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Application of nanofluids in heating buildings and reducing pollution

Devdatta P. Kulkarni, Debendra K. Das *, Ravikanth S. Vajjha

Department of Mechanical Engineering, University of Alaska Fairbanks, P.O. Box 755905, Fairbanks, AK 99775-5905, USA

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ABSTRACT

This paper presents nanofluid convective heat transfer and viscosity measurements, and evaluates how they perform heating buildings in cold regions. Nanofluids contain suspended metallic nanoparticles, which increases the thermal conductivity of the base fluid by a substantial amount. The heat transfer coefficient of nanofluids increases with volume concentration. To determine how nanofluid heat transfer characteristics enhance as volume concentration is increased; experiments were performed on copper oxide, aluminum oxide and silicon dioxide nanofluids, each in an ethylene glycol and water mixture. Calculations were performed for conventional finned-tube heat exchangers used in buildings in cold regions. The analysis shows that using nanofluids in heat exchangers could reduce volumetric and mass flow rates, and result in an overall pumping power savings. Nanofluids necessitate smaller heating systems, which are capable of delivering the same amount of thermal energy as larger heating systems using base fluids, but are less expensive; this lowers the initial equipment cost excluding nanofluid cost. This will also reduce environmental pollutants because smaller heating units use less power, and the heat transfer unit has less liquid and material waste to discard at the end of its life cycle.

1. Introduction

Energy costs have escalated rapidly in the last decade and there is a tremendous need for new kinds of heating/cooling fluids, which will increase heating system thermal performance, reduce the overall size and energy consumption. Nanofluids are the new generation of heat transfer fluids for various industrial and automotive applications because of their excellent thermal performance [1]. Nanofluids are nanometer-sized particles (<100 nm) dispersed in a base fluid such as water, ethylene glycol or propylene glycol. Addition of high thermal-conductivity metallic nanoparticles (e.g., copper, aluminum, silicon and silver) increases the thermal conductivity of such mixtures; thus enhancing their overall energy transport capability [2].

Eastman et al. [3] showed a 40% increase in thermal conductivity by adding 0.3 vol.% copper nanoparticles to ethylene glycol. Recently, Prasher et al. [4] described that the increase in nanofluid thermal conductivity is primarily due to the convection caused by the Brownian movement of nanoparticles. Using nanofluids in heat transfer applications will provide numerous benefits including improved heat transfer, minimal clogging, and miniaturization of heat exchangers with microchannels. Use of nanofluids will conserve energy by reducing the necessary pumping power [5]. These benefits make nanofluids a future generation heat transfer fluid.

In cold climates like those found in Alaska, Canada and other circumpolar regions, heat transfer fluids regularly encounter very low temperatures on the order of −40 °C. It is a common practice to use ethylene or propylene glycol mixed with water in different proportions as a heat transfer fluid [6] for automobiles, heat exchangers and baseboard heaters in buildings. Inhibited ethylene glycol and propylene glycol are used as aqueous freezing point depressants and heat transfer media in heating, ventilation and air conditioning (HVAC) systems [7]. Their main attributes are the ability to lower the freezing point of water, low volatility and relatively low corrosivity. Ethylene glycol solutions have better heat transfer properties than propylene glycol solutions, especially at low temperatures. The commonly used mixture in cold climates is 60% by weight ethylene glycol and 40% by weight water (60:40 EG/W) [8], and the thermal performance could be enhanced by adding metallic oxide nanoparticles. These nanoparticles affect the viscosity of the mixture to a great degree than other thermophysical properties.

Therefore, major goals of this study were: (1) investigation of the viscosity of various nanofluids, (2) determination of heat transfer coefficients and pressure loss of various nanofluids, and (3) application of nanofluids to building heating systems.

Since no viscosity data is currently available in the literature for such nanofluids at subzero temperatures, investigating and reporting on nanofluid rheology is very important to expanding nanofluid applications in cold regions. In the present paper, the viscosity of ethylene glycol and water with up to 6% by volume of copper oxide

* Corresponding author. Tel.: +1 907 474 6094; fax: +1 907 474 6141.
E-mail address: ffdkd@uaf.edu (D.K. Das).

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Nomenclature

A	area, m^2
C_p	specific heat, J/kg K
D	inside diameter of the tube, m
h	convective heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
m	mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Q	heat transfer rate, W
q''	heat flux, W/m^2
Re	Reynolds number
Δt_m	log mean temperature difference, K
v	mean velocity, m/s
V	volumetric flow rate, m^3/s
X	axial length in meter
x	thickness, m

Greek letters

μ	coefficient of dynamic viscosity of the fluid, mPa s
φ	particle volume concentration
η	efficiency
ρ	density of the fluid, kg/m^3

Subscripts

i	inside
f	fluid
F	fin
m	mean
nf	nanofluid
o	outside
p	pipe and Particle
s	solid material
w	wall

(CuO), aluminum oxide (Al_2O_3) and silicon dioxide (SiO_2) nanoparticles has been investigated with temperatures ranging from −35 °C to 50 °C, which is the range of operation encountered in cold regions. Furthermore, the convective heat transfer coefficient and pressure loss for these nanofluids have been reported herein from the experiments. The uniqueness of these experiments is that the nanofluids are prepared in 60:40 ethylene glycol and water solution (binary fluid) as the base fluid. The currently published papers show the heat transfer enhancement in water or ethylene glycol alone. Finally, these values of heat transfer coefficient are applied in a building case study to determine the surface area reduction of the heating system, volumetric flow reduction of the heat transfer fluid and pumping power reduction by using nanofluids.

The nanoparticles were obtained from Alfa Aesar [14] as colloidal dispersion with dispersant in 50% water by weight and subsequently different volume concentrations up to 6% were prepared by adding proper amounts of 60:40 EG/W (by weight) with a precision mass balance. The average particle size for CuO was 30 nm, Al_2O_3 was 45 nm and SiO_2 was 50 nm. Before each experiment the nanofluid sample was placed in a sonicator bath for approximately 2 h to ensure proper dispersion of nanoparticles and to prevent any agglomeration. The prepared samples were inspected using dynamic light scattering (DLS) technique for average particle diameter in the fluid. From the results of DLS, it is confirmed that the nanoparticles are dispersed uniformly in the suspension.

2. Theory

The convective heat transfer coefficient in a heat transfer coil in building heating is strongly dependent upon the Reynolds number in internal flow.

Reynolds number for nanofluid flowing through a pipe is given by:

$$Re = \frac{\rho_{nf} v D}{\mu_{nf}} \tag{1}$$

The density of nanofluid, ρ_{nf} is given by [9]

$$\rho_{nf} = \phi\rho_s + (1-\phi)\rho_f \tag{2}$$

where the volumetric concentration is given by [5]

$$\phi = \frac{\rho_f \phi_m}{\rho_f \phi_m + \rho_s(1-\phi_m)}; \text{ where } \phi_m \text{ is the mass fraction} \tag{3}$$

The thermal conductivity of nanofluid is obtained from the well-known Hamilton–Crosser model:

$$K_{nf} = k_f\left[\frac{k_s + (n-1)k_f - (n-1)\phi(k_f - k_s)}{k_s + (n-1)k_f + \phi(k_f - k_s)}\right] \tag{4}$$

The effective specific heat of nanofluids was given by [9]:

$$C_{pnf} = (1-\phi)C_{pf} + \phi C_{ps} \tag{5}$$

However, Buongiorno [10] asserted that the specific heat of a nanofluid should be calculated assuming that the nanoparticles and the base fluid are in thermal equilibrium. He presented the equation as:

$$C_{pnf} = \frac{\phi\rho_s C_{ps} + (1-\phi)\rho_f C_{pf}}{\rho_{nf}} \tag{6}$$

Volumetric flow rate of a nanofluid in the pipe is given by:

$$V_{nf} = A_{p,i} v \tag{7}$$

Based on a pressure loss ΔP the pumping power (W) to circulate the nanofluid is given by:

$$W = \frac{V_{nf}\Delta P}{\eta_{pump}} \tag{8}$$

where η_{pump} is pump efficiency, assumed here as 70%.

3. Heat transfer experiment

An experimental apparatus was built to study the heat transfer and flow characteristics of a conventional ethylene glycol/water mixture and various nanofluids flowing through a tube. The experimental setup is shown in Fig. 1 and consists of a pump, heat transfer test section, a counterflow cooling heat exchanger, flowmeter, a flow totalizer, differential pressure transducer, bypass valve, reservoir and several datalogges.

As nanofluids are expensive at the present time, the test system was designed in such a way that a small amount of the nanofluid (approx. 2 l) would be sufficient to investigate nanofluid heat transfer and fluid dynamic performance. The heat transfer test section is a straight copper tube with inside diameter of 3.37 mm (0.131 in.) and a length of 1 m (3.28 ft). Six type-T (copper-constantan) thermocouples mounted on the tube surface along the length measure the wall temperature. Two thermowells at the inlet

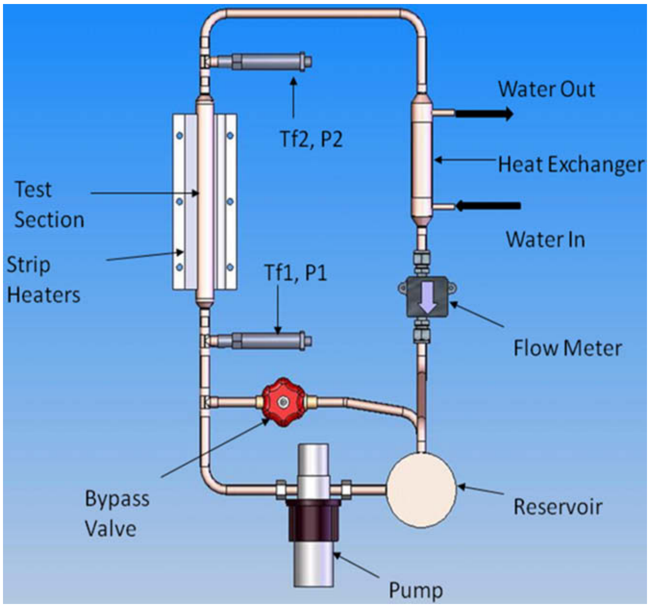

Fig. 1. Experimental setup to investigate the convective heat transfer coefficient of nanofluids.

and outlet of the test section measured the nanofluid inlet and outlet temperature. Two plastic fittings at inlet and outlet section of the copper tube provide a thermal barrier to axial heat conduction. For turbulent flow, the hydrodynamic and thermal entry length in a tube is $X/D = 10$ [11]. In our experimental setup, this length is 3.37 cm beyond which all measurements are taken to ensure that the measurements are in the fully developed regions. To attain the constant heat–flux boundary condition, the test section is heated electrically by four strip heaters capable of delivering 1 kW each. To measure the power input accurately, four power meters are connected to four variacs. To minimize the heat loss from the heat transfer test section to ambient air, the test system is insulated by 10 cm of fiber glass. A four-pass shell and tube counterflow heat exchanger cools the nanofluids to keep the inlet fluid temperature constant using shop water. A bypass valve controls the nanofluid circulation rate. A differential pressure transducer placed across the inlet and the outlet of the test section measured the pressure drop accurately.

During the experiments, the tube wall temperatures, fluid inlet and outlet temperatures, volumetric flow rate of the fluid and power supplied are measured. Using this data, the convective heat transfer coefficient of the nanofluid (h_{nf}) is determined as:

$$h_{nf} = \frac{q''}{t_w - t_f} \tag{9}$$

where t_w is the average wall temperature and t_f is the average of fluid inlet and outlet temperatures; q'' is the heat flux supplied to the test section. The heat provided can be equated to heat gained by fluid flowing through the test section. It is obtained from:

$$Q = mC_p\Delta t_f \tag{10}$$

Dividing Q by the outer surface area of the test section tube the heat flux q'' is derived. Before determining the nanofluid convective heat transfer coefficient, the apparatus was calibrated using deionized water. The experimental results were compared with the Dittus–Boelter equation from Bejan [11] given below for fully developed turbulent flow.

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{11}$$

It was found that the experimental results and values obtained by the Dittus–Boelter equation were within ±10%. To ensure consistency, the test runs were repeated and the results were reproduced with similar accuracy. In the present study, the Reynolds number was varied from 3000 to 12000, which is commonly employed in hydronic heating systems in buildings.

4. Experimental results

4.1. Viscosity

The experimental setup for viscosity measurements using Brookfield viscometer for CuO nanofluid is described by Kulkarni et al. [12] and for Al_2O_3 and SiO_2 nanofluids by Namburu et al. [13]. About 8–14 viscosity measurements were recorded at various shear rates at specific temperatures for each nanofluid volume concentration. As an example, Fig. 2 shows the shear strain rate versus shear stress (dyne/cm^2) for 6.12% copper oxide nanoparticles in EG/water at -35 °C. From this figure, it is evident that the nanofluid behaves as Newtonian fluid as shear stress is directly proportional to the shear strain rate. The slope of the graph can be represented as apparent viscosity of the nanofluid at that particular temperature. Similar procedure is repeated every time to derive a viscosity value of each concentration of nanofluid at set temperatures.

From these results, the viscosity plot was generated (see Fig. 3), and it illustrates that viscosity is dependent on nanoparticle concentration of copper oxide nanoparticles. As the volume concentration increases, the nanofluid viscosity increases at a specific temperature. Also, nanofluid viscosity is very high at lower temperatures and decreases exponentially as temperature increases [15]. Similar viscosity measurements were carried out for Al_2O_3 and SiO_2 nanofluids. Fig. 4 shows the comparison of viscosity of three different nanofluids of same particle volumetric concentration dispersed in EG/water as a function of temperature. For subsequent heat transfer and pressure loss analysis, we have used these viscosity values at average fluid temperature between inlet and outlet in the test section to get more accurate results.

4.2. Convective heat transfer coefficient

Using Eqs. (9) and (10), the heat transfer coefficient is calculated and plotted in relation to the Reynolds number in Fig. 5. In this analysis, we have considered the specific values for density, thermal conductivity and specific heat of nanofluids from Eqs. (2), (4), and (6). Fig. 5 depicts that as the Reynolds number increases, the heat transfer increases. Also, from this figure it can be concluded that as the particle concentration in the base fluid increases, the heat transfer coefficient at the same Reynolds number increases. As a typical value, at a fixed Reynolds number of 8000, copper oxide nanofluid with 6%, 4% and 2% volume concentration exhibit 61%, 37.5% and 16.7% enhancement in heat transfer coefficient over the base fluid, respectively.

The heat transfer coefficient results are also compared with different types of nanofluids as shown in Fig. 6. It can be concluded from this figure that the copper oxide nanofluid shows highest enhancement in heat transfer coefficient followed by aluminum oxide and silicon dioxide for the same concentration (6%) of nanoparticles. This enhancement in heat transfer coefficient is mainly due to the higher thermal conductivity of the nanofluid. As thermal conductivity is higher for copper compared to other materials in this study, it shows higher increase in heat transfer coefficient. As a typical value, at a fixed Reynolds number of 8000, copper oxide, aluminum oxide and silicon dioxide exhibit 61%, 35.4% and 18.4% enhancement in heat transfer coefficient, respectively.

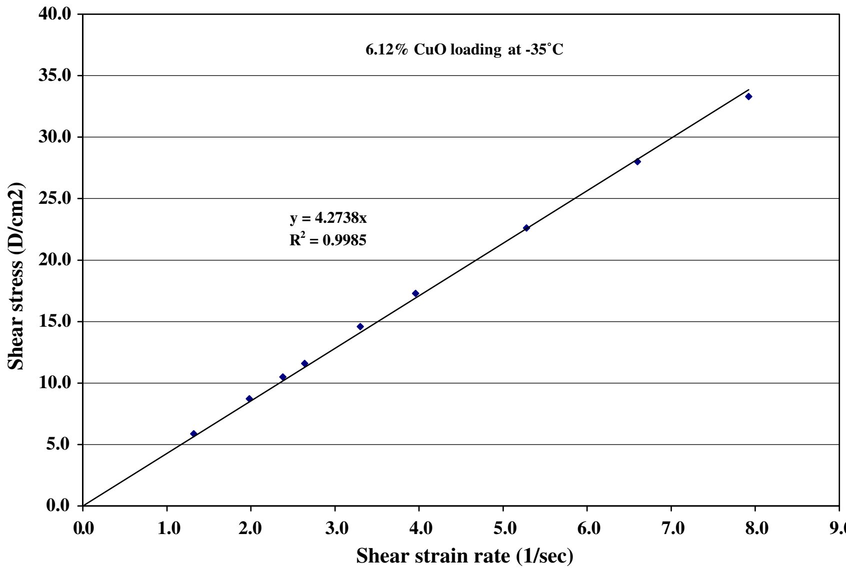

Fig. 2. Shear stress in dyne/cm^2 versus shear strain rate for 6.12 vol.% CuO loading in EG/water at −35 °C.

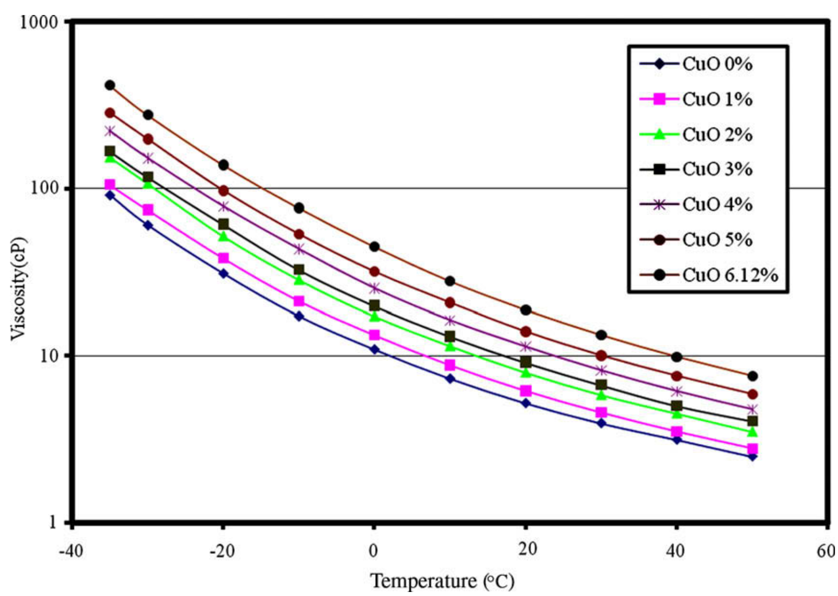

Fig. 3. Viscosity in centi poise of copper oxide nanofluids in EG/water mixture at various concentrations and with varying temperatures.

4.3. Pressure loss

For a fixed Reynolds number, as the heat transfer coefficient increases; there is also an increase in pressure loss. This increase in pressure loss is mainly attributed to an increase in the viscosity and density of the nanofluid. CuO nanofluid demonstrates the highest viscosity and density and therefore, has the highest pressure loss at a given Reynolds number as shown in Fig. 7. Nanofluid pressure loss and heat transfer performance should be carefully considered when choosing a nanofluid for a particular application.

5. Analysis

5.1. Reduction in flow rates and pumping power

After characterizing different nanofluids for their heat transfer and fluid dynamic performance, we performed a detailed analysis of energy requirement for a cold climate building. In this analysis, we held constant thermal performance (overall heat transfer coefficient) for all fluids. In other words, the inside heat transfer coefficient is maintained same for all fluids. At a Reynolds number of 8000, the base fluid has a 14,400 W/m^2 K convective heat transfer coefficient from Fig. 6. Taking this as a reference, an analysis was carried out for a constant inside heat transfer coefficient of this magnitude for the other nanofluids. Based on the results from earlier experiments presented in Fig. 6, the corresponding Reynolds numbers for this heat transfer coefficient of the different fluids were used to determine velocity, volumetric flow rate and pumping power. The results were compared with the ethylene glycol and water mixture. These calculations used the same viscosity values reported in Fig. 4 and were taken at the average of inlet and outlet fluid temperature. This analysis was performed for the comparison of performance for flow through a tube of 4 mm I.D. and 1 m long, which was essentially the size of our test section. The results from the analysis are summarized in Table 1.

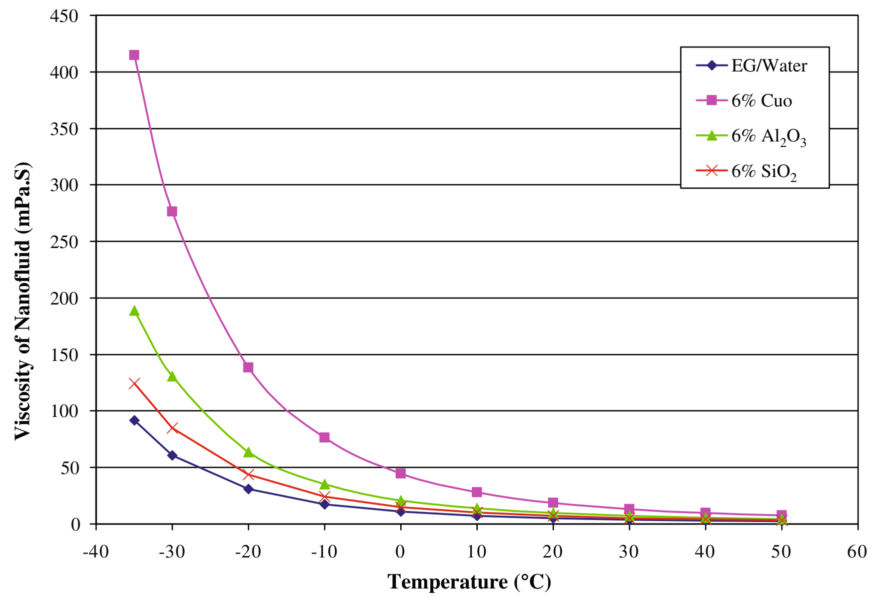

Fig. 4. The viscosity variation of various nanofluids of same concentration and the base fluid with respect to temperature.

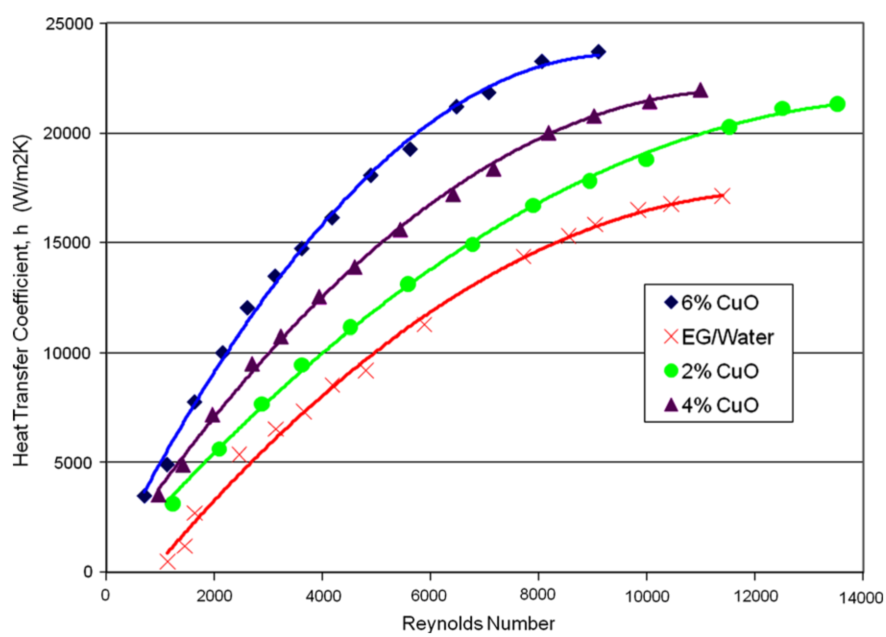

Fig. 5. Effect of nanoparticle (CuO) volume concentration on heat transfer coefficient.

From Table 1, it can be observed that for same heat transfer coefficient (14,400 W/m^2 K); the conventional fluid corresponds to the highest Reynolds number of 8000. Conversely, for the same heat transfer coefficient, the copper oxide nanofluid has the lowest Reynolds number of 3600; the aluminum oxide nanofluid is 4500; and the silicon dioxide nanofluid is 6290. The density (Eq. (2)), specific heat (Eq. (6)) and viscosity (Fig. 4) values of nanofluids are summarized in Table 1. For the same heat transfer coefficient the volumetric flow rate decreases by 29% for the CuO nanofluid, 37.2% for the Al_2O_3 nanofluid and 22.2% for the SiO_2 nanofluid. The volume reduction required for a heat transfer fluid will also reduce the inventory and fluid production.

Accordingly from Eq. (8), the pumping power savings are 11.7% for the CuO nanofluid, 38.3% for the Al_2O_3 nanofluid and 27.6% for the SiO_2 nanofluid. Reducing pumping power will lead to less carbon dioxide (CO_2) release into atmosphere. Similarly, this reduces the mass flow rate by 6.5% for the CuO nanofluid, 28% for the Al_2O_3 nanofluid and 16.3% for the SiO_2 nanofluid. Therefore, it turns out that Al_2O_3 nanofluid is the fluid of choice among these candidates.

6. Case study of nanofluids in liquid–air heat exchanger

In this case study, we have considered the conventional method of heating a building with nanofluids by heating air in duct coils through a bank of tubes with finned surfaces. The coil consists of rectangular-plate aluminum fins bonded to copper tubes. The tubes are arranged in line in adjacent rows. The heat transfer rate (Q) from a heating coil circulating nanofluid to air in a building is expressed as [8]:

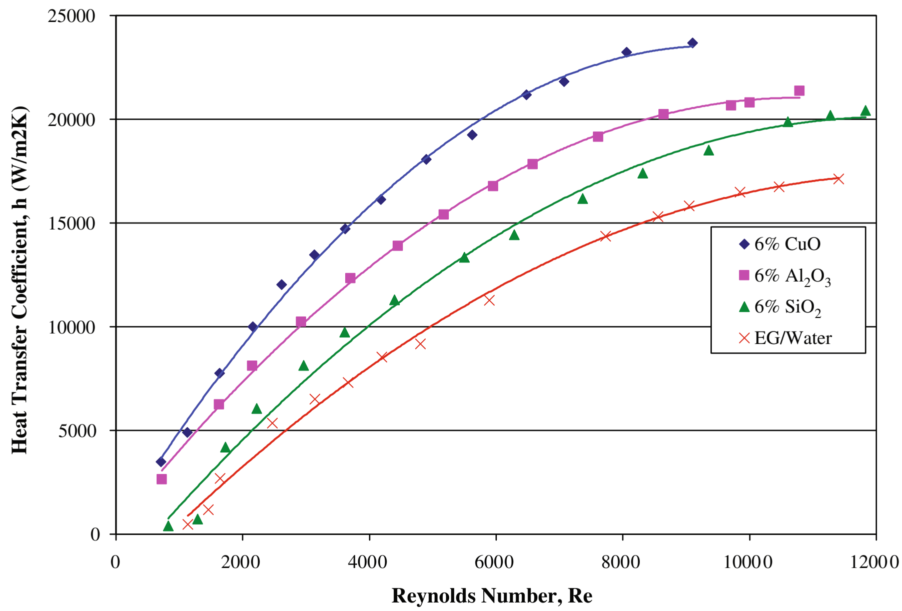

Fig. 6. Effect of different nanoparticles (copper oxide, aluminum oxide and silicon dioxide) on heat transfer coefficient at same concentration.

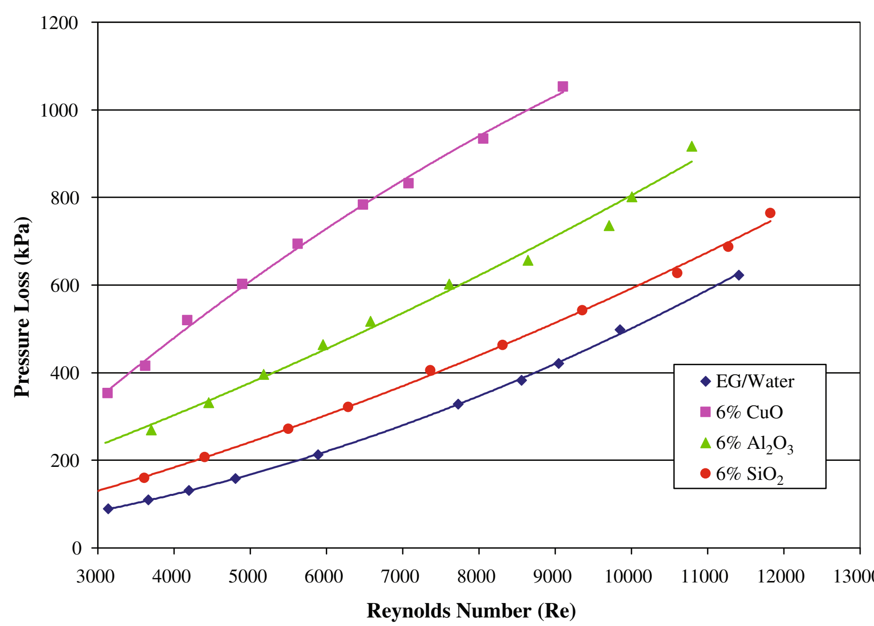

Fig. 7. Pressure loss of various nanofluids with respect to conventional glycol/water mixture.

$$Q = U_o A_o \Delta t_m \tag{12}$$

Rate of heat transfer from the nanofluid to the pipe is given by [8];

$$Q = h_i A_{p,i}(t_f - t_{p,i}) \tag{13}$$

Rate of heat transfer through the pipe wall is given by

$$Q = \frac{k_p A_{p,m}(t_{p,i} - t_{p,o})}{x_P} \tag{14}$$

Rate of heat transfer from the pipe and fin to the air is given by:

$$Q = h_{c,o,P} A_{P,o}(t_{P,o} - t) + h_{c,o,F} A_F (t_{F,m} - t) \tag{15}$$

Rate of heat transfer for the entire process is:

$$Q = U_o A_o (t_f - t) \tag{16}$$

Assuming $h_{c,o,P} = h_{c,o,F} = h_{c,o}$, we have:

$$Q = h_{c,o}(A_{P,o} + \eta A_F)(t_{P,o} - t) \tag{17}$$

From Eqs. (13), (14), (16), and (17), we obtain:

$$U_o = \frac{1}{\frac{A_o}{A_{P,i} h_i} + \frac{A_o x_P}{A_{P,m} k_P} + \frac{1-\eta}{h_{c,o}(A_{P,o}/A_F + \eta)} + \frac{1}{h_{c,o}}} \tag{18}$$

Subscripts i and o represents inner and outer surface area. Typical value for fin efficiency for aluminum fins is about 0.75.

Typical data are given as: $A_{P,i} = 0.03536\ \text{m}^2$, $A_{P,o} = 0.03658\ \text{m}^2$, $A_{P,m} = 0.0375\ \text{m}^2$, $A_F = 0.948\ \text{m}^2$, $A_o = 0.09845\ \text{m}^2$, $h_{c,o} = 57\ \text{W/m}^2\ \text{K}$ and fin efficiency $\eta = 0.75$.

We considered a specific Reynolds number of 4000 and read the corresponding convective heat transfer coefficients, h for all

Table 1
Comparison of the performance of various nanofluids with conventional 60:40 EG/Water mixture.

Type of fluid parameters	60/40 EG/water	6% Copper oxide	6% Aluminum oxide	6% Silicon dioxide
Heat transfer coefficient (W/m^2 K)	14,400	14,400	14,400	14,400
Reynolds number (Re)	8000	3600	4500	6290
Pressure loss (k Pa)	346	430	340	322
Viscosity (m Pa s)	1.1	2.27	1.41	1.17
Density (kg/m^3)	1038	1366	1192	1116
Specific heat (J/kg K)	3120	2339	2718	2821
Velocity (m/s)	2.12	1.5	1.33	1.65
Volumetric flow rate (10E + 5 m^3/s)	2.66	1.89	1.67	2.07
Reduction in volumetric flow rate (%)	–	28.95	37.22	22.18
Power (W)	11.500	10.16	7.1	8.33
Power advantage (W)	–	1.34	4.4	3.17
Power advantage (%)	–	11.65	38.26	27.57
Mass flow rate (kg/s)	0.028	0.0258	0.0199	0.023
Reduction in mass flow rate (%)	–	6.495	27.904	16.333

Table 2
Comparison of the reduction in heat transfer surface area for various nanofluids with conventional 60:40 EG/water mixture.

Component	60:40 EG/water	6% Copper oxide	6% Aluminum oxide	6% Silicon dioxide
Reynolds number	4000	4000	4000	4000
Inside heat transfer coefficient, h_i (W/m^2 K)	3407	10,000	8000	4900
Inside surface resistance × 10E3 $R_i = A_o/A_{P,i}h_i$ (m^2 K/W)	8.17	2.78	3.475	5.67
Pipe wall resistance × 10E5 $R_P = A_o x_P/A_{P,m}k_P$	4.8	4.8	4.8	4.8
Fin resistance × 10E3 $RF = \frac{1}{h_{c,o}}\left(\frac{1-\eta}{\eta+A_{P,o}/A_F}\right)$	6.16	6.16	6.16	6.16
Outside surface resistance × 10E3 $R_o = 1/h_{c,o}$	17.5	17.5	17.5	17.5
Total resistance × 10E3 $R_t = R_t + R_P + R_F + R_o$	31.88	26.48	27.175	29.37
Overall heat transfer coefficient $U_o = 1/R_t$ (W/m^2 K)	31.37	37.76	36.8	34.04
% Reduction in area	–	20.37	17.3	8.5

nanofluids from Fig. 6 which are summarized in Table 2. Based upon the heat exchanger geometry and Eqs. (12)–(18) our calculation yields an overall heat transfer coefficient U_o. The U_o is found to be the highest for CuO nanofluid. Maintaining the same mean temperature difference, the reduction in heat transfer surface area is 20.4% for the CuO nanofluid, 17.3% for the Al_2O_3 nanofluid and 8.5% for the SiO_2 nanofluid. Therefore, there is considerable reduction in the size of the heating coil by employing nanofluids. Reduction in heat exchanger surface area will lead to reduction in mining of materials and less demand for solid waste disposal site at the end of the useful life of the heat transfer system.

7. Conclusions

1. Nanofluid viscosity decreases exponentially as temperature increases from a subzero value. As the volume concentration of nanoparticles increases, the viscosity of nanofluid increases. From the nanofluids tested, the CuO nanofluid has the highest viscosity followed by the Al_2O_3 nanofluid and then by the SiO_2 nanofluid.
2. As the volume concentration of nanoparticles (ranging from 0% to 6%) increases, the heat transfer coefficient increases at the same Reynolds number.
3. The CuO nanofluid has the highest heat transfer coefficient followed by the Al_2O_3 nanofluid and the SiO_2 nanofluid. For example, at Re = 4000, h_{nf} is 10,000 W/m^2 K for the CuO nanofluid, 8000 W/m^2 K for the Al_2O_3 nanofluid and 4900 W/m^2 K for the SiO_2 nanofluid.
4. Pressure loss is also highest for the CuO nanofluid, followed by the Al_2O_3 nanofluid and then the SiO_2 nanofluid.
5. Replacing conventional ethylene glycol/ water mixture with nanofluids as heat transfer fluid, one can reduce the volumetric flow rate, mass flow rate and the pumping power for the same heat transfer rate.
6. Use of nanofluids to heat buildings can reduce the size of the heat transfer system and reduce the accompanying pressure loss and the subsequent pumping power. This will reduce energy consumption that comes from power plants and will thus indirectly reduce environmental pollution.
7. Similar benefits can be derived by considering nanofluids in place of chilled water in building cooling coils. An investigation similar to the one presented in this paper can quantitatively establish the benefits.
8. Use of nanofluids will reduce material volume necessary for heat exchanger, pump, piping and associated components plus the fluid inventory, thereby reducing the environmental pollution.

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