Evaluation of Using Al_2O_3/EG and TiO_2/EG Nanofluids as Coolants in the Double-tube Heat Exchanger

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Abstract: In the present paper, Al_2O_3/EG and TiO_2/EG nanofluids with volume concentration of up to 10% have been used as coolants in a double-tube heat exchanger under laminar flow conditions. Thermal conductivity is an important parameter in the field of nanofluid heat transfer. Therefore, this paper presents two experimental models to predict the thermal conductivity of nanofluids. Based on these models, heat transfer relation between the hot solvent and nanofluid coolants has been investigated theoretically, as the first step. Because nanofluid coolants radiate the same amount of heat, they can be used in the heat exchanger to optimize the heat transfer area and the flow rate of the coolant. The results proved that as the probability of collision between nanoparticles and the heat exchanger wall increases, due to using higher concentration of coolants, the total heat transfer coefficient increases. Consequently, in this paper, heat transfer coefficient, total heat transfer coefficient, friction factor, pressure drop and pumping power for Al2O3/EG and TiO2/EG nanofluid coolants in the double-tube heat exchanger are calculated.

Keywords: Double-tube Heat Exchanger, Nanofluid, Heat Transfer Enhancement, Optimization

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1 INTRODUCTION

Improving the heat transfer characteristics of a coolant liquid had been the focus of research for decades. The conventional heat transfer fluids such as water and ethylene glycol are often limited due to their low thermal conductivities. To overcome the limited heat transfer capability of these fluids, the use of nanosized metals and metal oxides as an additive suspended into the base fluid is a technique for heat transfer enhancement. Fluids with incremented particles of nanometer dimensions are called nanofluids, this term was proposed by Choi [1] in 1995 at the Argonne National Laboratory, U.S.A. Compared with traditional solid-liquid suspensions containing millimeter or micrometer sized particles, nanofluids used as coolants in the heat exchangers have shown better heat transfer performance because of the small size of incremented solid particles [2-5]. This is due to the fact that nanofluids have a behavior similar to base liquid molecules.

In 2007 Nguyen et al. [6] used Al_2O_3/water nanofluids in an electronic cooling system and found a maximum of 40% enhancement in convective heat transfer coefficient at an added particle concentration of 6.8 vol%. In 2009 Pantzali et al. [7] applied 4% CuO nanofluid to a commercial herringbone-type PHE. This study indicates that fluid viscosity has an important role in the performance of a heat exchanger. Hwang et al. [8] through experimental investigation of flow and convective heat transfer characteristics of Al_2O_3/water nanofluid, with particles varying in the range of 0.01-0.3% in a circular tube of 1.812 mm inner diameter with the constant heat flux in fully developed laminar regime reported improvement in convective heat transfer coefficient in the thermally fully developed regime. Demir et al. [9] investigated numerically laminar and turbulent forced convection flows of Al_2O_3/water nanofluids as working fluids in a horizontal smooth tube with constant wall temperature and showed heat transfer enhancement due to nanoparticles in the base fluid.

Mohammed et al. [10] studied the effects of nanofluid types on the performance of a square shaped microchannel heat exchanger (MCHE) numerically. Their results revealed that Al_2O_3 and Ag nanofluids have the highest heat transfer coefficient and the lowest pressure drop among all the nanofluids tested, respectively. In 2012 Saeedinia et al. [11] applied CuO-base oil particles varying in the range of 0.2-2% inside a circular tube. Their results showed that the CuO nanoparticles incremented in base-oil increases the heat transfer coefficient even for a very low particle concentration of 0.2% volume concentration. Moreover, a maximum heat transfer coefficient enhancement of 12.7% is obtained for a 2% CuO nanofluid. In the present paper, Al2O3/EG and TiO2/EG nanofluids are applied as coolants to optimize a double-tube heat exchanger. Al_2O_3 and TiO_2 nanoparticles are two commercially available suspensions, procured from Sigma Aldrich®. Also, TiO_2 and Al_2O_3 nanoparticles are stabilized at various ranges of PH. It should be noted that metal oxides such as Al_2O_3 and TiO_2 nanoparticles are chemically more stable than their metallic counterparts. Therefore, the reasons for choosing TiO_2 and Al_2O_3 nanoparticles can be stated as follows:

- Al_2O_3 and TiO_2 are generally regarded as safe materials for human being and animals (they are actually used in cosmetic products and water treatment).
- TiO_2 and Al_2O_3 nanoparticles are easily obtained (they are produced in very large industrial scales).
- TiO_2 and Al_2O_3 nanofluids have an excellent stability even without using any stabilizer.

2 PREDICTION OF THERMOPHYSICAL PROPERTIES

As previously mentioned, investigating the efficacy of Al_2O_3 and TiO_2 nanofluids as coolants in a double-tube heat exchanger is the aim of this study. At the first step, the heat characteristics of the nanofluids have been evaluated and at the next step the application of nanofluids as coolants have been considered for increasing the heat transfer performance of the double-tube heat exchanger. Some of the properties of nanoparticles and base fluid are listed in table 1. The necessary thermophysical properties in this paper are density, viscosity, specific heat and thermal conductivity.

Table 1 Thermophysical properties of nanoparticles and base fluid

Property	Ethylene glycol	Al_2O_3	TiO_2
c [J kg^{-1}K^{-1}]	2415	765	686.2
ρ[kg m^{-3}]	1111	3970	4250
k [Wm^{-1}K^{-1}]	0.252	40	8.9538

The commonly used models for these properties are given as follows.

For density [12]:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p \quad (1)$$

Specific heat [13]:

$$c_{p,nf} = \frac{(1-\phi)\rho_{bf}c_{p,bf} + \phi\rho_p c_{p,p}}{\rho_{nf}} \quad (2)$$

Viscosity [14]:

$$\mu_{nf} = \mu_f(306\phi^2 - 0.19\phi + 1) \text{ for } Al_2O_3/EG \quad (3)$$

$$\mu_{nf} = \mu_f(10.6\phi^2 + 10.6\phi + 1) \text{ for } TiO_2/EG \quad (4)$$

where ϕ is nanoparticle volume concentration and ρ_p, ρ_{bf} and $c_{p,p}$, $c_{p,bf}$ are the densities and the specific heats of the nanoparticles and base fluid, respectively.

2.1. Modeling thermal conductivity

Thermal conductivity is an important parameter in the field of nanofluid heat transfer. In order to verify the accuracy of the results, two experimental models are presented to predict the thermal conductivity of the nanofluids. The first model is the model presented by Yu and Choi (Y-C) as follows [15]:

$$k_{nf,Y-C} = \left[\frac{k_p + 2k_{bf} + 2(k_p - k_{bf})(1+\beta)^3\phi}{k_p + 2k_{bf} - (k_p - k_{bf})(1+\beta)^3\phi}\right]k_{bf} \quad (5)$$

where k_{nf} is the thermal conductivity of the nanofluid, k_p is the thermal conductivity of the nanoparticles, k_{bf} is the thermal conductivity of the base fluid and β is the ratio of the nanolayer thickness to the original particle radius, β=0.1 is used normally for calculating thermal conductivity of the nanofluids. In this model, the effect of a liquid nanolayer on the surface of the nanoparticle is considered. The second model is the Hamilton-Crosser (H-C) model. This model is presented for particles uniformly dispersed in a continuum medium as follows [16]:

$$k_{nf,H-C} = \left[\frac{k_p + (n-1)k_{bf} - (n-1)\phi(k_{bf} - k_p)}{k_p + (n-1)k_{bf} + \phi(k_{bf} - k_p)}\right]k_{bf} \quad (6)$$

where n is the shape factor given by $n=3/\psi$ and ψ is the sphericity defined as the ratio of the surface area of a sphere having an equal volume to that of the particle. (ψ=1 for spherical particles for ϕ <30%).

3 HEAT TRANSFER AND PRESSURE DROP MODELING

3.1. Heat transfer modeling

The rate of heat transferred to the cooling hot solvent in a double-tube heat exchanger can be written as follows:

$$Q = \dot{m}_h c_{p,h}(T_1 - T_2) \cong \dot{m}_{nf} c_{p,nf}(t_2 - t_1) \quad (7)$$

where h and nf pertain to the hot solvent and nanofluids as coolants, respectively. The total heat transfer area of a double-tube heat exchanger (A) is computed from the following equations:

$$A = \frac{Q}{U \times F \times LMTD} \quad (8)$$

$$LMTD = \frac{(T_1 - t_1) - (T_2 - t_2)}{\ln\frac{(T_1 - t_1)}{(T_2 - t_2)}} \quad (9)$$

where U is the total heat transfer coefficient and F is the temperature correction factor, which in the case of the countercurrent flow can be taken equal to 1. The total heat transfer coefficient (U) can be calculated by Eq. (10) as follows:

$$U = \left(\frac{1}{h_{h,o}} + \frac{1}{h_{nf}} + Rf\right)^{-1} \quad (10)$$

where Rf is the fouling resistance, $h_{h,o}$ is the heat transfer coefficient of hot solvent that referred to the external area and h_{nf} is the heat transfer coefficient of the nanofluid as coolant. Considering Eq. (10), the heat transfer coefficients of hot solvent and nanofluid must be calculated. The heat transfer coefficient of the hot solvent flowing inside the tube under a turbulent regime (Re>10000) can be calculated as follows [17]:

$$h_h = 0.023 Re_h^{0.8} Pr_h^{0.33} \left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.14} \frac{k_h}{D_i} \quad (11)$$

where D_i is the internal diameter of the internal tube, $(\frac{\mu_{nf}}{\mu_{wnf}})^{0.14}$ is the viscosity correction factor. In the above equation the Reynolds and Prandtl numbers are calculated considering the hot solvent properties as follows:

$$Re_h = \frac{\rho_h u_h D_i}{\mu_h} \quad (12)$$

$$Pr_h = \frac{c_{p,h}\mu_h}{k_h} \quad (13)$$

Consequently, the heat transfer coefficient of the hot solvent which referred to the external area, $h_{h,o}$, is defined as (For further study, see [17]):

$$h_{h,o} = h_h\left(\frac{D_i}{D_o}\right) \quad (14)$$

where D_o is the external diameter of the internal tube. The heat transfer coefficient of nanofluid coolant flowing for Re<2100 in the annular can be calculated as follows [17]:

$$h_{nf} = 1.86\left(Re_{nf} Pr_{nf} \frac{D_{eq}}{L}\right)^{0.33} \left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.14} \frac{k_{nf}}{D_{eq}} \tag{15}$$

where L is the heat transfer length and D_{eq} is the equivalent diameter which is expressed in the following form:

$$D_{eq} = \frac{4 \times \text{flow area}}{\text{internal tube perimeter}} = \frac{(D_s^2 - D_o^2)}{D_o} \tag{16}$$

where D_s is the internal diameter of the external tube. In Eq. (15), the Reynolds and Prandtl numbers are calculated considering the nanofluid properties as follows:

$$Re_{nf} = \frac{\rho_{nf} u_{nf} D_{eq}}{\mu_{nf}} \tag{17}$$

$$Pr_{nf} = \frac{c_{p,nf} \mu_{nf}}{k_{nf}} \tag{18}$$

It is important to note that the physical properties appeared in Eqs. (11) and (15) must be evaluated at average temperature (mean inlet and outlet temperatures). In addition, the viscosity correction factor is the ratio of nanofluid viscosity at the mean fluid temperature to viscosity of nanoflouid at the mean tube wall temperature. This factor must be exactly calculated for computing the heat transfer coefficients for both hot solvent and the nanofluid coolant. But the viscosity of the nanofluid at wall temperature cannot be calculated explicitly because this temperature is unknown. Therefore, as a first approximation, it is assumed to be 1. With this simplification, the first value for the coefficients $h_{h,o}$ and h_{nf} is obtained. Then, T_w is calculated by equating the heat transfer rates at both sides of the tube wall as follows:

$$q_{conv} = h_{nf}(T_w - t_{ave}) = h_{h,o}(T_{ave} - T_w) \tag{19}$$

Using the above equation, T_w was obtained in order to calculate the viscosity correction factor for modification of the previous values of $h_{h,o}$ and h_{nf}. Moreover, in this paper, flow rate of nanofluids as coolants in the double-tube heat exchanger was optimized to radiate the same amount of heat. According to the flow diagram depicted in Fig. 1, $\dot{m}_{nf}$ is assumed as the first coolant outlet temperature.

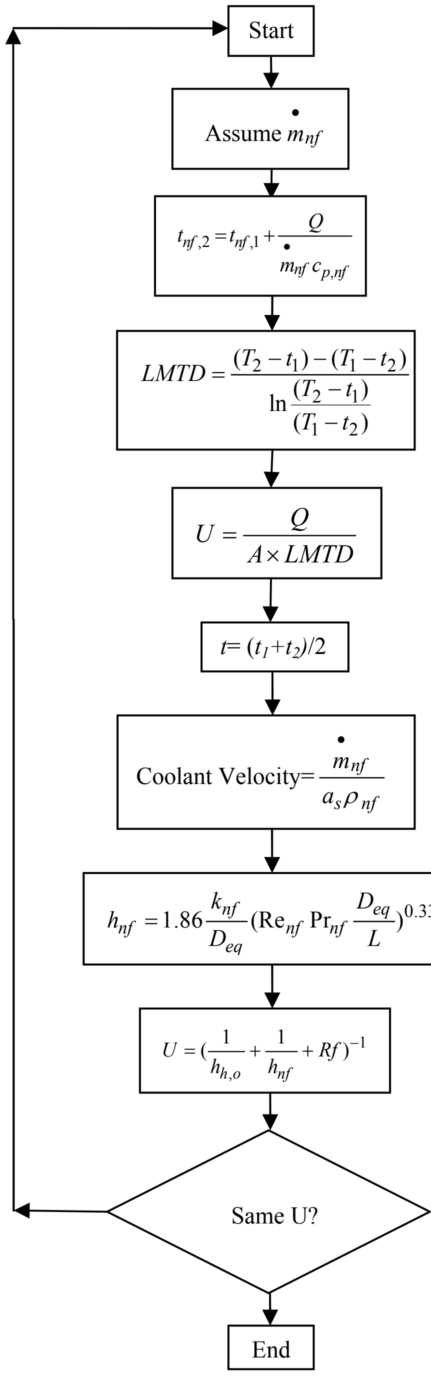

Fig. 1 The flow diagram illustrates the calculating steps of the flow rate optimization of nanofluids as coolants in the double-tube heat exchanger

Afterwards LMDT is calculated from Eq. (9), which is subsequently used to calculate the total heat transfer coefficient from Eq. (8). This coefficient must be compared with the total heat transfer coefficient calculated by Eq. (10). Finally each $\dot{m}_{nf}$ that makes both values approximately equal is chosen as the optimum value of $\dot{m}_{nf}$.

In this work, the inlet and outlet temperatures of hot solvent stream are equal to 40°C and 30°C, respectively. The flow rate of hot solvent stream is 0.8 kg s^{-1} and its specific heat is equal to 1922 J $kg^{-1}K^{-1}$. The inlet temperature of nanofluid coolant is equal to 5°C. The length of tube is 6m. The internal and external diameters of internal tube are 0.035m and 0.0421m and the internal diameter of external tube is 0.0525m. The fouling resistance is assumed to be 5×10^{-4} m^2KW^{-1}.

3.2. Pressure drop modeling

The friction factor of nanofluids in laminar flow regime can be calculated using the formula presented as follows:

$$f_{nf} = 16 / Re_{nf} \tag{20}$$

In this paper, the pressure drop (Δpnf) and pumping power (PP) for Al2O3/EG and TiO2/EG nanofluids used as coolants in the double-tube heat exchanger are calculated as follows [17]:

$$\Delta p_{nf} = 2\frac{f_{nf} L \rho_{nf} u_{nf}^2}{D'_{eq}}\left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.25} \tag{21}$$

$$PP = u_{nf} a_s \Delta p_{nf} \tag{22}$$

where D'_{eq} is the equivalent diameter of an annulus given by $D'_{eq} = D_s - D_o$ and as is the annular flow area.

4 SIMULATION RESULTS AND DISCUSSION

4.1. Heat transfer

As mentioned previously, two experimental models have been applied to predict the thermal conductivity of the nanofluids. As shown in Fig. 2, the thermal conductivity of nanofluids varying in the range of 1-10% has been calculated, using these models. The obtained results are important for optimizing the total heat transfer area and flow rate of the coolant in this work. As can be seen, thermal conductivity increases as nanoparticle volume concentration, is increased.

Fig. 3 shows the effect of nanoparticles on the heat transfer coefficient. Results show that the heat transfer coefficient can be enhanced by adding nanoparticles to the base fluid. It is important to note that increase in nanoparticle concentration, increases the fluid viscosity and hence decreases the Reynold's number whereby the heat transfer coefficient decreases. But, Fig. 3 shows that with increasing nanoparticle concentration the heat transfer coefficient increases. This indicates that thermal conduction enhancement plays a more significant role in the convective heat transfer compared with the viscosity increase under the conditions of this study. Enhancement of heat transfer by nanofluids may be resulted from the following two aspects: first the incremented particles which increase the thermal conductivity of the mixture; second, chaotic movement of ultrafine particles accelerates the energy exchange between the fluid and the wall of the heat exchanger. For example, Figs. 2 and 3 show that for 4% TiO_2/EG nanofluid, k_{nf}=0.2909, 0.2809 and h_{nf}=316.6094, 309.375 was calculated using the (Y-C) and (H-C) models, respectively. In the present paper, the total heat transfer coefficient for Al_2O_3/EG and TiO_2/EG nanofluids in the 1-10% vol. range, was calculated to estimate the efficacy of using nanofluids as coolants in the double-tube heat exchanger. According to Fig. 4, with increasing probability of collision between nanoparticles and heat exchanger wall, under higher coolant concentration, the total heat transfer coefficient increases. Accordingly, from the results shown in Fig. 4 it is evident that Al_2O_3/EG and TiO_2/EG nanofluids have considerable potential for use in the double-tube heat exchanger. For example, an inspection of Fig. 4 shows that total heat transfer coefficients of Al_2O_3/EG and TiO_2/EG nanofluids in the 1-10%vol. range have increased by ~ (1.81-18%), (1.37-13.6%) and (1.7-16.77%), (1.27-12.67%) using the (Y-C) and (H-C) models, respectively.

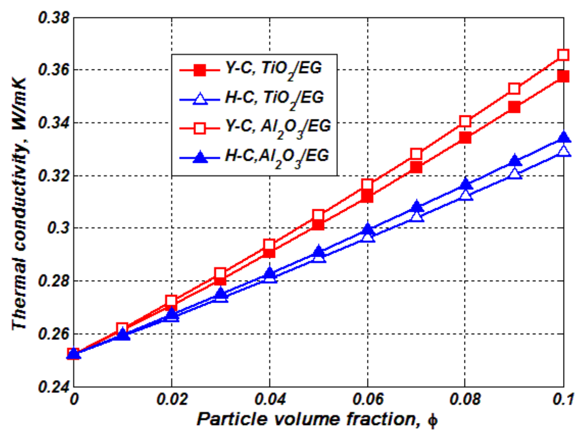

Fig. 2 Variation of thermal conductivity versus particle volume fraction for Al2O3/EG and TiO2/EG nanofluids

According to Fig. 5, wall temperature reduction (%) was used in the double-tube heat exchanger for Al_2O_3/EG and TiO_2/EG nanofluid coolants. For example, Fig. 5 shows that for Al_2O_3/EG and TiO_2/EG nanofluids at 2% volume concentration used as coolants, the wall temperature reduction corresponds to ~ 0.92 %, 0.7% and 0.85%, 0.65% using the (Y-C) and (H-C) models, respectively. As can be seen in Figs. 6 and 7, the total heat transfer area and flow rate of the coolant are optimized using Al_2O_3/EG and TiO_2/EG nanofluids as coolants to radiate 15.4 kW of heat from a double-tube heat exchanger. It can be seen that by using Al_2O_3/EG and TiO_2/EG nanofluids in the 1-10 % vol. range, the total heat transfer area of the double-tube heat exchanger has decreased by~ (1.78-15.3%), (1.36-12%) and (1.67-14.4%), (1.26-11.3%) using the (Y-C) and (H-C), models respectively. As shown in Fig. 7, it is possible to reach the desired heat transfer rate, reducing flow rate by~ (2.9-25.7%), (1.75-17%) using the (Y-C) and (H-C), models respectively, by using Al_2O_3/EG nanofluid in the 1-10 %vol. range.

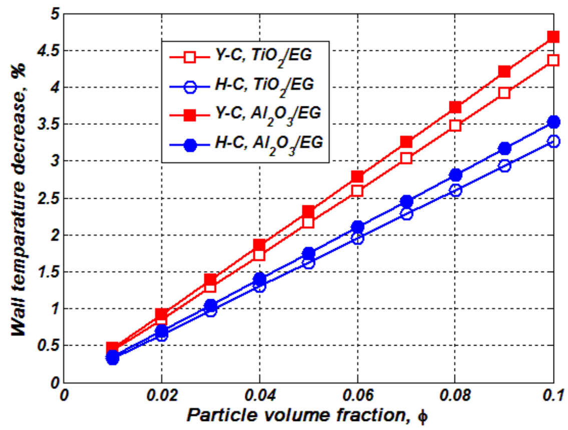

Fig. 5 Effect of nanoparticles volume fraction on wall temperature in a double-tube heat exchanger

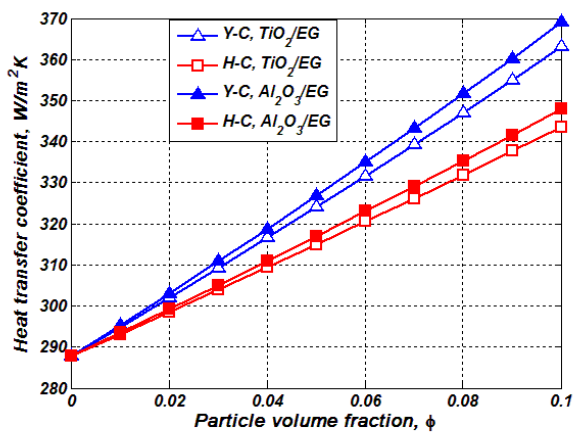

Fig. 3 Heat transfer coefficient of Al2O3/EG and TiO2/EG nanofluids as coolants in a double-tube heat exchanger

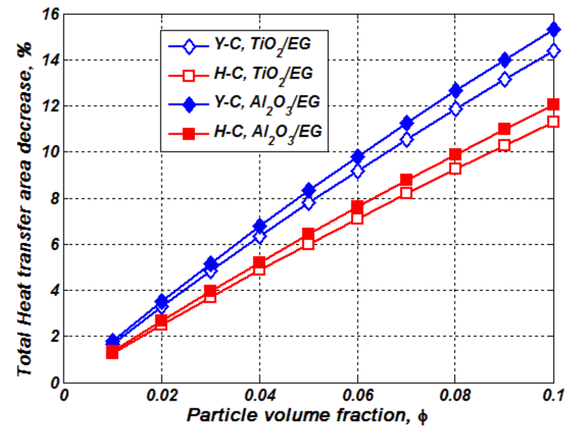

Fig. 6 Total heat transfer area reduction in a double-tube heat exchanger, using Al2O3/EG and TiO2/EG nanofluids as coolants

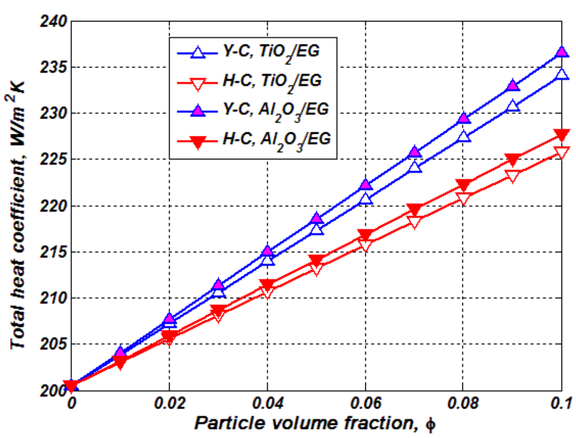

Fig. 4 Total heat transfer coefficient of Al2O3/EG and TiO2/EG nanofluids as coolants in a double-tube heat exchanger

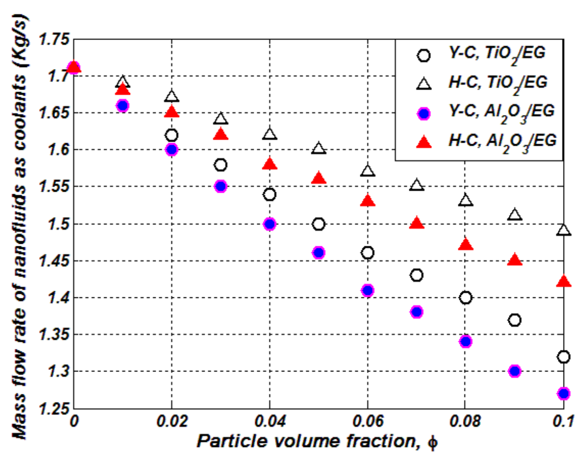

Fig. 7 Flow rate reduction of the coolant in a double-tube heat exchanger using Al2O3/EG and TiO2/EG nanofluids as coolants

4.2. Pressure drop

In practical application of nanofluids, it is essential to study the pressure drop, developed during the flow of such coolants. Therefore, the effect of Al_2O_3/EG and TiO_2/EG nanofluids on the friction factor, pressure drop and pumping power was studied according to Figs. 8-10. As the nanoparticles loaded into the base fluid increase, the viscosity and density of the base fluid also increases; causing higher friction factor and pressure drop.

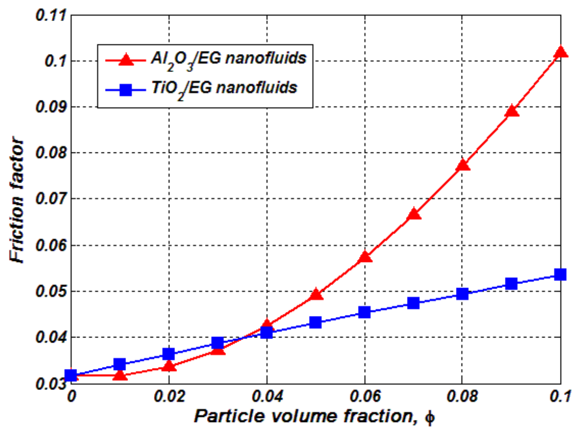

Fig. 8 Comparison of friction factor for Al2O3/EG and TiO2/EG nanofluids

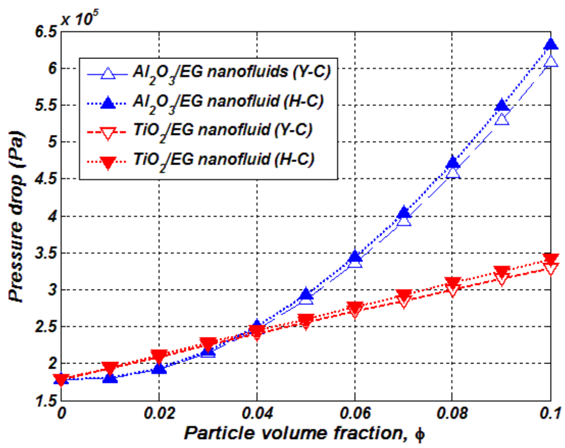

Fig. 9 Comparison of pressure drop for Al2O3/EG and TiO2/EG nanofluids

From the investigations, it can be inferred that, the pressure drop and pumping power of nanofluids at low nanoparticle volume concentration range (1%-3.0%) is approximately the same as in the base fluid, but higher nanoparticle volume concentration leads to a penalty drop in the pressure. For example, when Al_2O_3/EG nanofluid with concentrations of 3.0% and 4.0% was used as coolant in the double-tube heat exchanger, the increase of pumping power was calculated to be~ 20.32%, 21.89% of the original base fluid using the (Y-C) model and 38%, 40.4% of the original base fluid, using the (H-C) model, respectively.

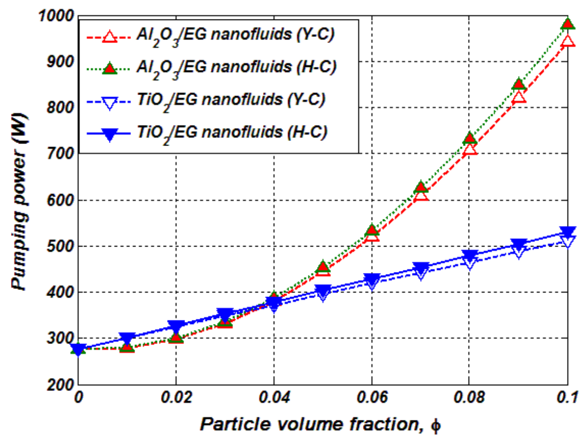

Fig. 10 Pumping power for Al2O3/EG and TiO2/EG nanofluids in a double-tube heat exchanger at different volume concentrations

5 CONCLUSIONS

In the present study, the use of Al_2O_3/EG and TiO_2/EG nanofluid coolants in the 1-10 %vol. range was investigated to optimize heat transfer performance of a double-tube heat exchanger in laminar flow. The (Y-C) and (H-C) models were applied to predict the thermal conductivity of nanofluids and the calculations have been carried out based on these models. For example, by using 3% Al_2O_3/EG nanofluid, alone, to radiate 15.4 kW of heat from the double-tube heat exchanger with an increase in the total heat transfer coefficient of the base fluid, the total heat transfer area and flow rate of the coolant was decreased by~4.1%, 5.26%, respectively, while the pumping power was increased by~ 21.9% (using the (H-C) model).

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