Thermal performance analysis of Al_2O_3/R-134a nanorefrigerant

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ABSTRACT

Nowadays, nanofluids are being considered as an efficient heat transfer fluid in various thermal applications. Refrigerant-based nanofluids, termed as "nanorefrigerants", have the potential to improve the heat transfer performances of refrigeration and air-conditioning systems. This study analyzed the thermophysical properties and their effects on the coefficient of performance (COP) resulted by addition of 5 vol.% Al_2O_3 nanoparticles into R-134a refrigerant at temperatures of 283–308 K. The analysis has been done for a uniform mass flux through a horizontal smooth tube using established correlations. The results indicate that the thermal conductivity, dynamic viscosity, and density of Al_2O_3/R-134a nanorefrigerant increased about 28.58%, 13.68%, and 11%, respectively compared to the base refrigerant (R-134a) for the same temperature. On the other hand, specific heat of nanorefrigerant is slightly lower than that of R-134a. Moreover, Al_2O_3/R-134a nanorefrigerant shows the highest COP of 15%, 3.2%, and 2.6% for thermal conductivity, density, and specific heat, respectively compared to R-134a refrigerant. Therefore, application of nanoparticles in refrigeration and air-conditioning systems is promising to improve the performances of the systems.

1. Introduction

A nanorefrigerant is one kind of nanofluid for which the base fluid is a refrigerant. Like other nanofluids, it is a mixture of refrigerant and solid particles. Nanorefrigerant is being considered as a potential to enhance the thermal performance of refrigeration and air-conditioning systems because of the higher thermal conductivity of nanoparticles. Three main benefits have been reported when using nanorefrigerants in a refrigerator [1]; Firstly, use of nanoparticles can improve the solubility between the lubricant and the refrigerant [2]. Secondly, nanoparticles can enhance the thermal conductivity as well as heat transfer characteristics of a refrigerant [3,4]. Finally, reduction of the friction coefficient and wear rate is observed in nanorefrigerants compared to regular refrigerants [5]. It is hoped that, the addition of nanoparticles into conventional refrigerants will improve the heat transfer performance of refrigeration systems [6,7]. Almost all vapor compression refrigeration systems use lubricating oil making it possible to use nanoparticles in refrigeration systems in the form of a nanoparticles/oil suspension [8]. The existence of nanoparticles in oil suspension influences the overall performance of the refrigeration system since nanoparticles significantly enhance the thermophysical properties of the refrigerant. Consequently, energy consumption will be decreased along with reduction in emissions that lead to global warming and greenhouse-gas effects.

Thermophysical properties are the performance parameters that need to be analyzed in order to select the most suitable option for the energy conversion systems. Thermal conductivity is affected by temperature and density. High thermal conductivity of the refrigerant is crucial in order to gain the maximum output from the system [9]. Addition of nanoparticles with high thermal conductivity and increasing their concentration can enhance the thermal conductivity of a nanorefrigerant [10,11]. Viscosity is another property that affects the pumping power and pressure drop parameters [12]. It is known that pressure drop plays a significant role when designing and optimizing refrigeration systems [13]. Mahbubul et al. [14] studied the viscosity of R123-TiO_2 nanorefrigerant for different nanoparticle volume concentrations using Brinkman's model [15], and concluded that pressure drop increases significantly with the increase of viscosity. Moreover, rheological behavior of Al_2O_3/R141b nanorefrigerant was studied and the mixture was found to behave in a non-Newtonian way [16]. As like viscosity, density of a fluid also has influences on the pressure drop and pumping power capacity. A solid substance has a higher density in comparison to a liquid; therefore, the density of a nanofluid is found to be higher by increasing the concentration of nanoparticles within a fluid. Mahbubul et al. [17] measured the density of

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Nomenclature

A	heat transfer surface area (m^2)
bd	bubble departure diameter (m)
Bo	boiling number
Co	convection number
C_p	specific heat capacity (J/kg K)
COP	coefficient of performance
D	tube diameter (m)
E	enhancement factor
g	gravitational acceleration (m/s^2)
G	mass flux (kg/m^2 s)
h	heat transfer coefficient (W/m^2 K)
h_{fg}	latent heat (kJ/kg)
HTC	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
K	orifice constant
l	tube length (m)
L	temperature lift (K)
$\dot{m}$	mass flow rate (kg/s)
Nu	Nusselt number
P	pressure (Pa)
Pr	Prandtl number
q	heat flux (W/m^2)
Q_{out}	heat output (W)
r	radius of the tube (m)
r_p	radius of the nanoparticles (m)
Re	Reynolds number
S	suppression factor
t	thickness of interfacial layer (m)
T	temperature (K)
V	volumetric flow rate (m^3/s)
W_{net}	total work (W)
x	mass quality
X_{tt}	Martinelli parameter

Greek symbols

ϕ	particle volume concentration (%)
ρ	density ($kg/m^{3)}$
μ	dynamic viscosity (N s/m^2)
σ	surface tension (N/m)

Subscripts

cond	condenser
DB	pool boiling
down	downstream
evap	evaporator
in	input
l	interfacial layer/nanolayer
Npl	no pressure losses
nr	nanorefrigerant
p	nanoparticle
r	refrigerant
s	saturation
SA	single phase
up	upstream
v	vapor

Prefix

Δ	*gradient*

Al_2O_3/R141b nanorefrigerant and found that density increases linearly with increasing the volume concentration and decreases with increasing the temperature. Specific heat is a measure of energy storage capability of the working fluid. Fluids with large specific heat require significant amounts of energy input to sensibly increase or decrease their temperature. Specific heat is proportional to the change of internal energy of a system, thus when the temperature of the system increases, the fluctuation of molecules will be intensified and a higher heat capacity will be induced, as more energy levels can be filled up. This will be the reason of higher heat transfer rates. However, there is no literature available on the specific heat capacity of nanorefrigerants.

High COP and environmental friendliness are considered as the major selection criteria of a refrigerant. There are some studies available about the pool boiling [18], flow boiling [19], convective heat transfer [20,21], pressure drop [22,23], migration characteristics [24,25], and energy performance [1,6] of the nanorefrigerants. To the best of authors' knowledge, there are no studies available discussing the effect of thermophysical properties on the COP of a system using nanorefrigerants. The objective of this study is to investigate the effect of temperature on thermal conductivity, viscosity, density, and specific heat of Al_2O_3 nanoparticles suspended in R-134a refrigerant. Moreover, the effects of changed thermophysical properties of nanorefrigerant on the COP are investigated and compared with that of R-134a refrigerant.

2. Experimental method

The properties of Al_2O_3 nanoparticles and R-134a refrigerant are shown in Table 1 and Table 2, respectively. The analysis was carried out considering 5 vol.% of Al_2O_3 nanoparticles in R-134a refrigerant with the temperature range of 283–308 K. Thermophysical properties of Al_2O_3/R-134a nanorefrigerant were calculated using Microsoft Excel 2010 based on the established correlations from literature.

2.1. Thermal conductivity

Thermal conductivity of Al_2O_3/R-134a nanorefrigerant was predicted using the Sitprasert et al. [29] correlation. This model considers the effects of nanoparticle volume concentration, nanoparticle size, and temperature-dependent interfacial layer.

$$k_{nr} = \frac{(k_p - k_l)\phi k_l[2\beta_1^3 - \beta^3 + 1] + (k_p + 2k_l)\beta_1^3[\phi\beta^3(k_l - k_r) + k_r]}{\beta_1^3(k_p + 2k_l) - (k_p - k_l)\phi[\beta_1^3 + \beta^3 - 1]} \tag{1}$$

where,

$$\beta = 1 + \frac{t}{r_p} \tag{1a}$$

$$\beta_1 = 1 + \frac{t}{2r_p} \tag{1b}$$

Table 1
Properties of Al_2O_3 nanoparticles [26].

Properties	Value
Radius	15 nm
Molecular mass	101.00 kg/kmol
Density	3880 kg/m^3
Thermal conductivity	40* W/m K
Specific heat	729 J/kg K

* Source: Wang and Mujumdar [27].

Table 2
Properties of R-134a refrigerant [28].

Temperature (K)	Pressure (Mpa)	Liquid Density (kg/m^3)	Vapor density (kg/m^3)	Liquid Specific heat (kJ/kg k)	Liquid Thermal conductivity (W/m k)	Liquid Viscosity (mPa s)	Surface Tension (N/m)
283	0.41461	1261.0	20.226	1.3704	0.087618	0.23487	0.010138
288	0.48837	1243.4	23.758	1.3869	0.085444	0.22066	0.009441
293	0.57171	1225.3	27.780	1.4049	0.083284	0.20737	0.008756
298	0.66538	1206.7	32.350	1.4246	0.081134	0.19489	0.008081
303	0.77020	1187.5	37.535	1.4465	0.078992	0.18313	0.007417
308	0.88698	1167.5	43.416	1.4709	0.076853	0.17200	0.006766

The thickness and the thermal conductivity of the interfacial layer are calculated from Eqs. (1c) and (1d),

$$t = 0.01(T - 73)r_p^{0.35} \tag{1c}$$

$$k_l = C\frac{t}{r_p}k_r \tag{1d}$$

where $C = 30$ is a constant for Al_2O_3 nanoparticles.

COP is equal to heat output divided by total work input.

$$COP = \frac{Q_{out}}{W_{net,in}} \tag{2}$$

Eq. (2a) shows the basic equation used to calculate the heat transfer coefficient (HTC).

$$Q_{out} = hA\Delta T \tag{2a}$$

where, Q_{out} is the heat output, h is the HTC, A is the heat transfer area, and ΔT is the temperature difference.

Eq. (2b) introduces the relationship between force convective boiling heat transfer of pure refrigerant and the output heat, taken from Wen et al. [30]

$$h = Eh_{DB} + Sh_{SA} \tag{2b}$$

In this equation, E is the enhancement factor, S is the suppression factor, h_{DB} is the pool boiling HTC obtained from the correlation by Dittus and Boelter [31]. h_{SA} is the single phase heat transfer suggested by Stephan and Abdelsalam [32]. Eqs. (2c) – (2f) express all four parameters.

$$E = C_1 Bo^{C_2} X_{tt}^{C_3} \tag{2c}$$

$$h_{DB} = 0.023\frac{k}{D}\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{2d}$$

The Reynolds number was calculated using, $\mathrm{Re} = \frac{GD}{\mu}$ and Prandtl number by $\mathrm{Pr} = \frac{C_p\mu}{k}$. Here, mass flux,$G$, and tube diameter, D, were assumed to be 150 kg/m^2 sand 6 mm, respectively.

$$S = C_4 Co^{C_5} \tag{2e}$$

$$h_{SA} = 207\frac{k}{bd}\left[\frac{q(bd)}{kT_s}\right]^{0.674}\left(\frac{\rho_v}{\rho}\right)^{0.581}\mathrm{Pr}^{0.533} \tag{2f}$$

Referring to Eq. (2c), boiling number can be found from, $Bo = \frac{q}{h_{fg}G}$, while X_{tt} is the Martinelli parameter defined by $X_{tt} = \left[\frac{(1-x)}{x}\right]^{0.9}\left(\frac{\rho_v}{\rho}\right)^{0.5}\left(\frac{\mu}{\mu_v}\right)^{0.1}$. In Eqs. (2e) and (2f), Convection number, $Co = \left[\frac{(1-x)}{x}\right]^{0.8}\left(\frac{\rho_v}{\rho}\right)^{0.5}$ and Bubble departure diameter, $bd = 0.0146\,\alpha\left[\frac{2\sigma}{g(\rho-\rho_g)}\right]^{0.5}$ with $\alpha = 35^\circ$. T_s in Eq. (2f) is the saturation temperature corresponding to the test section pressure for the flow boiling. The constants C_1 to C_5 in Eqs. (2c) and (2e), were obtained by an iteration process to minimize the errors between the theoretical calculated HTC and experimental results [30]. The constant values are shown in Table 3.

Table 3
The constants in Eqs. (2c) and (2e) [30].

C_1	C_2	C_3	C_4	C_5
53.64	0.314	−0.839	0.927	0.319

Substituting the above equations, the final relationship between COP and thermal conductivity for refrigerant and nanorefrigerant are expressed in forms of Eqs. (3) and (4), respectively:

$$COP = \frac{\left\{E\left[0.023\frac{k_r}{D}\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}\right] + S\left\{207\frac{k_r^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}\left(\frac{\rho_v}{\rho}\right)^{0.581}\mathrm{Pr}^{0.533}\right\}\right\}A\,\Delta T}{W_{net,in}} \tag{3}$$

$$COP_{nr} = \frac{\left\{E\left[0.023\frac{k_{nr}}{D}\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}\right] + S\left\{207\frac{k_{nr}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}\left(\frac{\rho_v}{\rho}\right)^{0.581}\mathrm{Pr}^{0.533}\right\}\right\}A\,\Delta T}{W_{net,in}} \tag{4}$$

2.2. Viscosity

The viscosity of nanorefrigerant was calculated using Brinkman model [15] in the following form:

$$\mu_{nr} = \mu_r\frac{1}{(1-\phi)^{2.5}} \tag{5}$$

where, μ_{nr} and μ_r are the effective viscosity of nanorefrigerant and pure refrigerant, respectively. ϕ is the particle volume fraction which is 0.05 (5%) in our case.

Klein et al. [33] analyzed the impact of pressure drop on the refrigeration performance using liquid-suction heat exchanger. They proposed a dimensionless correlation that indicates COP in terms of pressure drop as follow:

$$\frac{COP}{COP_{npl}} = 1 - (2.37 - 0.0471L + 3.01 \times 10^{-4}L^2) \times \left(\frac{\Delta P}{P_{evap}}\right) \tag{6}$$

where, L is the temperature lift which equals to $T_{cond} - T_{evap}$ and P_{evap} is the evaporator pressure.

Pressure drop of refrigerant in the compressor was calculated using Hagen–Poiseuille equation [34]. In this analysis, refrigerant was assumed nearly incompressible. Hagen–Poiseuille model that is used to calculate the pressure drop for a fluid flow through a cylindrical tube, is expressed as follows:

$$\Delta P = \frac{8\mu l V}{\pi r^4} \tag{6a}$$

Replacing pressure drop in Eq. (6) by Eq. (6a) gives a new relationship between COP and viscosity:

$$\frac{COP}{COP_{npl}} = 1 - (2.37 - 0.0471L + 3.01 \times 10^{-4}L^2)\left(\frac{8\mu l V}{\pi P_{evap} r^4}\right) \tag{7}$$

2.3. Density

The density of nanofluid was calculated using Pak and Cho [35] correlation shown in Eq. (8):

$$\rho_{nr} = \phi\rho_p + (1-\phi)\rho_r \tag{8}$$

Performance of the system is dependent on the mass flow rate of the refrigerant. This was suggested by Bukac et al. [36] according to the following relation:

$$COP = \frac{\dot{m}\,\Delta h_E}{W_{net,in}} \tag{9}$$

where, Δh_E is the change of enthalpy in the evaporator.

The single-phase orifice equation [37] was used to calculate the mass flow rate through a short tube:

$$\dot{m} = KA\sqrt{2g\,\rho(P_{up} - P_{down})} \tag{9a}$$

To obtain the relationship between COP and density, the mass flow rate through a short tube in Eq. (9a) has been substituted into Eq. (9). The final equation is indicated as follows:

$$COP = \frac{\Delta h_E KA\sqrt{2g\,\rho(P_{up} - P_{down})}}{W_{net,in}} \tag{10}$$

2.4. Specific heat

For a given particle volume fraction, specific heat of a nanorefrigerant can be calculated using the correlation suggested by Pak and Cho [35].

$$C_{p,nr} = \phi C_{p,p} + (1-\phi)C_{p,r} \tag{11}$$

where, $C_{p,p}$ is the specific heat of Al_2O_3 nanoparticles and $C_{p,r}$ is the specific heat of R-134a refrigerant.

Substituting Prandtl numbers in Eqs. (3), (4) by expressions of specific heat, viscosity and thermal conductivity, the relationship between COP and specific heat for the refrigerant and nanorefrigerant can be obtained from Eqs. (12) and (13), respectively.

$$COP = \frac{\left\{E\left[\frac{0.023}{D}k_r^{0.6}\mathrm{Re}^{0.8}(C_{p,r}\mu_r)^{0.4}\right] + S\left\{\left[\frac{207}{bd}\right]\left(\frac{1}{k_r}\right)^{0.207}\left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_p}{\rho}\right)^{0.581}(C_{p,r}\mu_r)^{0.533}\right\}\right\}A\Delta T}{W_{net,in}} \tag{12}$$

$$COP_{nr} = \frac{\left\{E\left[\frac{0.023}{D}k_r^{0.6}\mathrm{Re}^{0.8}(C_{p,nr}\mu_r)^{0.4}\right] + S\left\{\left[\frac{207}{bd}\right]\left(\frac{1}{k_r}\right)^{0.207}\left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_p}{\rho}\right)^{0.581}(C_{p,nr}\mu_r)^{0.533}\right\}\right\}A\Delta T}{W_{net,in}} \tag{13}$$

3. Result and discussion

3.1. Thermal conductivity

Fig. 1 shows the variation of thermal conductivity of refrigerant and nanorefrigerant with temperature ranging from 283 to 308 K. It can be seen in Fig. 1 that, the thermal conductivity of Al_2O_3/R-134a nanorefrigerant was linearly increased with increasing temperature, while for pure refrigerant, thermal conductivity moderately decreased with increasing temperature. As the thermal conductivity of Al_2O_3 nanoparticle is higher than the base fluid (refrigerant) therefore, the thermal conductivity of the nanorefrigerant was found to be higher than pure refrigerant [3]. Again, with the rise of temperature, the Brownian motion of nanoparticles will intensify and the contribution of micro convection in heat transport will increase, which results in the augmentation of thermal conductivity [9]. Therefore, the thermal conductivity of nanorefrigerant tends to increase with increasing temperature. For pure refrigerant, thermal conductivity was decreased with increasing temperature. This is due to the fact that when temperature increases, the liquid is evaporated, which causes the atoms to be positively charged and vibrate with greater amplitude. This is why the thermal conductivity for any substance is lower at the vapor state compared with the liquid state. The increments in the thermal conductivity of Al_2O_3/R-134a nanorefrigerant are from 8.12% to 28.58% for 283 K to 308 K, respectively.

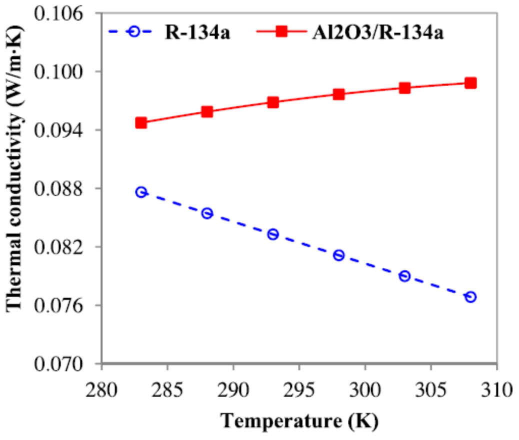

Fig. 1. Variation of thermal conductivity as a function of temperature.

Fig. 2 demonstrates the effect of thermal conductivity on COP of the refrigeration system at different temperatures for R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant. As noted from Fig. 2, COP increases with the increase of temperature (calculated using Eqs. (3) and (4), respectively). A maximum rise of 15% in COP is observed for the nanorefrigerant compared to that of the refrigerant due to its higher thermal conductivity. Since thermal conductivity is proportional to HTC, HTC of the nanorefrigerant with higher thermal conductivity is larger than that of the fluid with lower thermal conductivity at the same Nusselt number [3]. It can be stated that addition of more particles contributes to the

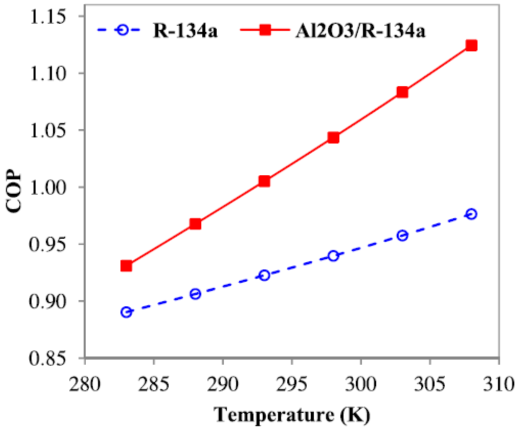

Fig. 2. Effect of the thermal conductivity of Al_2O_3/R-134a nanorefrigerant on COP at different temperatures.

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