free stream temperature of 30°C on the air side is given.
- The nucleate boiling is not taken into account.
- The tube walls are assumed to be perfectly smooth.
- Turbulent viscosity is considered.
- The compression work and the viscous dissipation are considered negligible.

an average value between the 72 km/h maximum road speed and inactive state for a fixed free stream temperature on the air side (Vajja *et al.*, 2010). An ambient air temperature of 30°C was selected. The viscous dissipation and the compression work are meagre for a radiator fluid flow. The heat dissipated by the nanofluid in full is absorbed by the air flow in the radiator.

For fin material, aluminium having 237.0 W/m-K, 903.0 J/kg-K, 2,702.0 kg/m<sup>3</sup> and 1,000-5,000 K thermal conductivity, specific heat, density and the minimum-maximum allowable temperature values respectively is used. The inlet axial velocity is kept uniform with an inlet temperature of 80°C in the flat tube of the radiator. The inlet velocity determines the Reynolds number of the flow, and the inlet temperature has been taken as 80°C. The actual non-uniformity flow pattern is idealized by the assumption of the inlet uniform velocity. Both the outlets of coolant and air are set to pressure outlets and the walls are set to convection and respective heat transfer coefficients are given. For the finer meshing of fins it is necessary to increase the mesh density for accurate result.

### Grid Independence Test

A grid with varying size and shape in different places is easier for implementation than a uniform fine grid. For grid refinement, a point in the polygon is joined with the central point of edges of the polygon and this distance is used to form control volumes around the nodes of the grid. The selection of these points at different locations results different flow terms and grids. Control Volume Finite Element type grid is resulted when this point is at the barycentre and calls this, the median grid. In a Shell-tube heat exchanger, fluid-fluid heat transfer takes place as one fluid runs through the shell and another through the tubes. The heat from the hotter fluid flowing inside a pipe system is taken by the cooler fluid flowing over the pipe system and carries it away. The two working fluids in a radiator is air and coolant and as the air flows through the radiator fins, the heat is transferred from the coolant in the tube to the air, which makes the coolant, still cooler. The fluids flow perpendicular to one another is similar to a heat exchanger with cross-flow, utilizing the coolant flowing left and right is cooled by the air moving up and down.

Rectangular cells are used in this study, for meshing the tube wall surfaces, face of the flat area and gap with subdivisions in the axial directions. The complex fin geometry and repeating tube geometry in the complete radiator is exorbitantly high for solving as the required elements are very high. Modelling simplified radiator tube with nanofluid medium and simplified shell with air

medium is used in this study by decreasing the number of elements as with the present computer technology, handling such a large mesh would be very cumbersome. The flow and heat transfer characteristics for a shell-tube heat exchanger could be thus determined. Mao *et al.* (2010) modelled a full radiator using 3-D model one-equation for energy and is the only one work. This model is not appropriate as there is an appreciable the temperature difference between the fluid and fins is high.

The physical model is subjected to the grid independence test to decide the best mesh face size. The grid independence for water at Re = 2,000 was checked using the four rectangular mesh faces, such as 430×650, 430×800, 430×950, and 430×1,100. The Nu for all the four-mesh faces were 15.5, 17, 17.5, and 19.5 respectively. The mesh face of size 430×950 registered the grid independence. A mesh face of size, 430×950 was hence adopted in all the simulation cases, being the best in terms of accuracy (Figure 2), except for laminar regime, for which 430×800 mesh face size was adopted. The Nu simulation results are compared with the Shah-London equation (Shah and London, 1978). The Nu results all agreed with each other and the Shah-London equation (Figure 2).

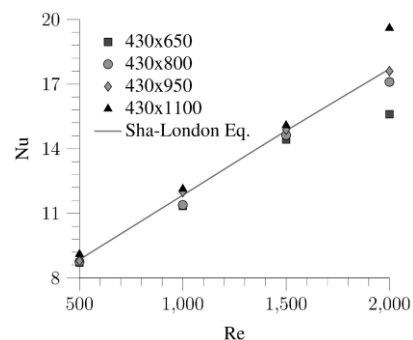


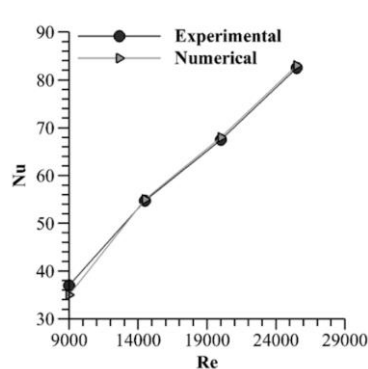
Figure 2. Details of the grid independence

In the grid test, wall  $y^+$  is used as the near-wall treatment appropriately with 4 different mesh sizes. Mesh refining at near wall region to get the wall to near-wall cell centroid distance or a reasonable wall  $y^+$ . The first point distance of the grid used in the calculation to achieve the condition  $y^+ < 1$  is 0.0012 mm. The results had a significant relation between the wall  $y^+$  and mesh configurations with respect to their Nu values and the 430 x 950 mesh configuration ensured grid independence, having the wall  $y^+$  values resolved at  $1 < y^+ < 2$ .

### Validation

The numerical analysis results, validated by matching the results of the experimental tests from

the study of the heat transfer enhancements of the nanofluids reported (Chougule *et al.*, 2014). The experimental results of an automobile radiator of 310 mm length, 20 mm width and 300 mm height with a total effective heat transfer area of 4.45 m<sup>2</sup> for the tube and fins using 1% CNT-water nanofluid at the inlet temperature of 90°C and Re 25,000 is used (Chougule, *et al.*, 2014). The results were demonstrated numerically for the accuracy and the reliability by varying the Nu with the Re, using water in the radiator. The comparison of the results of the numerical modelling with the experimental results is depicted in Figure 3. The per cent deviation of the numerical results from the experimental data is less with the maximum percentage difference of 5.7 and the data agree well with a high R<sup>2</sup> value of 0.999. Based on the confidence that the numerical simulations predict the experimental results for the water-only case, the same model pertains to simulate the nanofluids to distinguish between the potential heat transfer enhancements for the various cases under consideration.



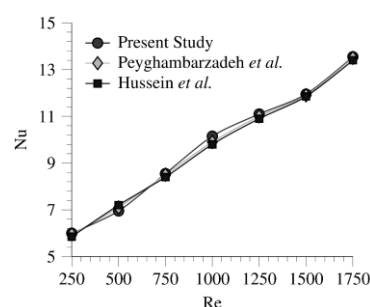
**Figure 3.** Validation of the simulated Nu with the experimental data of Chougule *et al.* (2014) at higher Re

Validation of the curve fitted data of the present study, in respect of the Nu at the Re 250-1,750 (Figure 4) showed an R<sup>2</sup> of 0.999 with the curve fitting of (Peyghambarzadeh *et al.*, 2013), using the Al<sub>2</sub>O<sub>3</sub> nanofluid and the experimental study (Hussein *et al.*, 2014a) using 1% SiO<sub>2</sub> nanofluid. The experimental results of an automobile radiator of 384 mm length, 22 mm width and 330 mm height with a total effective heat transfer area of 4.605 m<sup>2</sup> for the tube and fins using 0.65% Fe<sub>2</sub>O<sub>3</sub>-water nanofluid at the inlet temperature of 80°C and Re 1,000 is used (Peyghambarzadeh, *et al.*, 2013). The experimental results of an automobile radiator of 345 mm length, 30 mm width and 400 mm height with a total

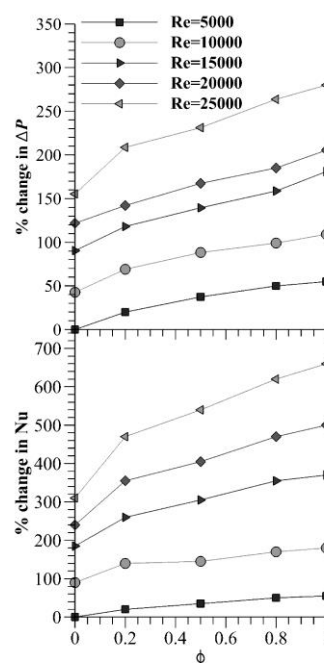
effective heat transfer area of 4.2 m<sup>2</sup> for the tube and fins using 2.5 % SiO<sub>2</sub>-water nanofluid at the inlet temperature of 80°C and Re 1,750 is used (Hussein, *et al.*, 2014a).

## Results and Discussion

There is enhancement in the Nu and the pressure drop for the CNT-water nanofluid in comparison with that of water for the Re from 5,000 to 25,000 and the nanoparticle concentration from 0 to 1% (Figure 5). The greatest enhancement in the Nu was



**Figure 4.** Validation of the simulated Nu with the experimental data of Peyghambarzadeh *et al.* (2013) and Hussein *et al.* (2014a) at lower Re



**Figure 5.** Effect of the nanoparticle volume fraction and the Re on Nu and pressure drop in 0.6096 m automotive radiator

85.4% by using 1% CNT-water nanofluid, compared with the results by using water at  $Re = 25,000$ . But, the per cent increase of  $Nu$  was found lower beyond 0.8% volume concentration, for all the  $Re$ . The energy exchange rate and Brownian motion of nanoparticles with inter particle and particle to wall collisions result the enhancement in heat transfer and  $Nu$ . Enhanced specific surface area and thermal conductivity augment the heat transfer enhancement. The influence of the  $Re$  on the  $Nu$  is stronger than the particle volumetric concentration. The removal of the heat in greater proportions could decrease the size of automobile radiators leading to an improved fuel and an overall engine cost economy. Similar heat transfer results of enhancement at 1% volume fraction was already reported using other nanofluids, and hence regarded as the critical volume concentration. The heat transfer enhancement was twenty-six per cent for the nanofluids with 1% of  $TiO_2$  nanoparticles, but at 2% volume fraction, it was 14% lower than the base fluid (Duangthongsuk and Wongwises, 2009).

The pressure drop also increases on adding the nanoparticles as observed for the  $Nu$ . The significant increase in the pumping power and pressure drop, in spite of the similar  $Re$  for the nanofluid and water might be due to the increased kinematic viscosity of the nanofluids. The highest enhancement in the pressure drop was found as 55% at the  $Re = 5,000$ . Similar enhancements in pressure drop and pumping power reported earlier, varies from 40-100% using other nanofluids. The pressure drop was found double in comparison to water by using the 4%  $Al_2O_3-H_2O$  nanofluid (Lee and Mudawar, 2007). There was a forty per cent increase in the pumping power and the pressure drop for the nanofluid in the plate heat exchanger than the base fluid (Vasu *et al.*, 2009). The performance index was found unity up to the  $Re = 10,000$  and higher than one for the  $Re$  from 15,000 to 25,000. The profound enhancement in heat transfer is more evident in the higher  $Re$  than the lower  $Re$  and highly economical. The per cent increase in heat transfer was found lower by 4.1% beyond the 0.8% volume concentration for all the  $Re$ . The optimum nanoparticle concentration was 0.8%. Vasu *et al.* (2009) reported that the nanofluids are found more useful in laminar regime, in comparison to the turbulent regime. Even though, there was 8% heat transfer improvement in the laminar flow, Pantzali *et al.* (2009) reported that there was no enhancement in heat transfer at the turbulent regime using the amorphous carbonic nanofluids.

The heat transfer performance of the selected radiator at the  $Re = 25,000$  using the 1% CNT-water nanofluid possesses 157.62% change in the  $Nu$  and 158% change in the pressure drop (Figure 6). The

heat transfer performances under the similar working conditions of the three new designs with changes only in the radiator length sizes of 0.4572, 0.3048, and 0.1524 m are found. The heat transfer performance of our new radiator assemblies was lower compared to the reference radiator, which has radiator lengths 25%, 50% and 75% lower than the reference radiator length. The  $Nu$  decreased 14.5%, 34% and 61.2% with the corresponding decreases in the pressure drop of similar magnitudes (14.3%, 34% and 61.2%). This drop might be due to a decrease in the surface area to coolant ratio.

By increasing the surface area of the proposed designs, changing the fin number from 384 to 840, 1,248, and 1,600; the heat transfer performance is enhanced. For this, the number of tubes are lowered from 33 to 25, 17, and 9 together with increasing the fin number/row from 12 to 35, 78, and 200 as laid out in Table 2, which sums up the radiator dimensions for the original and proposed designs. The heat transfer performance of the new radiator assemblies could be set up equal to that of the current radiator assembly by varying the fin spacing and the fin height values as illustrated in Table 2 and Figures 7(a) to 7(c). Figure 7(a) corresponds to the first proposed design with the dimensions given in the third column of Table 2; Figures 7(b) and 7(c) are the second and third proposed designs having the changes in dimensions as provided in the fourth and fifth columns of Table 2 respectively. Optimizing these radiator designs under the constant coolant flow and the airflow

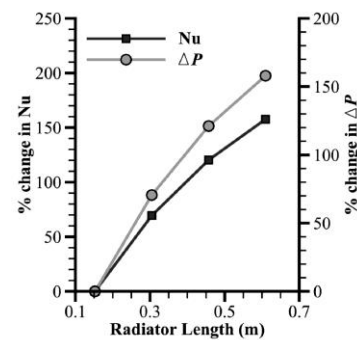


Figure 6. Effect of different radiator lengths on the  $Nu$  and the pressure drop

Table 2. Compactness of the different automotive radiator designs

| Dimensions of Designs |        |       |       |       |
|-----------------------|--------|-------|-------|-------|
| Radiator length (mm)  | 609.6  | 457.2 | 304.8 | 152.4 |
| Fin height (mm)       | 1.638  | 7.023 | 0.745 | 1.108 |
| Fin spacing (mm)      | 34.343 | 5.313 | 4.79  | 1.051 |
| No. of fins/row       | 12     | 35    | 78    | 200   |
| Total No. of fins     | 384    | 840   | 1248  | 1600  |
| No. of tubes          | 33     | 25    | 17    | 9     |

conditions, through a series of simulations provided the designs dissipating the same amount of heat. The details of the dimensions of the selected radiator are given in Figure 8. It is the third design proposed with the least length, fin spacing and number of tubes with values of 152.4, 1.051, and 9 mm respectively. The selected radiator design is having the maximum number of fins/row of 200 and fin number of 1,600 with an optimum fin height of 1.108 mm.

Figure 9 illustrates the effect of the different inlet radiator temperatures and the nanofluid volume concentration on the Nu at the Re 25,000. The Nu increases with the increasing of the inlet radiator temperature and the nanofluid volume concentration. The highest value of 115.85% enhancement in the Nu using the 1% nanofluid is found at the highest inlet temperature of 110°C. This has reference to a higher radiator heat transfer upon a higher nanofluid inlet temperature. The inlet temperature is almost a linearly increasing function of the Nu. The heat transfer enhancement is 116% when adding the nanoparticles in water, building up the inlet radiator temperatures. Even though, the heat transfer enhancement with respect to the inlet radiator temperature changed only from 16-27% that with respect to the nanofluid volume concentration varied 70-94%. The increase in the nanoparticle concentration results in an increase in the effective thermal conductivity, the Brownian motion, the particle migration and the energy exchange rate of the nanoparticles resulting in the

heat removal in radiator through an increase in the heat transfer. Fuel economy is the major achievement, consequent to the reduction in radiator size by this enhancement in heat transfer. The present observation showed that the inlet temperature had a lower contribution than the nanofluid volume fraction, but using all the three variables contributed much to the heat transfer enhancement. Similar experimental results of 31 to 46% Nu enhancement for 1 to 4% volume fractions, and 26 to 35% heat transfer enhancement behaviours were for 60 to 80°C inlet temperature (Hussein *et al.*, 2013b). Use of the nanofluids in the high-temperature applications was more beneficial (Vajjha and Das, 2009).

To meet a constant Nu of 152, the relationship of the radiator length with the number of fins and the number of tubes in the original radiator and the three new designs developed in the study, leading to an effective compact design of the radiator (Figure 10). The 25% radiator length reduction, 27.3% tube number reduction and 4.2% fin number addition could keep the same Nu by using the 1% CNT-water nanofluid at the inlet temperature of 110°C and the

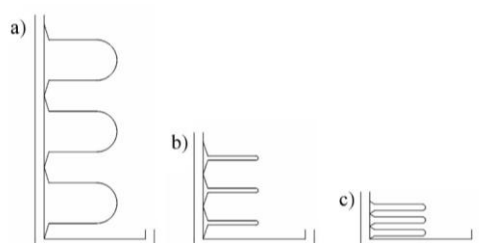


Figure 7. Design of the automotive radiators with length a) 457.2 mm, b) 304.8 mm, c) 152.4 mm

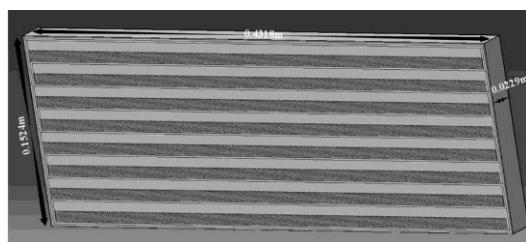


Figure 8. The compact radiator design developed with an effective cooling

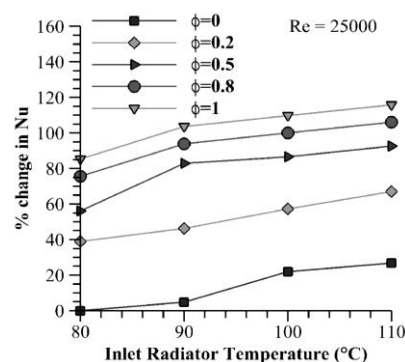


Figure 9. Effect of different radiator lengths on the Nu and the pressure drop

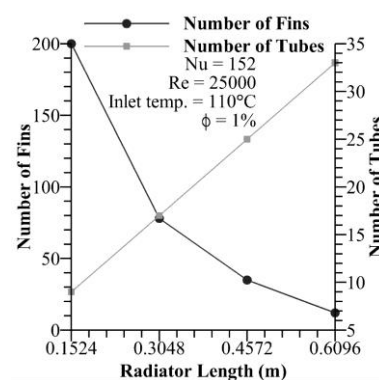


Figure 10. Effect of the compactness in the radiator cooling design

Re 25,000 in an automotive radiator with height 0.4318 m and width 0.02286 m.

The orange colour in the tube and blue colour around it in the velocity contour (Figure 11(a)) reveals high nanofluid velocity in the tube than the air velocity. The pressure drop results a maximum velocity between the tubes. The blue colour at the upstream shows low velocity of air and at the downstream, the vortex formation results lowering of the air velocity again. The orange colour near the inside wall of the tube justifies cooler nanofluid as it touches the tube wall in the temperature contour (Figure 11(b)) with a hotter area at the middle. Radiator centre is cooler as the fluid is slow, requiring more time for transferring the heat. At other parts of the radiator, the fluid flows faster resulting in hotter radiator areas. From the pressure contour (Figure 11(c)), inlet and outlet pressure distributions can be seen as high and low respectively, depicting the direction of the flow as forward and conforms to vice versa velocity contour variations. The pressure gets drop as the flow advances in the tube with a closer to zero value at the inlet (left of the figure). The value of total pressure drop is higher at the outlet.

There is one conduction (fin to tube wall) and two convection (nanofluid in tubes and air in fins). Kurabayashi (2007) reported that the standard heat transfer of present day radiators with 95°C inlet temperature is 140 kW of heat. The additional heat removing rate obtained for the 1% CNT-water nanofluid with the inlet temperature 110°C at the Re 25,000 is 194.63 kW.

## Conclusions

The cooling and the pressure drop performance of the CNT-water nanofluids at the nanoparticle concentration (0 to 1% by volume), the turbulent flow rates (Re 5,000-25,000) and the inlet flat tube temperatures from 80 to 110°C are investigated in the automobile radiator models in the present study. The results met with a very good agreement with the recent experimental results presented with the greatest deviation of 5.7%. The designing optimized the compactness for better effectiveness and economy using ANSYS FLUENT 15.0, as a replacement for the existing coolants in various scenarios. The major conclusions derived from the current investigation are given as follows.

The use of nanofluids enhances the heat transfer in the radiator in comparison to that of water. The heat transfer was enhanced by 116 per cent for the 1% CNT-water nanofluid, the inlet temperature 110°C and the Re 25,000. The increase in the flow rate and the volume concentration of CNT-water nanofluid together with the inlet radiator temperature increases the cooling performance. The effect of the nanoparticle volume concentration accompanied by the coolant flow rate is more pronounced than the inlet radiator temperature. An increase in the pressure drop is observed with the addition of nanoparticles and Re in a flat tube, but at the similar enhancement magnitudes of the Nu only, so that the performance index is one or higher. The reduction in the radiator length by 75% led to 61.2%

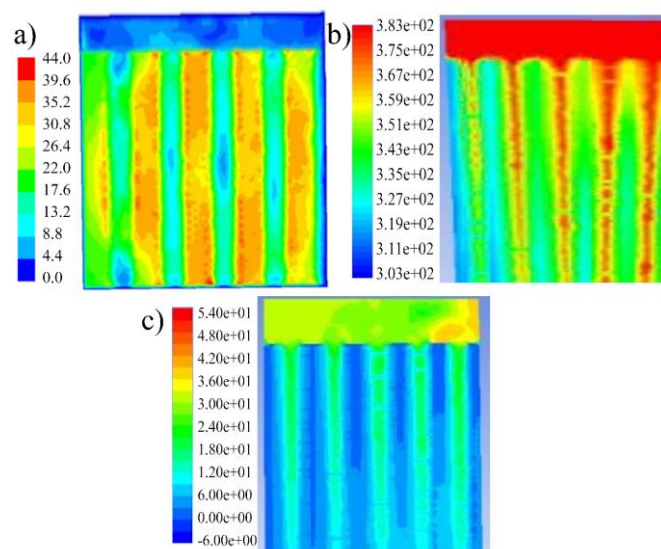


Figure 11. Contour plots for flow behaviour in terms of a) velocity b) temperature and c) pressure

reduction in the heat transfer performance. By changing the number of fins from 384 to 1,600, the fins per row from 12 to 200, reducing the fin spacing from 34.343 to 1.051, the number of tubes from 33 to 9 and varying the fin height from 1.638 to 7.023, could bring back the original heat transfer performance of the proposed designs. The areas of flow instabilities in the contour diagrams revealed the physical mechanism of flow behaviour and heat transfer. The additional heat removing rate enhancement obtained for the 1% CNT-water nanofluid with the average inlet temperature and road conditions with respect to the compact radiator developed in this study is 54.63 kW.

### Nomenclature

|             |                                                                                    |
|-------------|------------------------------------------------------------------------------------|
| $C$         | Specific Heat (J/kgK)                                                              |
| CFD         | Computational Fluid Dynamics                                                       |
| CNT         | Carbon Nano Tube                                                                   |
| $D$         | Diameter (m)                                                                       |
| $e$         | Favre-averaged specific internal energy (J)                                        |
| $f$         | Friction Factor                                                                    |
| $h$         | Favre-averaged specific enthalpy (J)                                               |
| $k$         | Thermal Conductivity (W/mK), Favre-averaged specific turbulence kinetic energy (J) |
| $L$         | Length (m)                                                                         |
| MWCNT       | Multiwalled Carbon Nano Tube                                                       |
| $Nu$        | Nusselt Number                                                                     |
| $P$         | Pressure (N/m <sup>2</sup> )                                                       |
| $Pr$        | Prandtl Number                                                                     |
| $R^2$       | Coefficient of determination                                                       |
| $Re$        | Reynolds Number                                                                    |
| $T$         | Temperature (K)                                                                    |
| $t_{ij}$    | Mean viscous stress tensor (Pa.s)                                                  |
| $u_i$       | Favre-averaged velocity vector (m/s)                                               |
| $u_\tau$    | Friction velocity (m/s)                                                            |
| $u$         | Velocity (m/s)                                                                     |
| $V$         | Volume (m <sup>3</sup> )                                                           |
| $\mu$       | Dynamic Viscosity (N s/m <sup>2</sup> ) Molecular viscosity (Pa.s)                 |
| $\mu_T$     | Eddy viscosity (m <sup>2</sup> /s)                                                 |
| $N$         | Kinematic viscosity (m <sup>2</sup> /s)                                            |
| $\rho$      | Density (kg/m <sup>3</sup> )                                                       |
| $\phi$      | Volume concentration                                                               |
| $\sigma^*$  | Closure coefficient (N s/m <sup>2</sup> )                                          |
| $\tau_{ij}$ | Favre-averaged specific Reynolds-stress tensor (Pa)                                |
| $\omega$    | Specific dissipation rate (J/Kg. s)                                                |
| $\Delta P$  | Pressure drop (Pa)                                                                 |
| $Av$        | Average                                                                            |
| $L$         | Laminar                                                                            |
| $Nf$        | Nanofluid                                                                          |
| $H$         | Hydraulic                                                                          |
| $T$         | Turbulent                                                                          |

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