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Comparative Analysis of Nanofluid Coolant in a Car Radiator using CFD

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Abstract

A computational analysis was performed to ascertain the effect of nanofluids on heat transfer in a flat tube heat exchanger of a car radiator. The nanofluids Al_2O_3/H_2O and CuO/H_2O were used with nanoparticles of different diameter in the range of 10-60 nm. A single-phase approach was implemented in the analysis. The heat transfer coefficient was calculated at various Reynolds numbers (250, 750, 1250, and 1750) with several nanoparticle concentration by volume of 1%, 3%, and 6%. Results indicated that the heat transfer rate increased with increase in the concentration by volume of nanoparticles. Whereas the decrease in the diameter of nanoparticle favored higher heat transfer rates. Therefore, the maximum heat transfer rate was observed at 6% concentration and at 10nm diameter size for both types of nanoparticles analyzed in this study for flat tube car radiator. The Al_2O_3/H_2O nanofluid showed higher heat transfer rates than the CuO/H_2O nanofluid at all Reynolds numbers.

Keywords: *Flat tube heat exchanger; Nanoparticles; Convection heat transfer coefficient; Nanofluids; Heat transfer; CFD*

1. Introduction

Various natural and forced convection techniques are used for improvement in the heat transfer processes. Enhancement in heat transfer can be accomplished by forced convection but with the addition of external equipment & devices the system becomes complex and operational cost increased. One method to boost the heat transfer rate of fluid flow is through altering the geometry of the pipe. However, a more effective technique is to upgrade the thermophysical properties of base fluids with the addition of nanoparticles. To study the effect of nanofluids on the heat transfer rate and friction factor, various experimental and numerical techniques have been used by the researchers. A large amount of data is available in the literature, however the work of some of the authors is presented here. Park and Pak [1] carried out computational study of ethylene glycol/water using laminar flow conditions in a flat tube heat exchanger with modification to the shape & dimensions. They demonstrated the results for an engine size of 1.8L, and Reynolds number (*Re*) ranging from 10 to 200, corresponding to the fluid flowrate of 18 – 75 L/min. They concluded that ethylene glycol/water mixture performed better than the water in terms of heat transfer. Vajjha et al. [2], performed numerical study on a flat tube automobile radiator using laminar flow conditions in a mixture of ethylene glycol/CuO with H_2O, and ethylene glycol/Al_2O_3 with H_2O. Their results showed 91% increment in local and average convective heat transfer coefficient, as compared to the base fluid, by using nanoparticles with a volume concentration of 10%. Huminic and Huminic [3], used CuO/ethylene glycol nanofluid and experimentally investigated the convective heat transfer coefficient in different cross-sectional tubes. They observed that at Re = 10 and 4% concentration of CuO by volume, the heat transfer coefficient improved by 19%. They also showed that the performance of nanofluid was linearly dependent on the nanoparticle concentration, whereas the heat exchanger with flat tubes performed better than the one with circular and elliptical tubes. Hussein et al. [4] performed experimental and numerical study utilizing SiO_2/water nanofluid for the automotive cooling system to investigate heat transfer performance. Their results illustrated a prominent increase in the heat transfer rate as well as the friction factor. Li and Xuan [5] experimentally analyzed the performance of

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CuO/H_2O nanofluid for the flow characteristics and convective heat transfer coefficient. They presented that the friction factor for nanofluid was in good comparison to that of the base fluid. This was observed because the size of the nanoparticles is so insignificant that they do not cause a great increment in the friction factor. Elsebay et al. [6] in their work, concluded that with the application of nanofluids in the radiators, the size of the radiator can be reduced owing to the better heat transfer performance. However, with the increase in the concentration of the nanoparticle in the base fluid, the pumping power also increases. Naraki et al. [7] performed experimental study and concluded that the temperature of the nanofluid affects the overall heat transfer coefficient adversely. At the concentration by volume of 0.15% and 0.4% of CuO nanoparticles in water, the enhancement in overall heat transfer coefficient is 6% and 8%, respectively, as compared to the base fluid (water). Devireddy et al. [8], in their study on automobile radiator stated that the presence of TiO_2 nanoparticles in 40:60 percent ethylene glycol/H_2O sufficiently improved the heat transfer rate. And, also stated that the heat transfer strongly depends upon the quantity of nanoparticles added in the base fluid. Similar results were reported by Ali et al. [9] in their experimental investigations using ZnO/water nanofluid. They reported stated that by using 0.002 concentration by volume of nanoparticles the fluid, enhancement of 46% in heat transfer rate can be obtained in automobile radiator. Arani and Amani [10] worked on to enhance the convective heat transfer coefficient in the heat exchanger using different nanofluids. Hojjat et al. [11] Kayhani et al. [12] performed experimental study to analyze the heat transfer and friction factor using nanofluids at different Reynolds number and concentration by volume. Their results showed good agreement with the data in the literature. Various base fluids such as ethylene glycol, water, and glycerol are being used for many years, but their performance is not as good as the nanofluids as they have poor thermal conductivity. Nanofluids surpass them in terms of heat transfer rate due to their ability to conduct heat more rapidly [13, 14]. Apart from the experimental techniques to study the effect of using nanofluids, instead of regular fluids, on the heat transfer rate, numerical methods have also emerged as promising techniques to carry out the analysis using nanofluids. Dehghandokht et al. [15] performed numerical study on the meso-channels heat exchanger by using the water & ethylene glycol/water mixture coolants. They evaluated the pressure drop, heat transfer rates, and temperature drop. Their results showed good agreement with the experimental results. Leong et al. [16] used CuO/ethylene glycol nanofluids in car radiator cooling system and reported heat transfer enhancement. Peyghambarzadeh et al. [17] also reported similar results using CuO and Fe_3O_2 nanoparticles in base fluid. Nanofluid consisting of SiO_2/water mixture was used by Ferrouillat et al. [18] under heating and cooling conditions at various inlet temperatures. They reported a maximum of 50% increase in the heat transfer coefficient.

The thermal performance of nanofluids was observed based on the Reynolds number and concentration by volume criteria in the above cited work. However, these are not the only factors that affects the heat transfer. Another important factor is the size of the nanoparticles. Most of the studies available in the literature only study the effect of one or two factors. The current study is aimed to compute the thermo-physical characteristics of CuO/H_2O and Al_2O_3/H_2O nanofluids and to evaluate the enhancement in cooling rate/heat transfer rate using flat tube car radiator. The aim of this research is to investigate the enhancement in the heat transfer using nanofluids in order to improve the cooling performance of an automobile radiator. The parameters tested in this study are the Reynolds number, size of the nanoparticles, and their concentration by volume in the base fluid. The rest of the manuscript is organized as follows. Section 2 is dedicated to the numerical technique including the computational domain where the geometry and the mesh is presented, and validation of the model is done. Governing equations are presented in Section 3 along with the thermo-physical properties of the nanofluids. The results are presented in Section 4 with the discussion of the results. Finally, conclusions and future recommendations are presented in Section 5.

2. Numerical Technique

A three-dimensional model of the a single flat tube was considered for the analysis instead of the complete heat exchanger. This technique is similar to the work of Ali and Kamran [19] where they took just a single passage for the gas turbine blade instead of the complete blade with multiple passages and worked on to enhance the heat transfer rate. Computational time is greatly reduced with not much loss in the accuracy of the results. The simulations were performed on commercial software Ansys/Fluent. Mesh independence study was performed to be certain that the mesh quality does not affect the results. For the pressure-velocity coupling Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm was used. Second order discretization schemes were used to discretize the domain in space. In addition, slip velocity was kept zero along with zero mass transfer, and the continuum state was considered for solid particles. Single phase flow was considered only. The residuals for the Continuity, Momentum, and Energy equations were set at 10^{-6} as the convergence criteria to get a good accuracy for the results. Calculations were performed under laminar flow conditions to analyze the heat transfer coefficient, heat transfer rate, and pressure drop in order to evaluate the performance of radiator.

2.1 Geometry

The model of a typical single tube of a flat-tube heat exchanger of a car radiator are shown in Fig. 1 along with the dimensions and the technical details presented in Table 1.

Table 1: Dimensions and the technical details

Dimension of a single tube (HxWxL)	0.0012x0.036x1.380 m^3
No. of tubes	68
No. of tubes in a row	34
Wall thickness of each tube	0.0005m

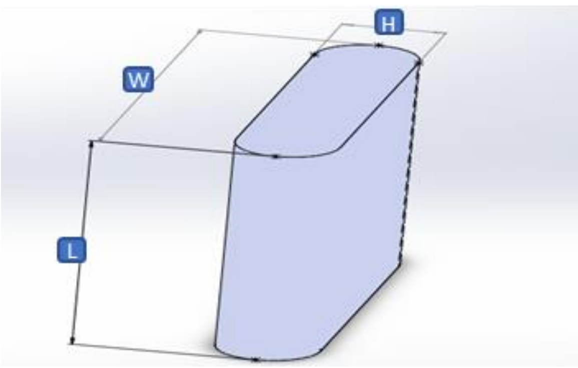

Fig. 1. Geometry of a typical flat tube heat exchanger

2.2 Meshing and model validation

The geometry used in this analysis is displayed in Fig. 2 along with the discretized mesh. For the analysis, a quarter portion of the geometry shown in Fig. 1 is used to reduce the number of elements in the mesh in order to decrease the computational time. Symmetry conditions are used on the cut portions to give the full domain. This decreased the number of elements from 80x10^4 to 20x10^4. Mesh independence study results are displayed in Fig. 3.

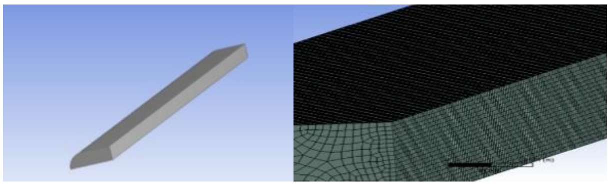

Fig. 2. Geometry and mesh of the one-fourth portion of the domain

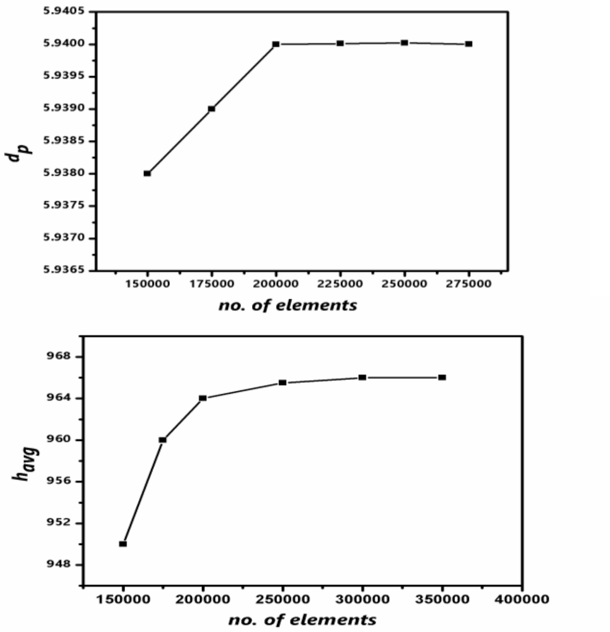

Fig. 3. Dependence of pressure drop (*dp*) and average convective heat transfer coefficient (h_{avg}) on the number of mesh elements

For the validation of the model and the solution procedure, the results of the average convective heat transfer coefficient for the simulation were compared with the experimental results of Elsebay et al. [6] at *Re* of 1750 as presented in Fig. 4. The results were found to be in good agreement with the maximum error of 2%.

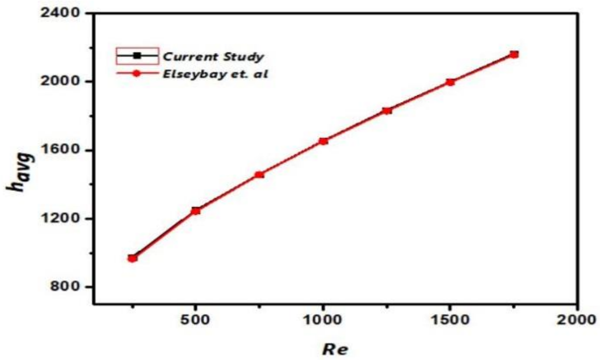

Fig. 4. Validation with the experimental results of Elseybay et al. [6] at *Re* = 1750

3. Governing Equations

The equations that govern the flow are presented below as continuity (eq. 1), momentum (eq. 2), and energy (eq. 3) [6]:

$$div(\rho\vec{V}) = 0 \tag{1}$$

$$div(\rho\vec{V}\vec{V}) = -grad(P) + \nabla \bullet (\mu\nabla\vec{V}) \tag{2}$$

$$div(\rho\vec{V}C_pT) = div(\kappa gradT) \tag{3}$$

These equations were solved by the numerical code using numerical techniques. A few assumptions were made for this analysis, which are mentioned below:

- Flow was incompressible
- Flow regime was laminar
- Viscous dissipation effects are neglected
- Single-phase approach used

3.1 Thermo-physical properties of the nanofluids

The effective thermo-physical characteristics of nano fluids were calculated by Corcione et al. [20]. Assuming, the flow incompressible and steady state, with uniform concentrations of nanoparticles throughout the system, the values of thermo-physical properties such as specific heat, density, thermal conductivity, and viscosity were found from the following given correlations:

Density of nanofluid:

$$\rho_{nf} = \phi_v \rho_{np} + (1-\phi_v)\rho_{bf} \tag{4}$$

where ρ is the density and Φ_v represents the concentration by volume. The subscripts *nf*, *np*, and *bf* represent the nanofluid, nanoparticle, and base fluid respectively.

Specific heat of nanofluid:

$$(\rho C_p)_{nf} = \phi_v (\rho C_p)_{np} + (1-\phi_v)(\rho C_p)_{bf} \tag{5}$$

where C_p is the specific heat capacity.

Dynamic viscosity of nano fluid:

$$\mu_{nf} = \mu_{bf} \left(\frac{1}{\left(1 - 34.87 \left(\frac{d_{np}}{d_{bf}} \right)^{-0.3} \phi_v^{\,1.03} \right)} \right) \tag{6}$$

where μ is the dynamic viscosity, and *d* represents the diameter.

Thermal conductivity of nanofluids:

$$k_{nf} = k_{bf} \left[1 + 4.4 Re_{np}^{0.4} Pr_{bf}^{0.66} \left(\frac{T}{T_{fr}} \right)^{10} \left(\frac{k_{np}}{k_{bf}} \right)^{0.03} \phi_v^{\,0.66} \right] \tag{7}$$

where *k* is the thermal conductivity, *Re* is the Reynolds number, *Pr* is the Prandtl number, and *T* is the temperature.

The values of diameter and Reynolds number were calculated from below given equations at the given values of other parameters:

$$d_{bf} = 0.1\left(\frac{6M}{N\pi\rho_{bf}}\right)^{1/3} \quad (8)$$

$$Re_{np} = \frac{2\rho_{bf}K_BT}{\pi\mu_{bf}^2 d_{np}} \quad (9)$$

where K_B is the Boltzmann constant, M is the molar mass, and N is the number of moles.

The properties of nanoparticles and the base fluid used in this study are given in Table 2.

Table 2. Thermo-physical properties of nanoparticles and the base fluid

Material	Sp. heat capacity (J/kg.K)	Thermal conductivity (W/m.K)	Density (kg/m^3)	Dynamic viscosity (Pa.s)
Al_2O_3	765	40	3970	-
CuO	540	18	6000	-
Water	4182	0.611	998.8	0.00089

3.2 Average convective heat transfer coefficient

Heat transfer rate (Q) can be determined using Newton's of cooling as discussed by Park et al. [1]:

$$Q = h_{avg} \times A_s \times (T_b - T_s) \quad (10)$$

where A_s is the surface area of the tube.

Hydraulic diameter (D_h) of a flat tube depends upon perimeter and cross-sectional area of flat tube and can be obtained by the following equation:

$$D_h = (4 \times A) / P_m \quad (11)$$

Following formula was used to calculate the cross-sectional area (A) of the flat tube. Cross-sectional area depends upon the width and the height of the flat tube as shown in the following relation:

$$A = (\pi \times H^2)/4 + (W - H) \times H \quad (12)$$

The perimeter (P_m) of the flat tube can be obtained by the following equation:

$$P_m = \pi \times H + 2 \times (W - H) \quad (13)$$

Bulk temperature (T_b) was found from average value of inlet temperature (T_{in}) and outlet temperature (T_{out}) of flat tube as:

$$T_b = (T_{in} - T_{out}) / 2 \quad (14)$$

Heat transfer rate depends upon the area, velocity (V), density, specific heat capacity, and the change in the temperature. Heat transfer rate was calculated using the following equation:

$$Q = \rho \times A \times V \times C_p \times (T_{in} - T_{out}) \quad (15)$$

The average heat transfer coefficient (h_{avg}) was determined using following equation:

$$h_{avg} = \rho \times A \times V \times C_p \times (T_{in} - T_{out}) / (A_s \times (T_b - T_s)) \quad (16)$$

Reynolds number was calculated using equation (17) is shown below, which depends upon viscosity of working fluid, density, hydraulic diameter of flat tube, and velocity of fluid flow:

$$Re = (\rho \times V \times D_h) / \mu \quad (17)$$

4. Results and Discussion

Results were calculated for distilled/pure H_2O at Reynolds numbers of 250, 750, 1250, and 1750. Fig. 5 demonstrates the behavior of heat transfer coefficient against nanoparticles concentrations for Al_2O_3 at Reynolds number of 250 for varying nanoparticle diameters. The figure shows that as nanoparticle size decreases heat transfer coefficient escalates and similarly with the growth in the concentration of nanoparticles heat transfer also coefficient enhances. Fig. 6, 7, and 8 show the variations in heat transfer coefficient against nanoparticle concentrations of (1-6%), at varying nanoparticle diameters with different values of Reynolds number as 750, 1250 and 1750, respectively.

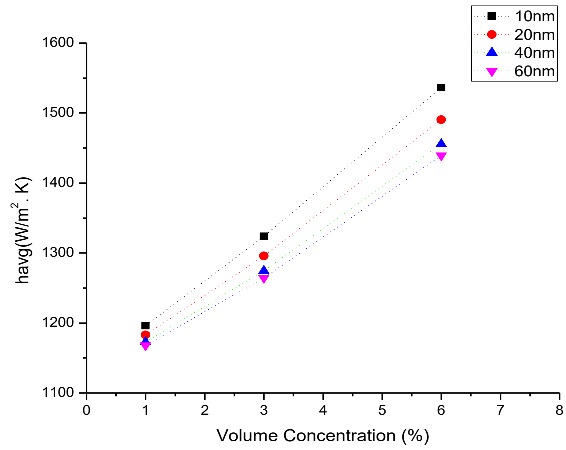

Fig. 5. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (Al_2O_3) in base fluid (water) with different nanoparticle size at *Re* = 250.

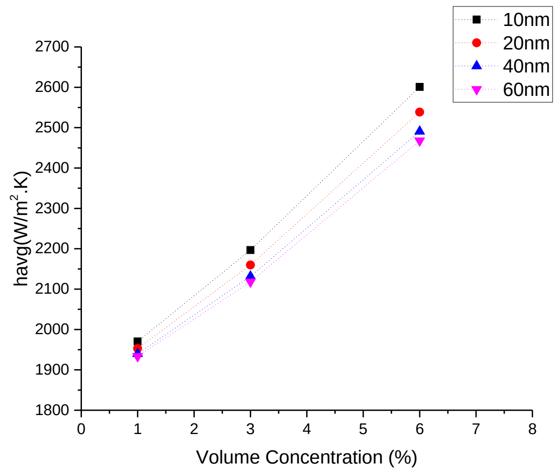

Fig. 6. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (Al_2O_3) in base fluid (water) with different nanoparticle size at *Re* = 750.

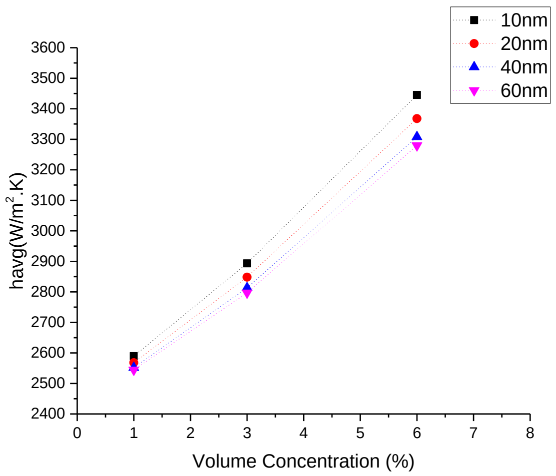

Fig. 7. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (Al_2O_3) in base fluid (water) with different nanoparticle size at *Re* = 1250.

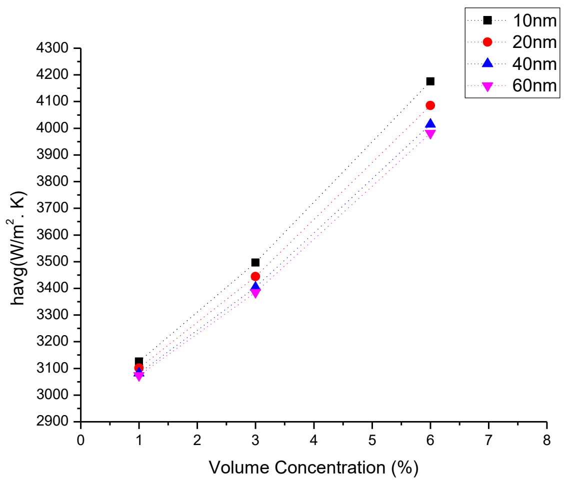

Fig. 8. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (Al_2O_3) in base fluid (water) with different nanoparticle size at *Re* = 1750.

Fig. 9 shows the behavior of heat transfer coefficient against nanoparticles concentrations of CuO at Reynolds number 250 for varying nanoparticle diameters. The results of Fig. 9-12 are similar to the results shown in Fig. 5-8 at the same corresponding Reynolds numbers. The only difference in Fig. 9-12 and Fig. 5-8 is the different type of nanoparticle used. It is observed that the increase and concentration of nanoparticle by volume, and the decrease in nanoparticle size is enhances the heat transfer rate.

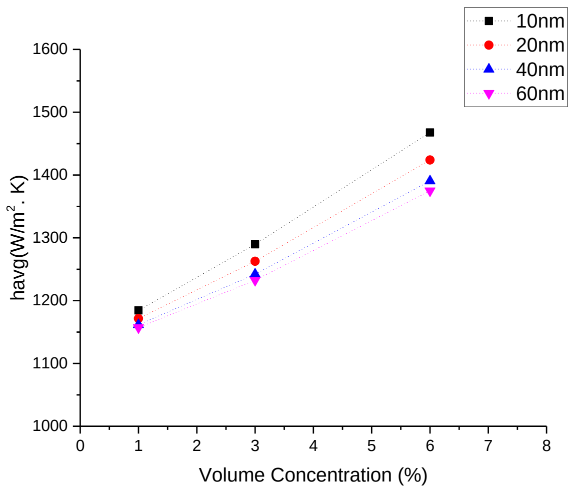

Fig. 9. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (CuO) in base fluid (water) with different nanoparticle size at *Re* = 250.

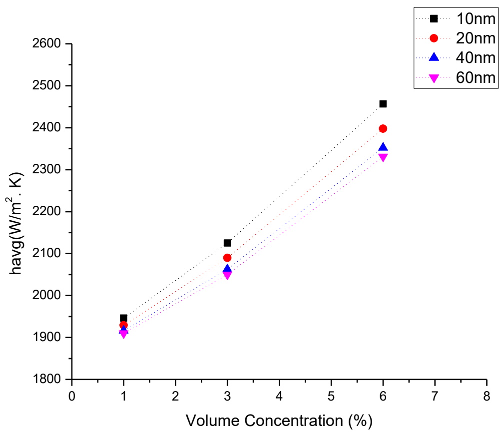

Fig. 10. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (CuO) in base fluid (water) with different nanoparticle size at *Re* = 750.

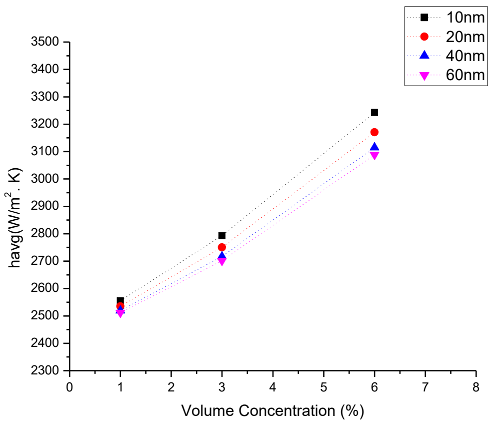

Fig. 11. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (CuO) in base fluid (water) with different nanoparticle size at Re = 1250.

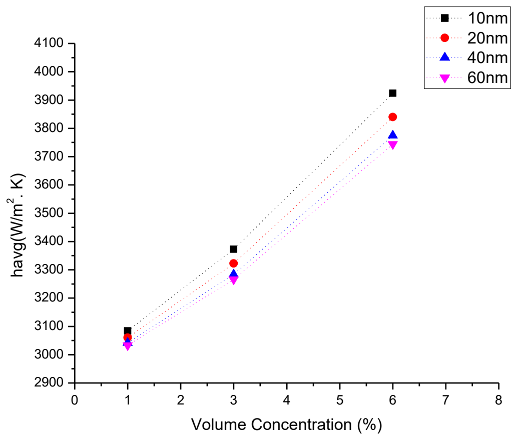

Fig. 12. Average convective heat transfer coefficient vs concentration by volume of nanoparticle (CuO) in base fluid (water) with different nanoparticle size at *Re* = 1750.

Fig. 13 shows the comparative analysis of the two nanofluids used in this study; Al_2O_3/H_2O and CuO/H_2O based on the heat transfer coefficient at different Reynolds numbers. It is observed that with the increase in the Reynolds number there is substantial growth in the heat transfer coefficient for both the nanofluids. The nanofluid with Al_2O_3 nanoparticles performed better as compared to CuO/water nanofluid in terms of average heat transfer coefficient. The maximum value of h_{avg} is obtained at *Re* of 1750 for Al_2O_3/water nanofluid.

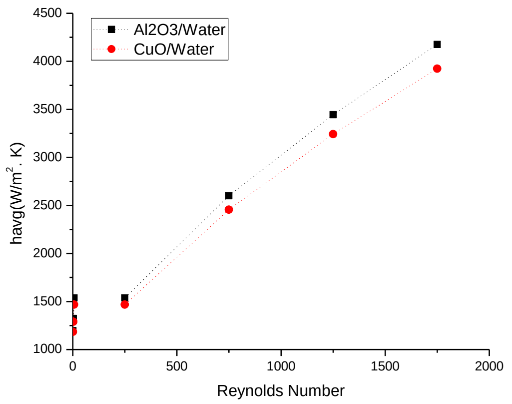

Fig. 13. Average convective heat transfer coefficient vs Reynolds number for two nanofluids (Al2O3/water and CuO/water) at 6% concentration by volume with 10nm particle size.

Similar results to Fig. 13 are shown in Fig. 14 in the bar chart format. The Al_2O_3/water nanofluid performs better at all the Reynolds number than CuO/water nanofluid in the Reynolds number range of 250-1750.

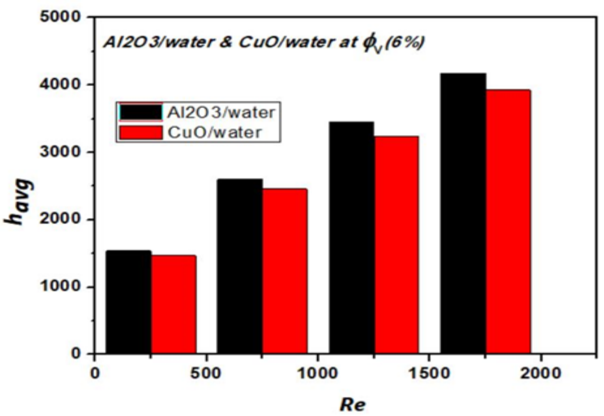

Fig. 14. Average convective heat transfer coefficient vs Reynolds number for two nanofluids (Al_2O_3/water and CuO/water) at 6% concentration by volume with 10nm particle size.

Fig. 15 shows the overall improvement in heat transfer rate for nanofluids Al_2O_3/H_2O and CuO/H_2O as compared to the base fluid (water). The results are normalized with the value of the base fluid. Both the nanofluids performed better than the base fluid, however Al_2O_3/ H_2O nanofluid performed better than the CuO/ H_2O nanofluid.

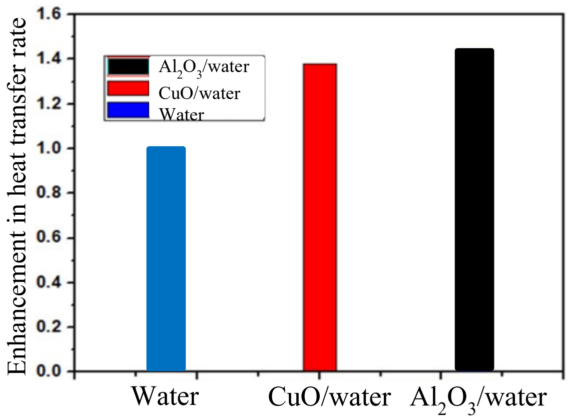

Fig. 15. Overall enhancement in the heat transfer rate as compared to the base fluid

5. Conclusion

In this work, heat transfer performance in a flat tube of an automobile radiator has been investigated numerically, with two working fluids: Al_2O_3/water & CuO/water by using water as a base fluid. From the analysis performed and the results obtained, following conclusions can be drawn:

- Addition of Al_2O_3 and CuO nanoparticles in water can enhance the heat transfer rate of an automobile radiator. The enhancement directly depends upon the concentration and size of the nanoparticles added to the base fluid.
- The increase in heat transfer coefficient reached up to 44% and 38% for Al_2O_3/water and CuO/water, respectively, with 10nm diameter and 6% concentration by volume in the base fluid.
- For the given range of the concentration by volume (1-6%) and nanoparticle size (10-60nm), the enhancement in the heat transfer is proportional to the of the nanoparticles in the base fluid, however it is inversely proportional to the size of the nanoparticles.
- Heat transfer rate can be increased by using any of the two nanofluids tested in this study, however Al_2O_3/water showed better results than the CuO/water nanofluid.
- Depending upon the cooling capacity of the car radiator, its size can be reduced by using any of the two tested nanofluids.
- Smaller sized radiator with efficient heat transfer can lead to better space utilization and can improve the fuel economy of the automobile.

For future work, more nanofluids should be tested with a wider range of concentration and particle size. Optimization should be carried out using an algorithm to provide the most suitable combination for given set of conditions.

Nomenclature

A	Cross sectional area of the tube, m^2
C_p	Specific heat, kJ/kg. K
D_h	Hydraulic diameter, m
H	Height of the flat tube, m
L	Length of the flat tube, m
W	Width of the flat tube, m
h	Convective heat transfer coefficient, $W/m^2.K$
k	Thermal conductivity, W/m. K
p	Pressure, Pa
Q	Heat transfer rate, J/s
q	Heat flux, W/m^2
T	Temperature, K
V	Velocity, m/s

Greek Symbols

μ	Dynamic viscosity, Pa.s
ρ	Density, kg/m^3
Φ	Particle concentration by volume, %

Subscripts

avg	Average
s	Surface
b	Bulk
in	Inlet
out	Outlet
np	Nanoparticle
bf	Base fluid
nf	Nanofluid

Non-dimensional Numbers

Re	Reynolds number

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