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Performance investigation of an automotive car radiator operated with nanofluid-based coolants (nanofluid as a coolant in a radiator)

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ABSTRACT

Water and ethylene glycol as conventional coolants have been widely used in an automotive car radiator for many years. These heat transfer fluids offer low thermal conductivity. With the advancement of nanotechnology, the new generation of heat transfer fluids called, “nanofluids” have been developed and researchers found that these fluids offer higher thermal conductivity compared to that of conventional coolants. This study focused on the application of ethylene glycol based copper nanofluids in an automotive cooling system. Relevant input data, nanofluid properties and empirical correlations were obtained from literatures to investigate the heat transfer enhancement of an automotive car radiator operated with nanofluid-based coolants. It was observed that, overall heat transfer coefficient and heat transfer rate in engine cooling system increased with the usage of nanofluids (with ethylene glycol the basefluid) compared to ethylene glycol (i.e. basefluid) alone. It is observed that, about 3.8% of heat transfer enhancement could be achieved with the addition of 2% copper particles in a basefluid at the Reynolds number of 6000 and 5000 for air and coolant respectively. In addition, the reduction of air frontal area was estimated.

1. Introduction

Continuous technological development in automotive industries has increased the demand for high efficiency engines. A high efficiency engine is not only based on its performance but also for better fuel economy and less emission. Reducing a vehicle weight by optimizing design and size of a radiator is a necessity for making the world green. Addition of fins is one of the approaches to increase the cooling rate of the radiator. It provides greater heat transfer area and enhances the air convective heat transfer coefficient. However, traditional approach of increasing the cooling rate by using fins and microchannel has already reached to their limit [1]. In addition, heat transfer fluids at air and fluid side such as water and ethylene glycol exhibit very low thermal conductivity. As a result there is a need for new and innovative heat transfer fluids for improving heat transfer rate in an automotive car radiator.

Nanofluids seem to be potential replacement of conventional coolants in engine cooling system. Recently there have been considerable research findings highlighting superior heat transfer performances of nanofluids. Yu et al., [2] reported that about 15–40% of heat transfer enhancement can be achieved by using various types of nanofluids. With these superior characteristics, the size and weight of an automotive car radiator can be reduced without affecting its heat transfer performance. This translates into a better aerodynamic feature for design of an automotive car frontal area. Coefficient of drag can be minimized and fuel consumption efficiency can be improved.

Therefore, this study attempts to investigate the heat transfer characteristics of an automotive car radiator using ethylene glycol based copper nanofluids as coolants. Thermal performance of an automotive car radiator operated with nanofluids is compared with a radiator using conventional coolants. The effect of volume fraction of the copper nanoparticles with basefluids on the thermal performance and potential size reduction of a radiator were also carried out. Copper nanoparticles were chosen in this study since it has higher thermal conductivity compared to other nanoparticles such as alumina.

2. Nanofluids in enhancing thermal conductivity

Eastman et al. [3] reported that the thermal conductivity of ethylene glycol nanofluids containing 0.3% volume fraction of copper particles can be enhanced up to 40% compared to that of

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Nomenclature

A	total heat transfer area, m^2
A_f/A	fin area on one side/total area
C	heat capacity rate, W/K
C_p	specific heat J/kg K
C^*	$C_{p,\ min}/C_{p,\ max}$
D_h	hydraulic diameter, m
DP	pressure drop, Pa
EG	ethylene glycol
f	fanning friction factor, dimensionless
G	mass velocity, kg/m^2s
H	total water flow length, m
h	heat transfer coefficient, W/m^2 K
j	Colburn factor, dimensionless
k	thermal conductivity, W/m K
L	fin length for heat conduction from primary to midpoint between plates for symmetric heating, m
NTU	number of heat transfer units
Nu	Nusselt number
P	pumping power
Pr	Prandtl number
Re	Reynolds number
t	fin thickness, m
T	temperature, K
U	overall heat transfer coefficient W/m^2 K
W	mass flow rate
σ	minimum free flow area/frontal area
η_f	fin efficiency of plate fin
η_o	total surface temperature effectiveness
μ	dynamic viscosity, Ns/m^2
ν	volumetric flow rate, m^3/s
ρ	density, kg/m^3
ψ	volume fraction of particles
ε	heat exchanger effectiveness
α	total one side of heat transfer area/total volume

Subscripts

fr	frontal area
a	air
nf	nanofluids
f	fluid (basefluid)
p	particles
in	inlet

ethylene glycol basefluid. Hwang et al. [4] found that thermal conductivity of the nanofluids depends on the volume fraction of particles and thermal conductivity of basefluid and particles. Lee et al. [5] measured the thermal conductivity of low volume concentration of aqueous alumina (Al_2O_3) nanofluids produced by two-step method. Authors inferred that the thermal conductivity of aqueous nanofluids increases linearly with the addition of alumina particles. Thermal conductivity of zinc dioxide–ethylene glycol (ZnO–EG) based nanofluids was investigated by Yu et al. [6]. They obtained about 26.5% enhancement of thermal conductivity by adding 5% volume fraction of zinc dioxide nanoparticles in ethylene glycol. Present study concluded that size of nanoparticles and viscosity of the nanofluids played a vital role in thermal conductivity enhancement ratio of them.

Mintsa et al. [7] investigated the effect of temperature, particle size and volume fraction on thermal conductivity of water based nanofluids of copper oxide and alumina. Authors suggested that thermal characteristics can be enhanced with increase of particles' volume fraction, temperature and particle size. Authors found that the smaller the particle size, the greater the effective thermal conductivity of the nanofluids at the same volume fraction. Contact surface area of particles with fluid and Brownian motion can be increased when smaller particles are used in the same volume fraction. This consequently increased thermal conductivity of nanofluids.

3. Nanofluids in enhancing forced convective heat transfer and relevant pressure drop

Namburu et al. [8] numerically analyzed turbulent flow and heat transfer to three types of nanofluids namely copper oxide (CuO), alumina (Al_2O_3) and silicon dioxide (SiO_2) in ethylene glycol and water, flowing through a circular tube under constant heat flux. Results revealed that nanofluids containing smaller diameter of nanoparticles produce higher viscosity and Nusselt number. Nusselt numbers are also increased at higher volume fraction of particles. It is observed that at a constant heat flux (50 W/cm^2) with a constant Reynolds number (20,000), heat transfer coefficient of 6% CuO nanofluid has increased 1.35 times than that of the basefluid. At the same particle volume fraction, CuO nanofluid produced higher heat transfer coefficient compared to that of other types of nanofluids.

Ding et al. [9] found that convective heat transfer coefficient of nanofluids has the highest magnitude at the entrance length of a tube. It starts decreasing with axial distance and eventually accomplish at a constant value in the fully developed region. At a given flow and particle concentration, aqueous carbon nanoparticles offer highest improvement. Zeinali et al. [10] experimentally investigated convective heat transfer to alumina–water (Al_2O_3/water) nanofluids in laminar flow inside a circular tube with constant wall temperature under different concentrations of nanoparticles. They obtained augmentation of heat transfer coefficient of nanofluid with increase of nanoparticle concentration. They also obtained greater heat transfer coefficient of nanofluid in comparison to that of distilled water basefluid at a constant Peclet number. Authors have reported that the heat transfer augmentation results are much higher in experimental observation than that of predicted results. Yu et al. [11] conducted heat transfer experiments of nanofluids containing 170-nm silicon carbide particles at 3.7% volume concentration. The results showed that heat transfer coefficients of nanofluids are 50–60% greater than those of basefluids at a constant Reynolds number.

Kim et al. [12] investigated effect of nanofluids on the performances of convective heat transfer coefficient of a circular straight tube having laminar and turbulent flow with constant heat flux. Authors have found that the convective heat transfer coefficient of alumina nanofluids improved in comparison to basefluid by 15% and 20% in laminar and turbulent flow, respectively. This showed that the thermal boundary layer played a dominant role in laminar flow while thermal conductivity played a dominant role in turbulent flow. However, no improvement in convection heat transfer coefficient was noticed for amorphous particle nanofluids.

Ku et al. [13] conducted an experimental study on pressure drop of nanofluids containing carbon nanotubes in a horizontal tube. Authors had reported that the pressure drops of nanofluids became almost the same as that of distilled water (basefluid). The viscosity of nanofluids decreases with the shear rate and thus, the difference of pressure drop between nanofluids and distilled water at elevated flow rate also decreases.

Duangthongsuk and Wongwises [14] conducted an experimental study on the heat transfer performance and pressure drop of TiO_2–water nanofluids, flowing in a turbulent flow regime. They reported that the pressure drop of nanofluids was slightly higher than that of the basefluid and increased with the increasing volume concentrations.

4. Input data and operating characteristics

Necessary input data were taken from literature to perform the analysis in this study. The characteristics of the radiator considered in this study are obtained from Vasu et al. [15] and Kays and London [16] as shown in Tables 1 and 2. The radiator with given specifications is mounted on the turbo-charged diesel engine of type TBD 232V-12. It is a cross-flow compact heat exchanger with unmixed air and fluid as coolants. This compact heat exchanger consists of 644 brass flat tubes and 346 continuous copper fins. [15]. Compact heat exchanger is a unique and special class of heat exchanger having a large heat transfer area per unit volume. In addition, flat tubes are more popular in automotive applications due to the lower drag profile compared to round tubes [15]. These two features will enhance the cooling rate and at the same time minimizing the flow resistance [15]. Other important thermal properties of nanofluids are calculated from empirical correlations shown in Section 5. Thermo-physical properties of ethylene glycol are presented in Table 3 and the thermal conductivities of nanofluids used in this analysis are obtained from Eastman et al. [3] as shown in Fig. 1. The calculated thermo-physical properties of nanofluids were assumed constant and their stability also remained invariable after their application in an automotive car radiator. The thermal conductivity obtained from Eastman et al. [3] is considered as benchmark value. Higher value could be obtained in real application because the coolant of an automotive radiator is usually operated at a higher temperature. This condition will enhance the Brownian motion of the nanoparticles and subsequently increase thermal conductivity of the nanofluids. The density, viscosity and heat capacity of nanofluids were assumed to generate no substantial effects on those properties of basefluid because only a maximum of 2% volume fraction of nanofluids were considered in this study.

5. Mathematical formulation of ethylene glycol based copper nanofluids in an automotive car radiator

Mathematical correlations shown in Section 5.1 are taken from references [15–17,19]. In this paper a comparison is made between the heat transfer performance of radiator by operating with ethylene glycol and nanofluid coolants. It highlighted not only the influence of nanofluids but also volume fraction of copper nanoparticles to the heat transfer rate of a radiator. Described equations are being incorporated to aid the comparison. Thermal performances of a radiator can be calculated using Eqs. (1)–(27), the relevant data are shown in Tables 1–3 and Fig. 1. Calculations were done on air and coolant sides.

Table 1
Core geometry of flat tubes, continuous fins and operating conditions of a compact heat exchanger–radiator [15].

Serial number	Description	Air	Coolant
1	Fluid inlet temperature	20–55 °C (assume $T_a = 37.5$ °C)	70–95 °C (assume $T_a = 82.5$ °C)
2	Core width	0.6 m	
3	Core height	0.5 m	
4	Core depth	0.4 m	
5	Tube size	1.872 cm × 0.245 cm	

Table 2
Surface characteristics of a compact heat exchanger–radiator [17].

Serial number	Description	Air side	Coolant side
1	Tube arrangement	Staggered	
2	Fin type	Ruffled	
3	Fin pitch	4.46 fins/cm	
4	Fin thickness	0.01 cm	
5	Hydraulic diameter, D_h	0.351 cm	0.373 cm
6	Free flow area frontal area, σ	0.78	0.129
7	Heat transfer area/total volume, α	886 m^2/m^3	138 m^2/m^3
8	Fin area/total area, β	0.845	

5.1. Air side calculation

Initially, air side calculations were performed to determine air heat capacity, air heat transfer coefficient, fin efficiency and total surface temperature effectiveness. These data were needed to calculate heat exchanger effectiveness, NTU number and overall heat transfer coefficient for the nanofluids' side calculation. Air properties are based on temperature at 300 K. In addition, relevant input data from previous section were substituted into the mathematical formulations. The mathematical formulations are shown below.

(a) Air heat capacity rate, C_a can be expressed as Eq. (1):

$$C_a = W_a C_{p,a} \tag{1}$$

where,

$C_{p,a} = 1007$ J/kg K at 300 K.

$$W_a = G A_{fr} \sigma_a \tag{2}$$

(b) Heat transfer coefficient, h_a can be expressed as Eq. (3):

$$h_a = \frac{j_a G_a C_{p,a}}{Pr_a^{2/3}} \tag{3}$$

where,

$$j_a = \frac{0.174}{Re_a^{0.383}} \tag{4}$$

$$G = \frac{W}{A_{fr} \sigma_a} \tag{5}$$

$$Re_a = \frac{G_a D_{h,a}}{\mu_a} \tag{6}$$

where, $\mu_a = 0.00001846$ Ns/m^2 at 300 K.

Table 3
Thermal physical properties of ethylene glycol [18].

Thermal physical property	Ethylene glycol (at 82.5 °C)
Density (kg/m^3)	1076
Specific heat (J/kgK)	2664
Viscosity (Ns/m^2)	0.003036
Conductivity (W/m K)	0.261

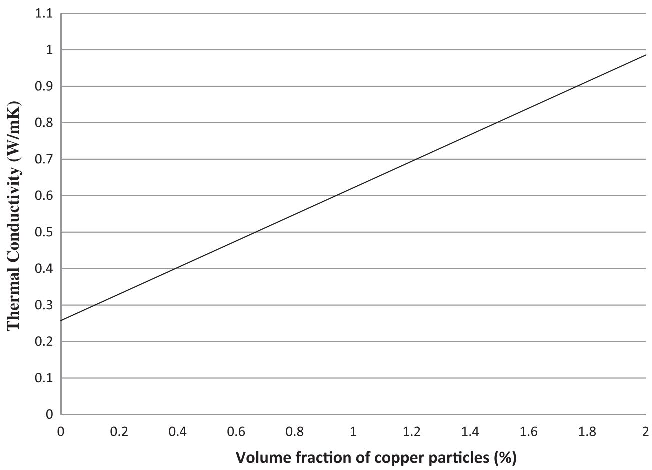

Fig. 1. Thermal conductivity of ethylene glycol based copper nanofluids.

(c) Fin efficiency of plate fin, η_f can be expressed as Eq. (7):

$$\eta = \frac{\tanh mL}{mL} \tag{7}$$

where,

$$m = \sqrt{\frac{2h_a}{kt}}, \tag{8}$$

$k = 3994$ W/m K is the thermal conductivity of copper fin.

(d) Total surface temperature effectiveness, can be expressed as Eq. (9):

$$\eta_o = 1.0 - \left(1.0 - \eta_f\right) \times \frac{A_f}{A} \tag{9}$$

5.2. Nanofluid side calculation

The parameters needed for nanofluid side calculation are nanofluid heat transfer coefficient, nanofluid heat capacity rate, heat exchanger effectiveness for cross-flow unmixed fluid, heat transfer coefficient based on air side, pressure drop, pumping power and total heat transfer rate. Similar to previous section, the relevant input data are substituted into the mathematical formulations as shown below.

(a) Heat transfer coefficient can be expressed as Eq. (10):

$$h_{nf} = \frac{Nu_{nf} k_{nf}}{D_{h,nf}} \tag{10}$$

k_{nf} is obtained from Eastman et al. [3] where,

$$Nu_{nf} = 0.023 Re_{nf}^{0.8} Pr_{nf}^{0.3} \tag{11}$$

$$Re_{nf} = \frac{G_{nf} D_{h,nf}}{\mu_{nf}} \tag{12}$$

μ_{nf}, $C_{p,nf}$ and ρ_{nf} were calculated based on correlations obtained from Tsai and Chein [19]. Authors in their study also used these correlations to analyze the performance of nanofluid-cooled microchannel heat sink. Similar correlations for $C_{p,nf}$ and ρ_{nf} were used by other researchers [20].

$$\mu_{nf} = \mu_f \frac{1}{(1-\varphi)^{2.5}} \tag{13}$$

$$G_{nf} = \frac{W_{nf}}{A_{fr}\sigma_{nf}} \tag{14}$$

$$v_{nf} = \frac{W_{nf}}{\rho_{nf}} \tag{15}$$

$$Pr_{nf} = \frac{\mu_{nf} C_{p,nf}}{k_{nf}} \tag{16}$$

$$C_{p,nf} = \frac{\left(1-\varphi\right)\rho_f C_{p,f} + \varphi\rho_p C_{p,p}}{\rho_{nf}} \tag{17}$$

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_p \tag{18}$$

(b) Heat capacity rate, C_{nf} can be expressed as Eq. (19):

$$C_{nf} = W_{nf} C_{p,nf} \tag{19}$$

(c) Heat exchanger effectiveness for cross-flow unmixed fluid, ε can be expressed as Eq. (20):

$$\varepsilon = 1 - \exp\left[\frac{1}{C^*}\right](NTU)^{0.22}\left[\exp\left[-C^*(NTU)^{0.78}\right] - 1\right] \tag{20}$$

where,

$$C^* = \frac{C_{minimun}}{C_{maximum}} \tag{21}$$

$$NTU = \frac{U_a A_{fr,a}}{C_a} \tag{22}$$

(d) Overall heat transfer coefficient, based on air side can be expressed as Eq. (23), where wall resistance and fouling factors are neglected.

$$\frac{1}{U_a} = \frac{1}{\eta_o h_a} + \frac{1}{\left(\frac{\alpha_{nf}}{\alpha_a}\right) h_{nf}} \tag{23}$$

(e) Pressure drop can be expressed as Eq. (24):

$$DP_{nf} = \frac{G_{nf}^2 \times f_{nf} \times H}{2 \times \rho_{nf} \times \left(\frac{D_{h,nf}}{4}\right)} \tag{24}$$

where,

$$f_{nf} = 0.079 \times \left(Re_{nf}\right)^{-0.25} \tag{25}$$

(f) Pumping power can be expressed as Eq. (26):

$$P = V_{nf} \times DP \tag{26}$$

(g) Total heat transfer rate can be expressed as Eq. (27):

$$Q = \varepsilon C_{min}\left(T_{nf,in} - T_{a,in}\right) \tag{27}$$

5.3. Test conditions

The analysis was performed based on a radiator specifications and conditions of fluids shown in Tables 1 and 2. However, nanoparticles volume fraction, air and coolant Reynolds number were varied in order to determine the thermal performance of the radiator using nanofluids. The test conditions for each analysis are explained below.

(a) Influence of volume fraction of copper nanoparticles on the thermal performance of an automotive car radiator: In this analysis, air and coolant Reynolds number were kept fixed at 4000 and 5000, respectively. However, concentrations of copper nanoparticles were increased from 0% to 0.2%. Coolant mass and volume flow rate, coolant Prandtl and Nusselt number, overall heat transfer coefficient based on air side and total heat transfer rate of the radiator were then determined.
(b) Influence of air Reynolds number on the thermal performance of a radiator: Air Reynolds number was varied from 4000 to 6000 while coolant Reynolds number was kept fixed at 5000. The analysis also included a comparison of the thermal performance of a radiator with nanofluids at different volume fractions. This part focused on the air heat transfer coefficient, overall heat transfer coefficient based on air side and total heat transfer rate of an automotive radiator.
(c) Influence of coolant Reynolds number on the thermal performance of a radiator: Coolant Reynolds number was varied from 5000 to 7000 while air Reynolds number was kept fixed at 4000. The analysis also included a comparison of thermal performance of the radiator with nanofluids at different volume fractions. This analysis focused on the overall heat transfer coefficient based on air side and total heat transfer rate of an automotive radiator.
(d) Comparison of coolant pressure drop and pumping power: In this section, coolant flow rate was kept fixed at 0.2 m^3/s and air Reynolds number at 4000 but the volume fraction of copper nanoparticles was varied. It focused on the effects of volume fraction of copper nanoparticles on the coolant pressure drop and pumping power.

6. Results and discussions

6.1. Influence of volume fraction of copper particles to thermal performance of an automotive car radiator

In this section, analysis of thermal performance of an automotive car radiator at constant Reynolds number of air (4000) and coolant (5000) have been carried out. With the increase of volume fraction of copper nanoparticles, the dynamic viscosity of nanofluids has been increased. Dynamic viscosity of nanofluids in this study was calculated using the correlation developed by Tsai and Chein [15] as shown in Eq. (13). This parameter influences mass flow rate of nanofluids in an automotive car radiator. From Eq. (12), it is found that the mass velocity of nanofluids increased with the increase in volume fraction of copper nanoparticles due to higher dynamic viscosity of the nanofluids. It can also be explained from Eq. (28).

$$G_{nf} = \frac{Re_{nf} \times \mu_{nf}}{D_{h,nf}} \tag{28}$$

In this equation μ_{nf} was varied and other parameters were kept constant. This value is then substituted into Eq. (14) to calculate the coolant mass flow rate, which can be manipulated as shown in Eq. (29).

$$W_{nf} = G_{nf} \times A_{fr} \times \sigma_{nf} \tag{29}$$

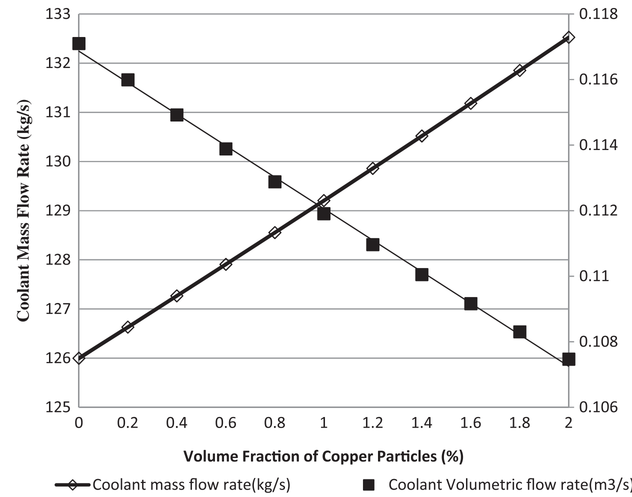

Fig. 2. Effect of copper volume fraction to coolant mass and volumetric flow rate at constant air and coolant Reynolds number.

However, volumetric flow rate of nanofluids is decreased with increase of volume fraction of copper nanoparticles as calculated using Eq. (15) and presented in Fig. 2. In this study, it is found that Prandtl number of nanofluids based coolant decreases exponentially with volume fraction of copper nanoparticles, mainly due to higher thermal conductivity of nanofluids. Nanofluids exhibit higher thermal conductivity due to increase in Brownian motion, formation of nanolayer etc. Prandtl number of nanofluids is calculated using Eq. (16) and the k_{nf} is an influential parameter for Prandtl number. It may be stated that higher value of this property will lead to lower value of Prandtl number. These relationships are shown in Fig. 3. Fig. 3 also showed lower coolant Nusselt number with the addition of copper particles, based on Eq. (11). In this equation, the coolant Reynolds number is kept constant while the coolant Prandtl number is varied. This study also found that ethylene glycol based copper nanofluids demonstrated higher overall heat transfer coefficient for air side as calculated using Eq. (23). The relationship is shown in Fig. 4 where overall heat transfer coefficient for air side is increased with copper nanoparticles. For instance, an overall heat transfer coefficient, 164 W/m^2 K can be achieved for 2% Cu + EG nanofluid compared to 142 W/m^2 K for the basefluid. This indicates lower air side area is required to achieve 142 W/m^2 K overall heat transfer

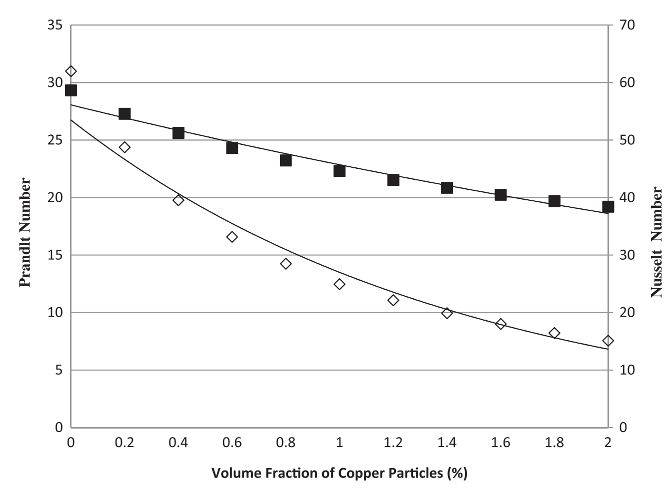

Fig. 3. Effect of copper volume fraction to coolant Prandtl and Nusselt number at constant air and coolant Reynolds number.

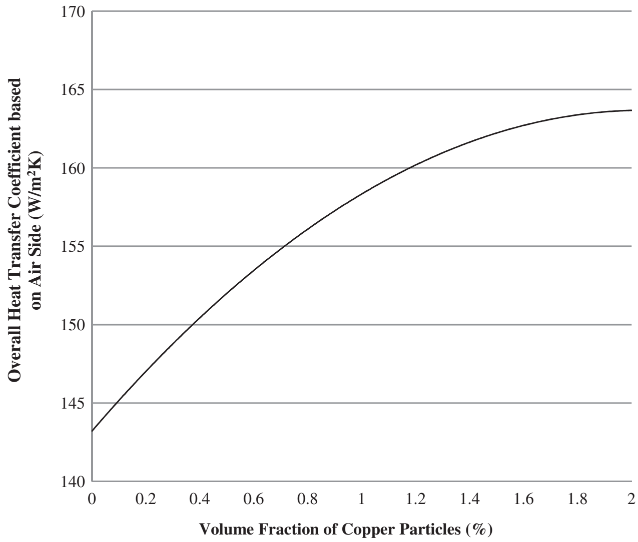

Fig. 4. Effect of copper volume fraction to overall heat transfer coefficient based on air side at constant air and coolant Reynolds number.

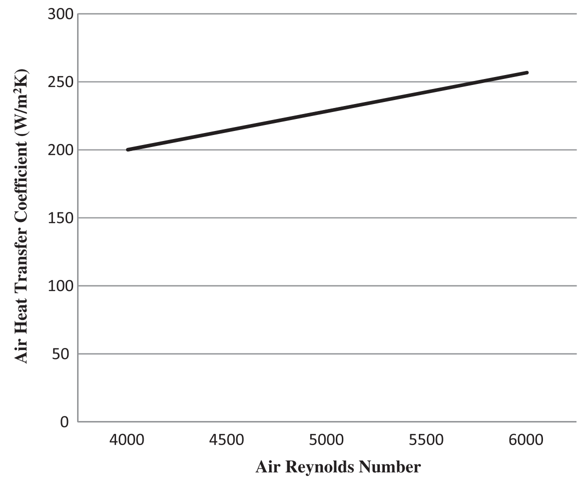

Fig. 6. Effect of air Reynolds number to air heat transfer coefficient.

coefficient if 2% Cu + EG nanofluid is used. Hence, estimated reduction of air side area up to 15.31% at 2% Cu + EG particles was achieved. This study also found that heat transfer rate is increased exponentially as the volume fraction of copper particles are increased as shown in Fig. 5. This improvement is calculated using Eq. (27). It can be deduced that effectiveness of the radiator is increased with the application of nanofluids. However the percentage of effectiveness does not increase substantially, although the improvement of overall heat transfer coefficient is significant.

6.2. *Influence of air Reynolds number on thermal performance of a radiator*

The effect of air Reynolds number on the thermal performance of a radiator is discussed in this section. A coolant's volumetric and mass flow rates, Nusselt and Prandtl numbers did not experience any change since the coolant Reynolds number was kept fixed at 5000. Only air Reynolds number and fraction of copper nanoparticles were varied in this section. With the increase of air Reynolds numbers, the air heat transfer coefficient was increased as shown in Fig. 6. This will influence the thermal performance of the radiator system. Substituting higher value of air heat transfer coefficient into Eq. (23), overall heat transfer coefficient based on air side of the radiator is obtained and shown in Fig. 7. From the study, air heat transfer coefficient is found proportional with air Reynolds number. It can be explained using Eqs. (3) and (6). Eq. (6) can be manipulated as shown the Eq. (30):

$$G_a = \frac{\mathrm{Re}_a \mu_a}{D_{h,a}} \tag{30}$$

In Eq. (30), the hydraulic diameter and air viscosity were kept constant. Therefore, air mass velocity is increased when air Reynolds number increases. The higher value of air mass velocity is then substituted into Eq. (3) to determine the air heat transfer coefficient. In Eq. (3) the air heat capacity and Prandtl number were kept constant. Although the Colburn factor is decreased with air Reynolds number, the increment of air mass velocity is much higher and subsequently air heat transfer coefficient is increased.

Nanofluids with higher copper volume fraction generates higher overall heat transfer coefficient than that of a basefluid. Same scenario happened for heat transfer rate where it is proportional to air Reynolds number as shown in Fig. 8. About 3.8% of heat transfer improvement can be achieved with addition of 2% copper particles at 6000 and 5000 Reynolds number for air and coolant respectively. Based on the overall heat transfer coefficient and heat transfer rate improvement, percentage reduction of air frontal area can be estimated, at these Reynolds numbers. Percentage improvement of heat transfer rate is increased with air Reynolds number. Although

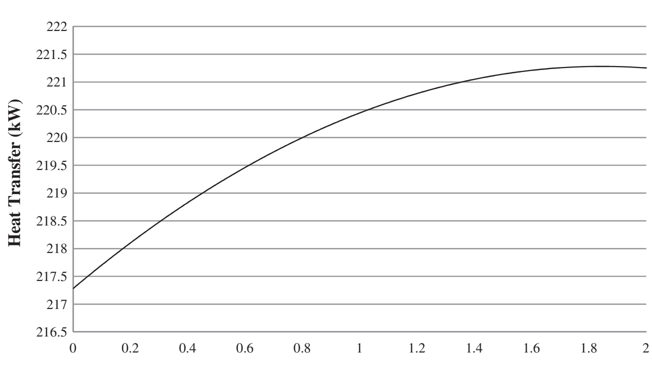

Fig. 5. Effect of copper volume fraction to heat transfer rate at constant air and coolant Reynolds number.

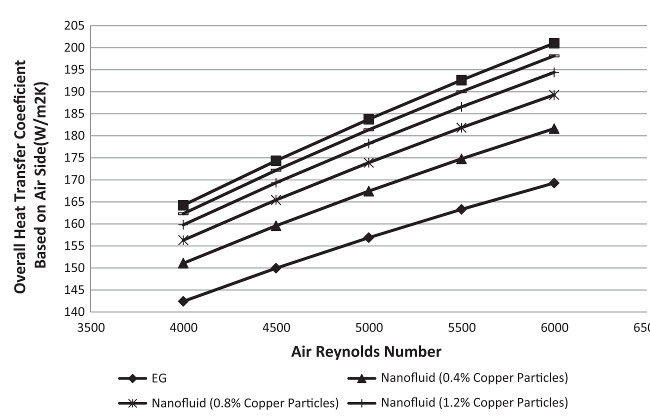

Fig. 7. Effect of air Reynolds number and copper volume fraction to overall heat transfer coefficient based on air side.

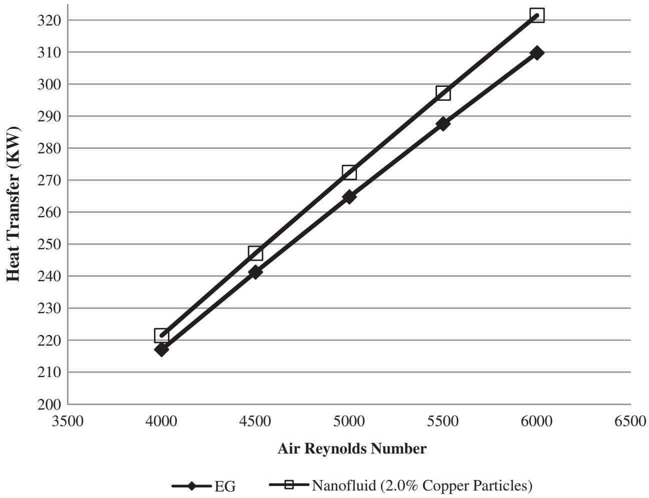

Fig. 8. Effect of air Reynolds number and copper volume fraction to heat transfer rate of radiator.

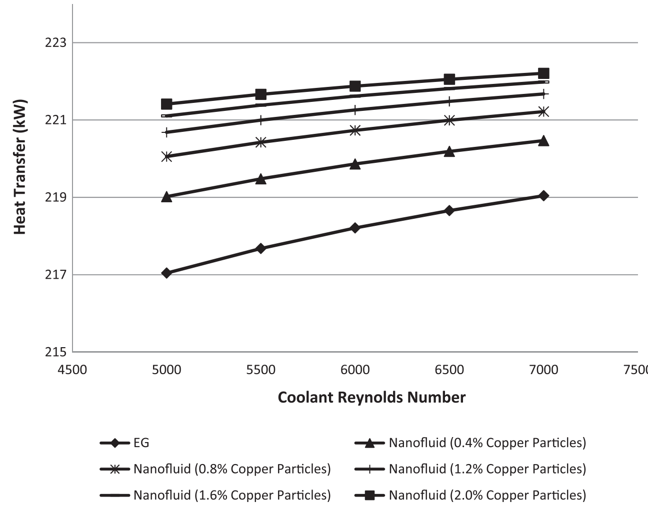

Fig. 10. Effect of coolant Reynolds number to heat transfer rate of radiator.

results indicated that higher air Reynolds number leads to better heat dissipation process, design of the radiator must ensure the engine operation at optimum temperature. Driving conditions or its speed and engine load must be considered. For instance, car's engine needs to operate at higher load when driving up hill and at the same time air Reynolds number is low due to lower air velocity. Hence, there is possibility for engine to get overheated at this condition. However when driving downhill, an engine only requires operating at lower load and at the same time high air Reynolds number is observed. Eventually engine might be overcooled. Therefore, these aspects must take into consideration when designing automotive radiator.

6.3. *Influence of coolant Reynolds number on thermal performance of a radiator*

This section presents the effect of coolant Reynolds number on the thermal performance of a radiator at a fixed air Reynolds number (4000). Coolant Reynolds number plays vital role in determining the radiator's thermal performance. Engine might be overcooled or overheated if coolant Reynolds number is not properly controlled. The main function of a radiator is to ensure that engine is operating at optimum temperature by not only controlling the air Reynolds number but also the coolant Reynolds number. Although the coolant pump is usually driven by an engine, thermostat is also playing an important role to control the coolant Reynolds number. From this study, it indicates that as the coolant Reynolds numbers are increased, the mass and volumetric flow rates are also increased. Overall heat transfer coefficient based on air side is increased with coolant Reynolds number as shown in Fig. 9. The magnitude of this property for nanofluids is higher than that of a basefluid. Therefore, heat transfer area reduction for the same value of overall heat transfer coefficient can be achieved by using nanofluids. Heat transfer enhancement was also observed with coolant Reynolds number. For instance, with the addition of 2% copper particles, 1.4% improvement of heat transfer rate has been achieved at 4000 and 7000 Reynolds number for air and coolant respectively. It is also observed that the percentage of improvement is decreased with decrease of coolant Reynolds number. Fig. 10 shows heat transfer rate of a radiator using nanofluid is higher than that of a radiator using ethylene glycol.

6.4. *Comparison of coolant pressure drop*

In this section analyses of the coolant pressure drop and pumping power of a radiator at a fixed coolant flow rate (0.2 m^3/s)

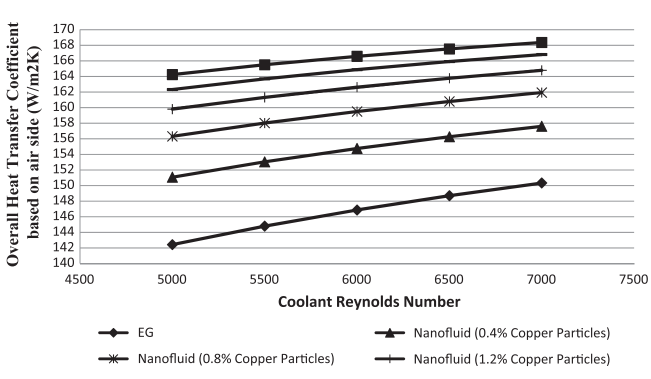

Fig. 9. Effect of coolant Reynolds number to overall heat transfer coefficient based on air side.

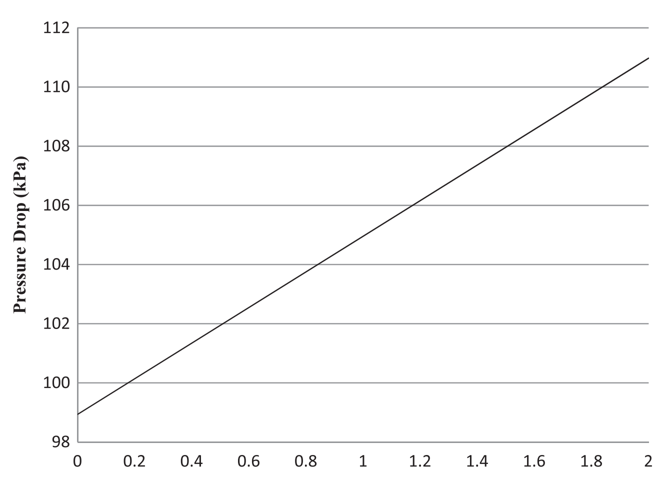

Fig. 11. Influence of copper volume fraction to coolant pressure drop at fixed coolant volumetric flow rate.

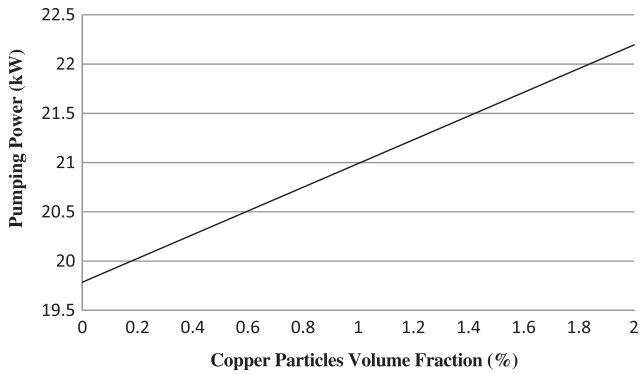

Fig. 12. Influence of copper volume fraction to pumping power at fixed coolant volumetric flow rate.

and an air Reynolds number of (4000) with varying copper nanoparticles volume fraction is incorporated. Initially, the coolant volumetric flow rate was converted to mass flow rate by using Eq. (15). This equation is manipulated as shown in Eq. (31) to calculate the coolant mass flow rate.

$$W_{nf} = \upsilon_{nf} \times \rho_{nf} \tag{31}$$

Coolant mass flow rate is found to be increased with the addition of copper nanoparticles due to its higher density. Then, coolant mass velocity is calculated using Eq. (14). This value is substituted into Eq. (12) to get the coolant Reynolds number. The pressure drop has been calculated by using fanning friction factor based on Eq. (25), which then substituted into Eq. (24).

It was observed that the coolant pressure drop increased with the addition of copper nanoparticles. The result reveals that a pressure drop of 110.97 kPa was obtained by adding 2% copper particles compared to a pressure drop of 98.93 kPa for a basefluid. Due to this extra pressure drop, a higher coolant pumping power is needed. The pumping power is calculated using Eq. (26). Calculated results indicate that about 12.13% increase in pumping power is observed at 2% addition of copper nanofluids compared to a basefluid. These trends are shown in Figs. 11 and 12. Increase in density increases pressure drop of flowing liquids. Adding particles in a base liquid increases density of the fluid and augments pressure drop at a low percentage as observed in the present study. Similar results were reported by Ko et al. [13] and Duangthongsuk and Wongwises [14].

7. Conclusion

Following conclusions can be drawn from this study:

(a) Heat transfer rate is increased with increase in volume concentration of nanoparticles (ranging from 0% to 2%). About 3.8% heat transfer enhancement was achieved with addition of 2% copper particles at 6000 and 5000 Reynolds number for air and coolant respectively.
(b) Thermal performance of a radiator using nanofluid or ethylene glycol coolant is increased with air and coolant Reynolds number.About 42.7% and 45.2% heat transfer enhancement were observed for pure ethylene glycol and ethylene glycol with 2% of copper nanoparticles respectively when air Reynolds number was increased from 4000 to 6000. Only 0.9% and 0.4% heat transfer enhancement were observed for pure ethylene glycol and ethylene glycol with 2% copper nanoparticles respectively when coolant Reynolds number was increased from 5000 to 7000.
(c) Estimated 18.7% reduction of air frontal area is achieved by adding 2% copper nanoparticles at Reynolds number of 6000 and 5000 for air and coolant respectively.
(d) Additional 12.13% pumping power is needed for a radiator using nanofluid of 2% copper particles at 0.2 m^3/s coolant volumetric flow rate compared to that of the same radiator using only pure ethylene glycol coolant.

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