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ANALYTICAL INVESTIGATION OF ENHANCEMENT IN HEAT TRANSFER OF TUBE FIN HEAT EXCHANGER USING DIFFERENT NANOFLUIDS

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ABSTRACT

Continuous technological development in automotive industries has increased the demand for high efficiency engines. With the advancement of nanotechnology, the new generation of heat transfer fluids called, "nanofluids" have been developed. In the present study SiO_2, TiO_2, Al_2O_3 and Cu nanofluids are used in tube fin heat exchanger (automotive cooling system) and compared with base fluid Ethylene Glycol. It was observed that the thermal conductivity, heat transfer rate, overall heat transfer coefficient (based on air side) and heat transfer coefficient of fluid increases with the addition of nanoparticles. It is observed that the highest overall heat transfer coefficient is achieved by Cu nanofluids, which is 170 W/m^2K in 3% nanoparticle concentration at 5000 and 4000 Reynolds number for coolant and air respectively compared to 147.2 W/m^2K for the basefluid. In addition lower heat transfer air side area is required to achieve 147.2 W/m^2K overall heat transfer coefficient.

Keywords: Heat Transfer Enhancement, Nanofluids, Tube Fin Heat Exchanger (Automotive Cooling System)

I. INTRODUCTION

Traditional heat transfer fluids which include water, oils and glycol are normally used in heat exchangers to manage the thermal loads. The heat transfer performance of Pure single phase fluids are relatively low compared to two phase fluids. More over the heat transfer coefficient of fluids largely depends on their thermo physical properties. Hence there is a need to develop energy efficient fluids with substantantially high thermal conductivities and more heat transport capabilities to manage the thermal loads in heat exchange systems. Modern technology facilitates to produce process and characterize materials in nano size. Innovative heat transfer fluids-suspended by nanometer-sized solid particles are called 'nanofluids' which was coined by Choi in 1995 [1]. Enhancement of convective heat transfer and thermal conductivity of liquids was earlier made possible by mixing micron sized particles with a base fluid (Maxwell paper) [2]. Tube fin heat exchanger is a common type of heat exchanger having several benefits such as compact size, high heat transfer performance and large capacity made them more popular in automotive cooling system.

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 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 (Al2O3) 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 (ZnOeEG) based nanofluids were 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. Lee et al. [7] measured the thermal conductivity of base fluids (water and ethylene glycol) containing CuO and Al_2O_3 nanoparticles. Their experiments showed that the thermal conductivity of ethylene glycol based nanofluid was always higher than those of water based nanofluids. Torii et al. [8] conducted experimental study on thermal–fluid flow characteristics of nanofluids in heated circular tube and showed that the pressure loss increases slightly in comparison with that of pure water. Sunder et al. [9] evaluated experimentally the convective heat transfer coefficient of Fe3O4 nanofluid for flow in a circular tube at the range of 3000 < Re < 22,000 and the volume concentration range of 0 < C < 0.6%. Nanofluid heat transfer was higher compared with water and increased with volume concentration. The heat transfer coefficient was enhanced by 30.96% at 0.6% volume concentration compared with the flow of water at similar operating conditions.

The present study focuses on nanofluid application in tube fin heat exchanger. Since increasing the heat transfer rate and minimizing the pressure drop are considered as the main objective for the improving heat exchangers performance. Variation of heat transfer rate, overall heat transfer coefficient, pressure drop and thermal conductivity of the fluid is compared four different nanofluids.

II. MATHEMATICAL MODEL

Mathematical correlations used in section 2.1 and 2.2 are taken from reference [10, 11, 12].

2.1 Air side Calculation

1. Air heat capacity rate:

$$C_a = W_a C_{p,a}$$

1

Where $C_{p,a} = 1006.3$ J/kg-K at 300 K

2. Heat transfer coefficient:

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

2

Where

$$G_a = \frac{W_a}{A_{fr,a}\sigma_a}$$

3. Frontal area:

$$A_{fr,a} = L * H$$

3

4. Colburn factor for straight fin:

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

4

5. Hydraulic diameter:

$$D_{h,a} = \frac{4\,\sigma_a}{\alpha_a}$$

5

6. Reynolds number for straight fin:

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

6

Where,

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

7. Fin efficiency of plate fin:

$$\eta_{fin} = \frac{\tanh ml}{ml}$$

7

Where

$$m = \sqrt{\frac{2h_a}{K_{fin} t_{fin}}}$$

8. Total surface temperature effectiveness:

$$\eta_0 = 1.0 - \frac{A_{fin}}{A_{t,a}}\left(1 - \eta_{fin}\right)$$

8

2.2 Nanofluid side calculation

1. Heat transfer coefficient:

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

9

k_{nf} is obtained from correlation given by Mohammed et al. [11]

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

2. Nusselt number:

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

10

3. Reynolds number:

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

11

4. Dynamic Viscosity:

$$\mu_{nf} = \frac{1}{(1-\phi)^{2.5}} \mu_{bf}$$

12

5. Prandtl number:

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

13

6. Heat capacity rate:

$$C_{nf} = W_{nf} C_{p,nf}$$

14

7. Heat exchanger effectiveness for cross-flow unmixed fluid:

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

15

Where

$$CR = \frac{C_{minimum}}{C_{maximum}}$$

$$NTU = \frac{U_a A_{t,a}}{C_a}$$

8. Overall heat transfer coefficient, based on air side, where wall resistance and fouling factors are neglected:

$$\frac{1}{U_a} = \frac{1}{\eta_0 h_a} + \frac{1}{\left(\frac{\alpha_{nf}}{\alpha_a}\right) h_{nf}}$$

16

9. Pressure drop:

$$DP_{nf} = \frac{G_{nf}^2 \times f_{nf} \times L_1}{2 \times \rho_{nf} \times \left(\frac{D_{h,nf}}{4}\right)}$$

17

Where

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

10. Pumping power:

$$P = V_{nf}\, DP$$

18

Where

$$V_{nf} = \frac{W_{nf}}{\rho_{nf}}$$

11. Total heat transfer rate:

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

19

2.3 Input data and operating characteristic

The configuration of the tube fin heat exchanger used in this study are obtain from Vasu et al. [13] and Kays and London [14] as shown in table 1 and table 2. Nanoparticles and ethylene glycol property are shown in tale 3 and table 4.

TABLE 1: Core geometry and operating condition of tube fin heat exchanger [13].

Sr. No.	Description	Coolant	Air
1	Fluid inlet temperature	343 K	300 K
2	Core width	0.6 m	-
3	Core height	0.5 m	-
4	Core depth	0.4 m	-
5	Tube Size	1.872 cm x 0.245 cm	-

TABLE 2: Surface characteristics of a tube fin heat exchanger [15].

Sr. No.	Description	Coolant side	Air side
1	Tube arrangement	-	Staggered
2	Fin type	-	Ruffled
3	Fin pitch	-	4.46 fins/cm
4	Fin thickness	-	0.01016
5	Hydraulic diameter, D_h	0.003739 m	0.00351 m
6	Free flow area/Frontal area, σ	0.129	0.78
7	Heat transfer area/total Volume, α	138 m^2/m^3	886 m^2/m^3
8	Fin area/total area, β	-	0.845

TABLE 3: Thermal physical property of Ethylene Glycol [16].

Conductivity (W/m^2-K)	0.261
Viscosity (Ns/m^2)	0.003036
Specific Heat (J/kg-K)	2664
Density (kg/m^3)	1076

TABLE 4: Thermal physical property of nanoparticles [17,18].

Property	SiO_2	TiO_2	Al_2O_3	Cu
Density (kg/m^3)	2220	4157	3970	8933
Specific heat (J/kg-K)	745	710	765	385
Thermal conductivity (W/m-K)	1.38	8.4	36	400

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III. RESULT AND DISCUSSION

3.1 Effect of volume concentration of nanoparticles on thermal conductivity of ethylene glycol

The thermal conductivity of heat transfer fluid has a considerable effect on improving the heat transfer rate, and several studies have been reported on the thermal conductivity of nanofluids, particularly water and ethylene glycol based nanofluids. The theoretical investigation on thermal conductivity of the nanofluids showed a considerable enhancement compared to the base fluid. The thermal conductivity enhancement (with respect to that of base fluid) has been found to increase as a function of volume concentration. Maximum enhancement is obtained for EG/Cu nanofluid.

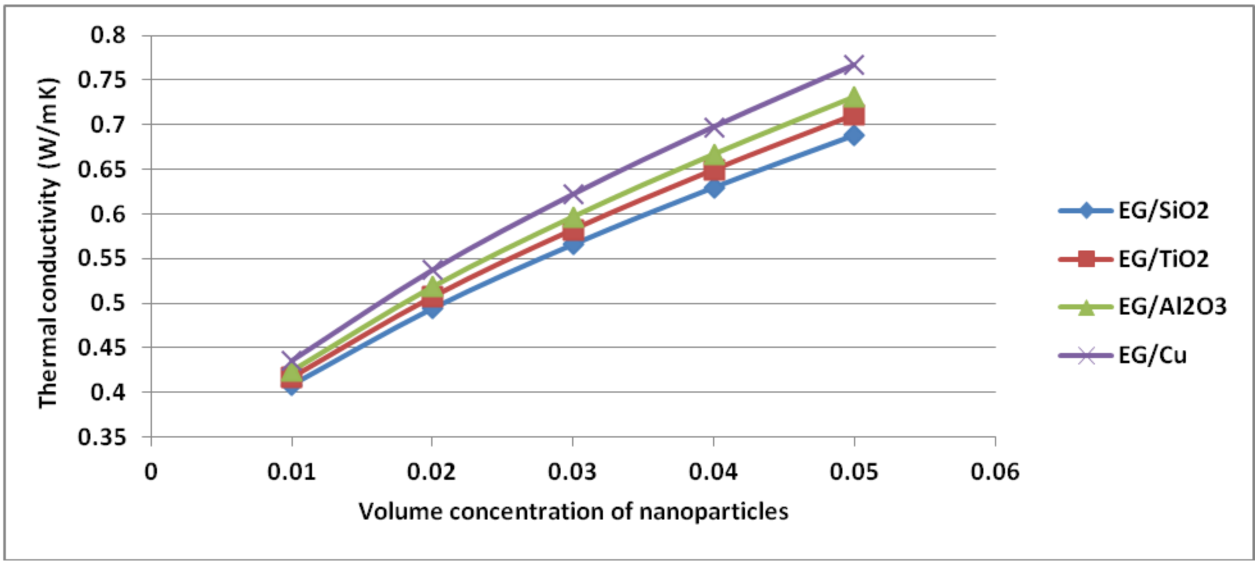

Figure 1. Effect of volume concentration of nanoparticles on thermal conductivity.

3.2 Effect of volume concentration of nanoparticles on heat transfer and overall heat transfer coefficient based on air side

This study shows that ethylene glycol based copper nanofluids demonstrated higher overall heat transfer coefficient for air side (calculated using Equation 16). Relation is shown in Figure 2 where overall heat transfer coefficient based on air side is increased with addition of nanoparticles.

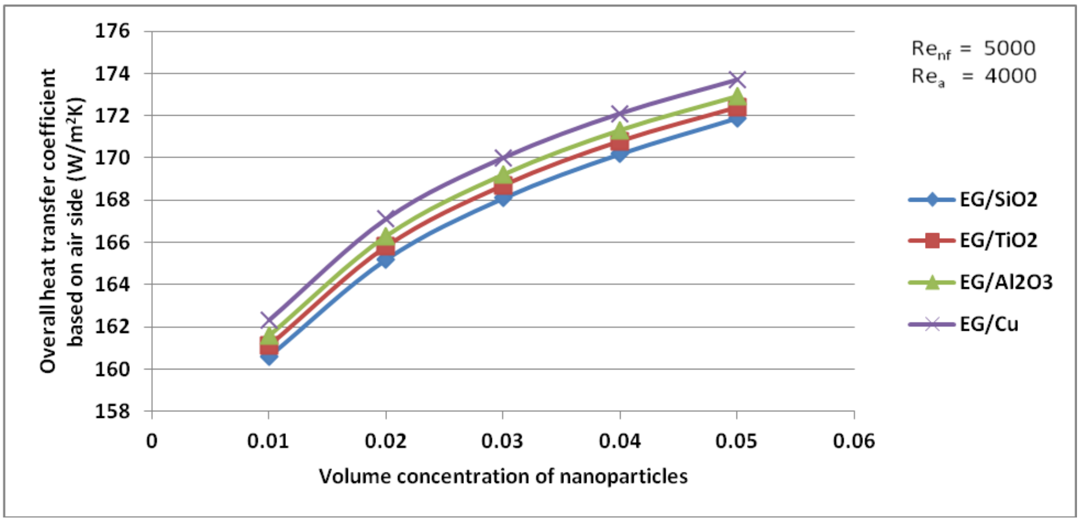

Figure 2. Effect of volume concentration of nanoparticles on overall heat transfer coefficient.

At 5%, 173.7 W/m^2-K overall heat transfer is obtained for EG/Cu nanofluids compared to 147.2 W/m^2-K for basefluid at 5000 and 4000 Reynolds number for coolant and air respectively. This study also show that by addition of nanoparticles in base fluid ethylene glycol heat transfer rate is found to increase.

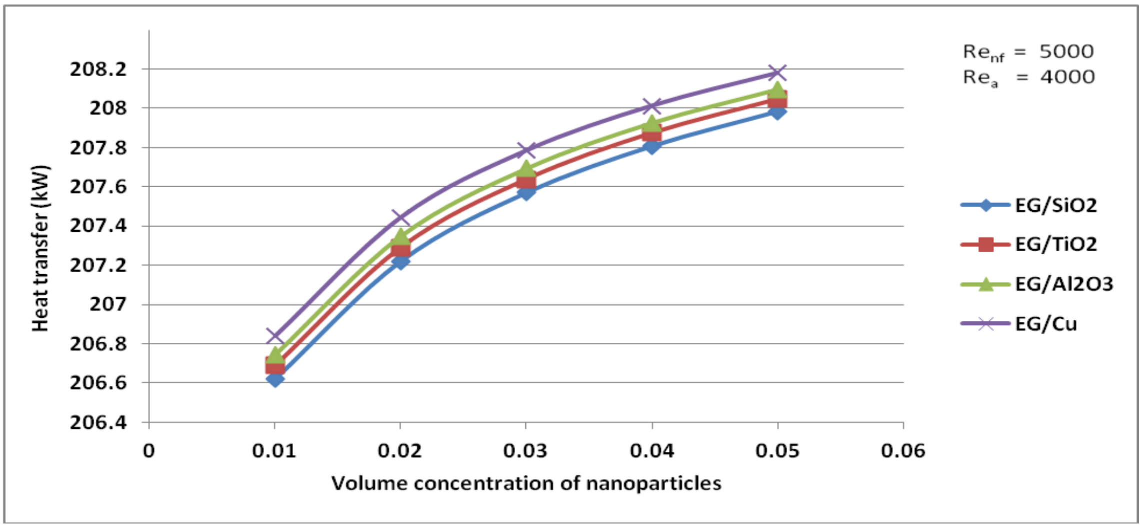

Figure 3. Effect of volume concentration of nanoparticles on heat transfer.

3.3 Effect of air Reynolds number and volume concentration of nanoparticles on heat transfer ratio and overall heat transfer coefficient ratio based on air side

In this section coolant Reynolds number (5000) and nanoparticles volume concentration (3%) are kept constant. Variation of overall heat transfer coefficient ratio based on air side with air Reynolds number is shown in Figure 4. At 3% nanoparticles volume concentration and 7313 air Reynolds number EG/Cu exhibit maximum enhancement in overall heat transfer ratio. As compare to base fluid (EG), EG/Cu nanofluid shows 21.29% enhancement in overall heat transfer coefficient based on air side which implies that by adding 3% nanoparticles for same performance we can reduce heat transfer area. At the same time total heat transfer ratio also increases

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with air Reynolds number as shown in Figure 5. Around 4.26% enhancement occur in total heat transfer at 3% nanoparticles volume concentration and 7313 air Reynolds number for EG/Cu nanofluid.

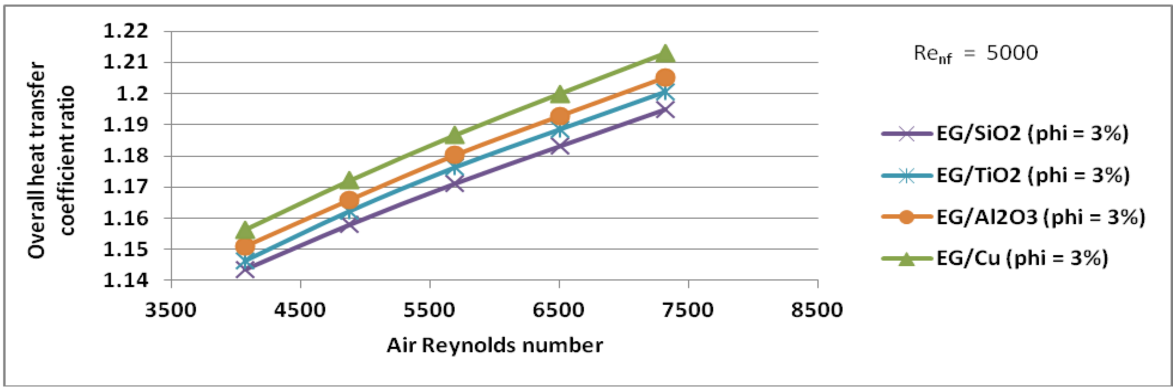

Figure 4. Variation of overall heat transfer coefficient ratio for various nanofluids with air Reynolds number at fixed nanoparticles volume concentration.

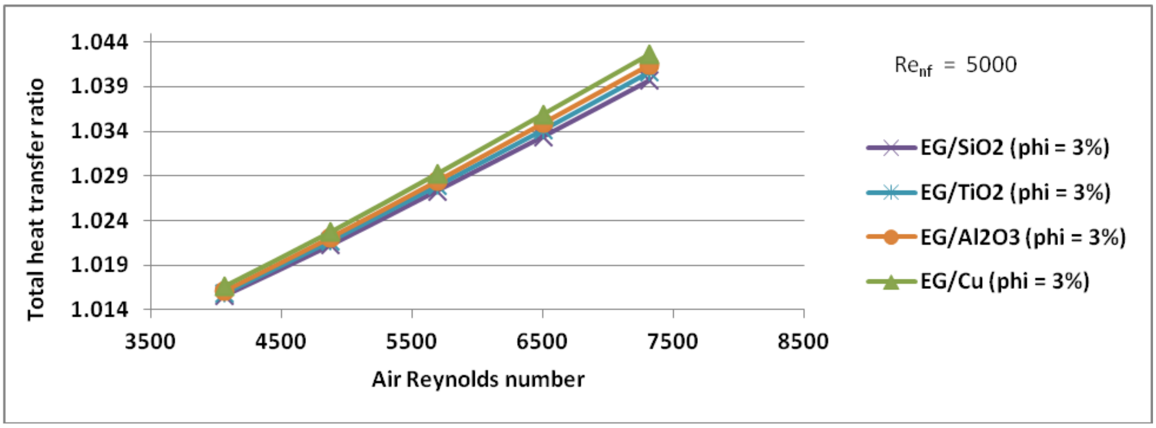

Figure 5. Variation of total heat transfer ratio for various nanofluids with air Reynolds number at fixed nanoparticles volume concentration.

3.4 Effect of volume concentration of nanoparticles on coolant heat transfer coefficient

For the given range of study, the maximum improvement of nanofluids heat transfer coefficient for EG/Cu is observed in the range of approximately 4094 W/m^2-K to 12026 W/m^2-K, whereas that for EG/SiO_2 is 4094 W/m^2-K to 10785 W/m^2-K, for EG/TiO_2 is 4094 W/m^2-K to 11158 W/m^2-K, for EG/Al_2O_3 is 4094 W/m^2-K to 11473 W/m^2-K. Variation is shown in Figure 6.

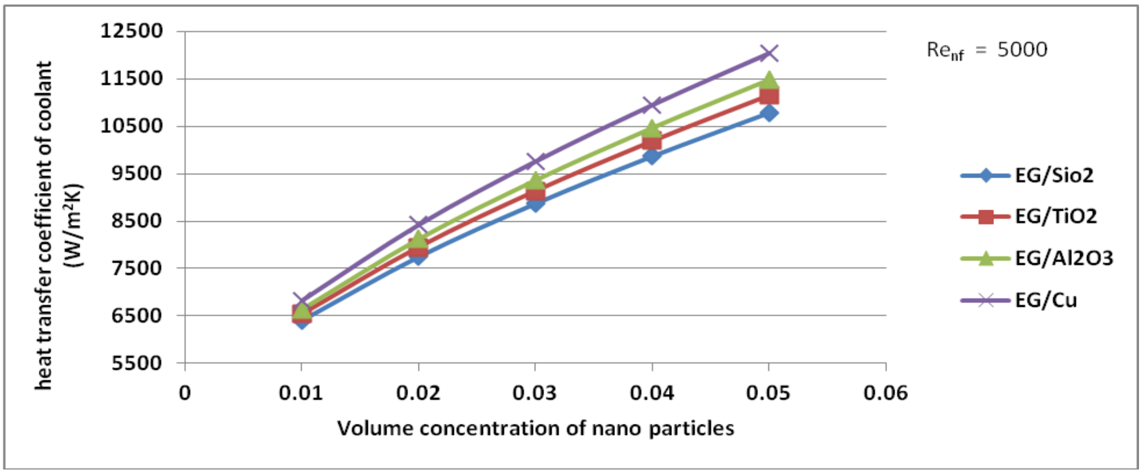

Figure 6. Variation of heat transfer coefficient for various nanofluids with nanoparticles volume concentration.

3.5 Effect of volume concentration of nanoparticles on pressure drop and pumping power

In this section coolant pressure drop and pumping power of tube fin heat exchanger at a fixed flow rate (0.1168 m^3/s) and air Reynolds number 4000 are determined. Figure 7 and Figure 8 show the variation of pressure drop and pumping power for various nanofluids with volume concentration of nanoparticles. Since by adding nanoparticles density is found to increase and coolant mass flow rate is directly proportional to density. It was observed that both pressure drop and pumping power is increases with addition of nanoparticles. Coolant mass flow rate is calculated using the following Equation.

$$W_{nf} = V_{nf} * \rho_{nf}$$

EG/Cu shows maximum increment in pressure drop and pumping power as 16.01% at 3% nanoparticles volume concentration.

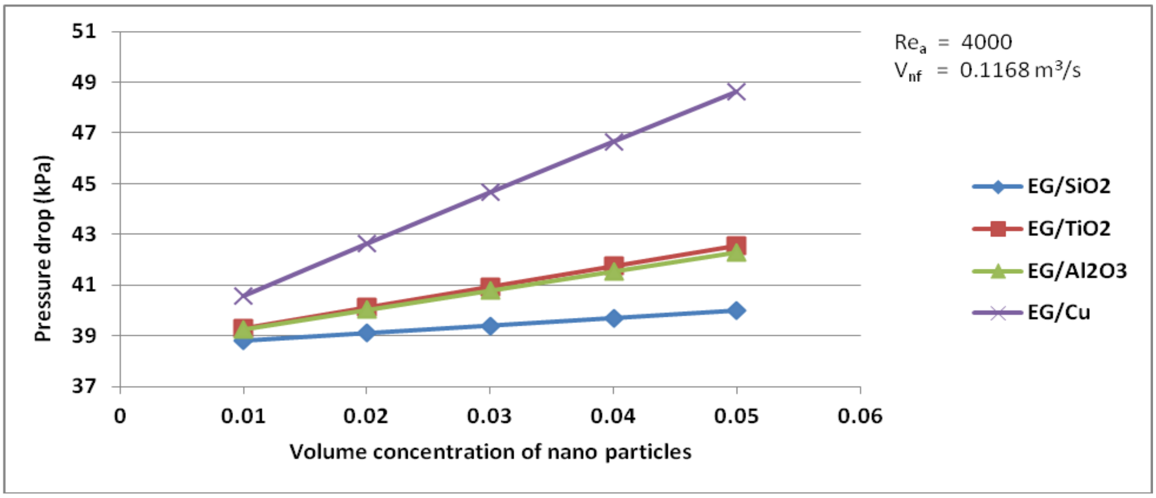

Figure 7. Variation of pressure drop for various nanofluids with nanoparticles volume concentration.

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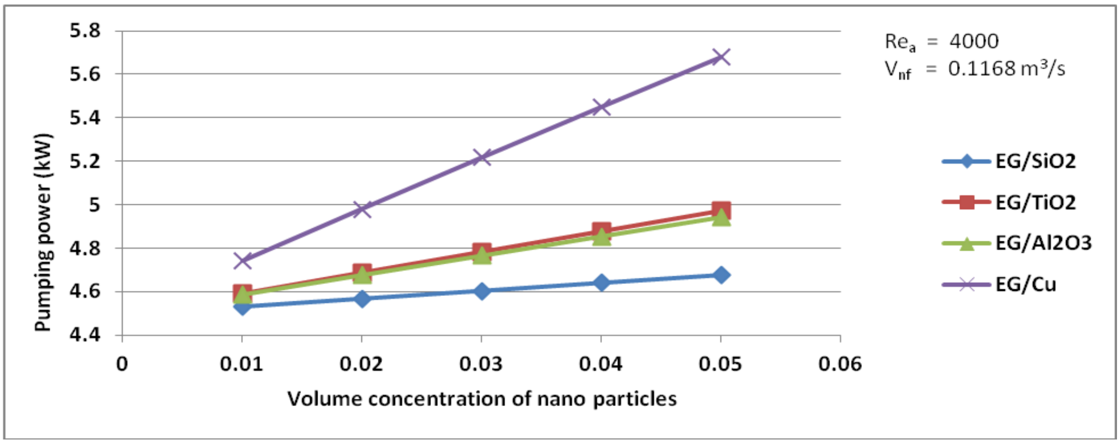

Figure 8. Variation of pumping power for various nanofluids with nanoparticles volume concentration

IV. CONCLUSION

1. Thermal conductivity of ethylene glycol is increased with increase in volume concentration of nanoparticles. Maximum enhancement is for EG/Cu nanofluid, whereas minimum for EG/SiO_2 due to lower thermal conductivity of SiO_2 nanoparticles as compare to other nanoparticles used in this study.
2. Overall heat transfer coefficient is found to increase with nanoparticles volume concentration. Maximum enhancement is observed for EG/Cu (15.49%) at 3% volume concentration of nanoparticles, whereas minimum for EG/SiO_2 is 14.20% at Reynolds number 5000 and 4000 for nanofluid and air respectively.
3. About 13.41% reduction in air frontal area is achieved by adding 3% nanoparticles volume concentration for EG/Cu nanofluid at Reynolds number 5000 and 4000 for nanofluids and air respectively.
4. Apart from enhancement in the performance of tube fin heat exchanger additional pumping power also needed and pressure drop is also increases.

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