ROLE OF THERMO-PHYSICAL PROPERTIES OF REFRIGERANT ON THE THERMAL PERFORMANCE OF A REFRIGERATION SYSTEM.

SA'ADAH BINTI AHMAD @ AHMAD SOWI

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ABSTRACT

The role of thermo properties of pure refrigerant and nanorefrigerant on refrigeration system performance are discussed in the report. The thermo-properties have, in fact, a very important part in designing the most optimum refrigeration system. R134a refrigerant is used in many refrigeration system including domestic refrigerators. Even though R134a refrigerant is widely used in refrigeration system, the research on finding another alternative refrigerant is widely conducted since the global warming potential of R134a is relatively high. Since then, there is rapid development of nanorefrigerant in exchange to the pure refrigerant. The addition of nanoparticles to the refrigerant give significant effect to the themo-performance of the refrigeration system, as it gives improvement in thermo-physical properties and heat transfer properties of the refrigerant. The nanorefrigerant that is used for this study is Al_2O_3/R-134a nanorefrigerant. The relationship between COP (coefficient of performance) and thermal properties of refrigerant R134a and Al_2O_3/R-134a nanorefrigerant are presented in this report. The correlations for each properties are obtained by combining some of the correlations from literature with basic thermodynamic approaches and assumptions. COP correlations are presented in four different equations according to its thermo-properties. COP correlations in term of viscosity is obtained by combining the dimensionless COP correlation from Klein et al. (2000) with Hagen–Poiseuille equation. COP correlations in terms of specific heat and thermal conductivity are obtained by substituting the derived equation from Wen and Ho (2005) into COP equation. COP correlations in terms of density is obtained by combining correlation from Payne and Neal (1995) and Bukac (2004). The Al_2O_3 particles concentration is fixed with 5 vol.% in order to simplify the calculations. Addition of nanoparticles to the refrigerant shows significant effect to the COP as the COP value for Al_2O_3/R-134a refrigerant is higher than that of R134a refrigerant. From the study, it is concluded that

COP of the refrigerant increases with increasing thermal conductivity, specific heat and density, and decreases with increasing viscosity. Therefore, it is proved that research on relationship between thermo-properties and COP is important in order to produce a refrigeration system than can operate at optimum operation (better cooling effect and higher system performance).

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LIST OF SYMBOLS AND ABBREVIATIONS

Nomenclature

$W_{net,in}$ net work input

ΔP pressure drop (Pa)

P_{evap} evaporation pressure (Pa)

μ dynamic viscosity refrigerant (Pa.s), (equivalent to N·s/m^2, or kg/(m·s))

l length of the tube (m)

V volumetric flow rate ($m^3 \cdot s^{-1}$)

r tube radius

r_p radius of nanoparticles (m)

COP coefficient of performance

T temperature (K)

T_s saturation temperature corresponding the test section pressure for the flow boiling (K)

L Temperature lift, $T_{cond} - T_{evap}$ (K)

h heat transfer coefficient (W/(m^2K))

A heat transfer surface area (m^2)

Q_{out} heat output (W)

E enhancement factor

B_0 boiling number

G mass flux (kg/m²s)

h_{fg} latent heat (kJ/kg)

X_{tt} Martinelli parameter

C_0 convection number

x mass quality

h_{DB} pool boiling heat transfer coefficient

h_{SA} single phase heat transfer

Δh_E change in the enthalphy in the evaporator (Joule)

g gravitational acceleration (m/s²)

q heat flux (kW/m²)

Pr prandlt number ($\mu C_p/k$)

Re reynolds number (GD/μ)

$\dot{m}$ mass flowrate (kg/s)

k thermal conductivity (kW/m K)

K orifice constant

A orifice area (m²)

t thickness of interfacial layer (m)

C constant of of Al_2O_3 nanoparticles, (C=30)

c_p specific heat (

D tube diameter

Greek Letters

$\emptyset$ particle volume fraction

μ dynamic viscosity refrigerant (Pa.s), (equivalent to N·s/m^2, or kg/(m·s))

ρ density (kg/m^3)

σ surface tension (N/m)

Subscripts

r pure refrigerant

nr nanorefrigerant

p particles

l interfacial layer/ nanolayer

crit critical

com compressor

cond condenser

evap evaporator

LIST OF APPENDICES

1.0 INTRODUCTION

Nowadays, refrigeration seems to be very important for human daily needs. These cover a wide range of applications such as domestic cooling, food processing, industrial air conditioning system etc. Refrigerator is a device that produces refrigeration system, which can transfer heat from a lower temperature medium to higher temperature medium. Refrigerant is the working fluid used in refrigeration. Therefore, it is important to have accurate knowledge of the thermo-physical properties of refrigerant in order to evaluate the performance of refrigeration cycles and also to determine the best possible alternative refrigerant.

Thermo-physical properties are the fundamental properties that need to be investigated in order to generate the optimum result from refrigerants, to select the top suited refrigerant for the set energy conversion systems. In this report, the thermo-physical properties that need to be considered are thermal conductivity, specific heat, viscosity and density. Thermal conductivity is a measure of the heat conducting capability of the working fluid. Thermal conductivity is affected by temperature and density. Fluids with high thermal conductivity are effective at transferring energy with small temperature differences. And also, thermal conductivity increase as density decrease. High thermal conductivity of the refrigerant is crucial in order to gain maximum output from the system (Mahbubul et al., 2013), making it possible to the system to achieve both maximum energy efficiency and high heat transfer coefficient, that finally result to increase system performance.

Specific heat is a measure of the energy storing capability of the working fluid . Fluids with large specific heats 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, so when the temperature of the system increase, the molecules of the liquid will fluctuates and caused higher heat capacity, as more of its energy levels can be filled up. When higher energy level can be easily accessed into the system, the heat transfer rate will be increased as well. A parameter that relates specific heat, thermal conductivity and fluid viscosity is known as prandtl number. Through this parameter, the relationship between COP and specific heat is obtained. Viscosity is the property that relates to pumping as pumping power and pressure drop parameters. It is known that pressure drop play a significant role in order to create the most accurate design and optimization of a refrigeration system (Didi et al., 2002) . Thermal conductivity, density and viscosity value of pure refrigerant obtained from Krauss et al. (1993) while specific heat is calculated using the basic thermodynamics approach.

1.1 Problem Statement

Engineers nowadays find several solutions in order to improve refrigerant performance including adding heat transfer devices and use nanorefrigerant as a replacement of pure refrigerant. Even though all these methods are able to improve the performance of refrigeration system, the basic way to ensure that the system will operate at its optimum operation is by choosing the best refrigerant possible. In order to find the most suitable refrigerant for the system, the study of the relationship between thermo-physical properties of refrigerant and thermal performance of refrigeration system is needed. Propainop and Suen (2012) have studied the effect of thermo-physical properties of refrigerant on system performance. The previous studies just study the effect based on several assumptions and theories. But, there is no formal correlations that relates the COP of the refrigerant with thermo-physical properties of refrigerant.

1.2 Research Objective

The objectives of the study are to establish the relationship between COP with thermo-physical properties of refrigerant, to calculate performance improvement of refrigerant and to compare the system performance of pure refrigerant and nanorefrigerant. The thermo-properties of refrigerant that are considered in the calculations are thermal conductivity, viscosity, specific heat and density. Nano particle that is studied is Al_2O_3 (Aluminum Oxide) nanoparticle with 5% volume concentration. Mahbubul et al. (2013) has conducted a research on thermo-physical properties and heat transfer performance of Al_2O_3/R-134a nanorefrigerant. The correlations for nanorefrigerant properties are discussed in methodology section. Previous study shows that thermo-physical properties of nanorefrigerant can be vary by varying the particle volume fraction, nanoparticles type, refrigerants, particle sizes, particle shapes, and temperature (Brown et al., 2002). Since this research is focuses on computing the relation between COP and thermo-properties of refrigerant, the particles volume concentration is fixed to 5% volume fraction (in order to make sure the calculation is less tedious without so many parameters), and the thermo-properties values are obtained by varying temperature.

2.0 LITERATURE REVIEW

Among all type of refrigerants, R134a (1,1,1,2-tetrafluoroethane) is considered to be environmental friendly and very commonly used. Due to the advantage, research on its thermo-physical properties and transport properties are widely conducted. The correlations of transport properties of R-134a refrigerant were published by some authors (Stephen and Krauss, 1992). Since the previous author did not take into consideration the critical enhancement of the transport coefficient, several papers that show the improved correlations of thermo-physical properties has been published. Krauss et al. (1993) came out with new correlations equation of thermal conductivity and viscosity based on residual concept. The paper presented the new equations for the two properties that is valid in a wide range of pressures and temperatures.

Tsvetkov et al. (1995) calculated the thermal conductivity of R134a refrigerant by using means of the coaxial cylinder method at steady state and the transient coaxial cylinder apparatus. The authors verified their result by comparing it with some theoretical and empirical correlations. Generally, higher thermal conductivity give rise to higher heat transfer coefficient. Payne and Neal (1995) propose a way on how to theoretically relate refrigerant density with mass flow rate. The authors generalized the correlation of refrigerant mass flow through a short tube. The correlations are useful in system design and system performance calculation. In predicting the capacity and coefficient of system performance of a compressor, Hubert Bukac (2004) has come out with a complex mathematical model. The model that is proposed by the author consisted of four major parts including the flow through the suction valve and the discharge valve, leakage of gas between the cylinder and piston and also model of electric motor torque.

Higher values of refrigerant viscosity in both refrigerant and compressor will cause higher pressure drop (Domanski and Didion, 1987). Due to higher pressure drop, the mass flow rate of the refrigerant will be decreased because of the decreasing in pressure value at the compressor inlet and increasing in pressure value at the compressor outlet. Lower mass flow rate caused lower capacity that result in decreasing compressor work. Klein et al. (2000) distinguished their works from previous researcher by proposing a new idea to relate the coefficient of performance of refrigerant system with pressure drop. In the paper, the author presented the net effect on COP in terms of the non-dimensional pressure loss and temperature lift.

Wen and Ho (2005) investigate the heat-transfer and pressure drop behaviour of refrigerant using three-lines serpentine small diameter tube bank. This study showed that refrigerant with higher thermal conductivity has higher heat transfer coefficient. The paper presented that heat transfer of R-290, R-600, and R-600/R290 is higher compared to R-134a since these refrigerant has higher thermal conductivity. Prapainop and Suen (2012) have reviewed the effect of relevant properties/parameters toward refrigerant performance. Refrigerant properties that are considered including density, viscosity, thermal conductivity, molar heat capacity, critical temperatures, specific refrigerating effect and molecular weight while derived parameters that are considered are volumetric refrigerating capacity, heat transfer coefficient and pressure drop. Effect of thermo-physical properties and derived parameters toward the refrigeration performance is tabulated in Table B.1. Generally, lower vapour density will result in a lower refrigerant capacity since it caused larger pressure drop in the evaporator and condenser.

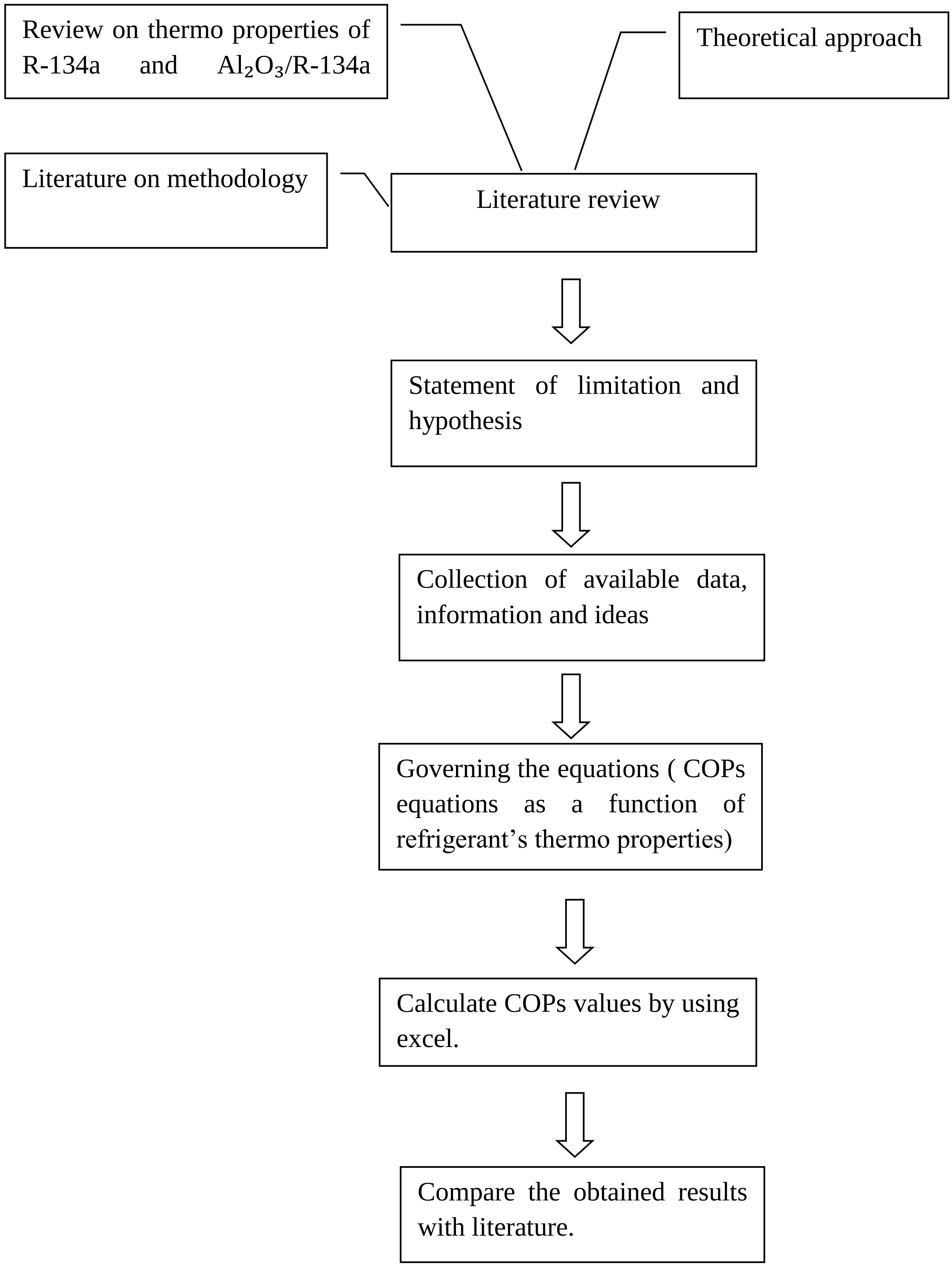
Since the past ten years, study showed that R-22a is widely used refrigeration system, but due to its low in system performance, researchers try to find a way in order to cut off its usage. There are several alternative refrigerants that have been introduced for instance Kurokawa et al. (1996) have conducted an experiment to investigated system performance of R-410a refrigerant. Thermo-properties of the refrigerants are measured and realistic models are formulated. Refrigeration R134a has been introduced and commonly used due to its environmental friendly and high thermo performance. Brown et al. (2002) compare the performance of CO_2 and R13a refrigerant that operate in the automotive air conditioning system using semi theoretical cycle models. The authors used two types of system; refrigeration system and CO_2 system which equipped with a liquid-line/suction line heat exchanger. The study compare the COPs of R134a and CO_2 systems using NIST's semi theoretical vapour compression model, CYCLE 11. UA and the transcritical cycle model, CYCLE-11.UA- CO_2 [27]. Based on the simulation, result showed that R134a gives better COP compared to CO_2. Figure 13 on appendix D shows the diagram of thermal conductivity measuring experimental setup of CuO nanofluids.

Scientists have come out with several attempts in the past ten years in order to create a better alternative refrigerant. The major criteria of refrigerant that is well considered are environmental friendly and high COP. From literature, there are several authors introduce refrigerant mixtures as well as nanorefrigerant as the alternative refrigerants. As the development of these types of refrigerants are rapidly growing in recent years, research on their thermo-physical properties need to be widely conducted in order to create the most optimum refrigeration system. Mahbubul et al. (2013) has introduced the mathematical model of thermo-physical properties and heat transfer performance of Al_2O_3/R-134a nanorefrigerant.

In summarising the literature review, Payne and Neal (1995) proved that increase pressure drop on both evaporators and condensers will decrease the mass flow rate of refrigerant. Domanski and Didion (1987) concluded that higher value of viscosity will result to higher pressure drop. Wen and Ho (2005) concluded that increase in heat transfer coefficient of R-134a will increase the mass flux and as the mass flow rate increase, the pressure drop of R-134a will increase. Prapainop and Suen (2013) concluded that high value of latent heat, liquid thermal conductivity and vapour density and low values of liquid viscosities and molecular weight will result to higher COP. The study also concluded that the change in heat transfer and pressure drop will effect system performance.

3.0 METHODOLOGY

3. 1 Flow chart



The correlations of COP and thermal properties of refrigerant R134a and Al_2O_3/R-134a nanorefrigerant are obtained by combining some of the correlation from literature and using some of the basics thermodynamics approaches and assumptions. The properties of Al_2O_3 and R134a refrigerant has shown in Table 3.2. The refrigerant's phase and temperature is assumed to be different for each thermal properties calculation since the thermo-physical data is obtained from different literature with different assumptions. Parameters that are used throughout the calculations are tabulated in Table B.8. All the calculations of thermo-physical properties and COPs are conducted by using Microsoft Excel 2013.

3.2 Pure Refrigerant

3.1.1 Viscosity

Pressure drop of refrigerant in the compressor is calculated using Hagen–Poiseuille equation since the assumption that is made for this calculation is the refrigerant is nearly incompressible. Hagen-Poisuille equation is a mathematical model to calculate pressure drop in a fluid flowing through a cylindrical pipe, the equation is expressed as,

$$\Delta P = \frac{8\mu_r lV}{\pi r^4} \tag{3.1}$$

where μ_r is the dynamic viscosity of refrigerant, l is tube length, V is the volumetric flow rate and r is the tube radius. Pressure drop is the reduction in air pressure from the compressor discharge until the actual point of use.

(a) COP and pressure drop

Klein et al. (2000) analyse the impact of pressure drop towards refrigeration performance using liquid-suction heat exchanger. The paper introduce new dimensionless correlation of COP by comparing the COP of refrigerant with and without pressure losses. The equation is valid by assuming that there is no flashing ahead of the valve so that the pressure drop is sufficiently small at that particular area but higher in the vapour (low pressure) leg of the liquid suction heat-exchanger. The dimensionless correlation that relates COP and pressure drop is expressed in Equation (3.2),

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481L + 3.01 \times 10^{-4}L^2)\left(\frac{\Delta P}{P_{evap}}\right) \qquad (3.2)$$

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

(b) COP and viscosity

Equation (2) then be modified by substituting pressure drop from hagen -poiseuille equation into Equation (2), the new set of correlation that shows relationship between viscosity and COP is given by Equation 3.3

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481L + 3.01 \times 10^{-4}L^2)\left(\frac{8\mu_r lV}{\pi P_{evap} r^4}\right) \qquad (3.3)$$

Equation (3.3) represents the dimensionless correlation of COP in terms of refrigerant viscosity.

3.1.2 Thermal Conductivity

(a) Heat transfer coefficient

Equation (3.4) shows the basic calculation in calculating the heat transfer coefficient. Heat transfer coefficient is basically how much heat is transfer by convection between two-phase region (solid and liquid).

$$h = \frac{Q_{out}}{A.\Delta T} \tag{3.4}$$

where Q_{out} is the heat output, A is the heat transfer surface area and ΔT is the temperature different. Equation (3.5) introduce the relationship between heat transfer for force convective boiling of pure refrigerant with heat output, derived from Wen and Ho (2005). The heat transfer coefficient correlation is expressed by using superposition model including both enhancement factor and suppression factor.

$$Q = (Eh_{DB} + Sh_{SA})A.\Delta T \tag{3.5}$$

where E is the enhancement factor, S is the suppression factor, h_{DB} is the pool boiling heat transfer coefficient that is obtained from correlation by Dittus and Boelter (1930) and h_{SA} is the single phase heat transfer that obtained from correlation by Stephan and Abdelsalam (1980) . All four parameters values are expressed by,

$$E = C_1 {B_o}^{C_2} {X_{tt}}^{C_3} \tag{3.6}$$

$$S = C_4 {C_o}^{C_5} \tag{3.7}$$

$$h_{DB} = 0.023 \frac{k}{D} Re^{0.8} Pr^{0.4} \tag{3.8}$$

$$h_{SA} = 207 \frac{k}{(bd)} \left[\frac{q(bd)}{kT_s}\right]^{0.674} \left(\frac{\rho_V}{\rho}\right)^{0.581} Pr^{0.533} \tag{3.9}$$

Referring to Equations (3.6) and (3.7), B_o is the boiling number that can be found from $B_0 = \frac{q}{h_{fg}G}$ while X_{tt} is the Martinelli parameter defined as, $\left[\frac{1-x}{x}\right]^{0.9}\left(\frac{\rho_v}{\rho}\right)^{0.5}\left(\frac{\mu}{\mu_v}\right)^{0.1}$. In Equations (3.8) and (3.9), C_0 which is the convection number can be calculated using equation $\left[\frac{1-x}{x}\right]^{0.8}\left(\frac{\rho_v}{\rho}\right)^{0.5}$while bd is equals to $0.0146\alpha\left[\frac{2\sigma}{g(\rho-\rho_g)}\right]^{0.5}$ with $\alpha = 35^0$. T_s in Equation (3.9) is the saturation temperature corresponding to the test section pressure for the flow boiling.

The $C_1$to C_5 constant in Equations (3.6) and (3.7) are obtained by an iteration process to minimize the errors between the theoretical calculated heat transfer coefficient and experimental results. (Wen and Ho, 2005). The $C_1$to C_5 constant value is shown in Table 3.1.

(b) Relationship between COP and thermal conductivity

Coefficient of performance is basically equals to heat output divided by total work input.

$$COP = \frac{Q_{out}}{W_{net,in}} \tag{3.10}$$

Substituting Equation (3.5) into Equation (3.10), the final equation goes as follow,

$$COP = \frac{\left\{E\left[0.023\frac{k_r}{D}Re^{0.8}Pr^{0.4}\right] + S\left\{207\frac{{k_r}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}\left(\frac{\rho_v}{\rho}\right)^{0.581}Pr^{0.533}\right\}\right\}A \times \Delta T}{W_{net,in}} \tag{3.11}$$

where k_r is the refrigerant thermal conductivity, D is the tube diameter, Pr is the prandtl number, T_s is the saturation temperature corresponding the test section pressure for the flow boiling, ρ_v is the vapor density and ρ is the liquid density. By substituting the superposition model of heat transfer coefficient into COP equation, the new equation that relates COP with thermal conductivity is obtained.

Table 3.1 : The constants in Equations (3.6) and (3.7)

C1	C2	C3	C4	C5
53.64	0.314	-0.839	0.927	0.319

Source : Wen and Ho (2005)

3. 1.3 Specific heat

Prandlt Number, Pr

Heat transfer introduce a process known as convective heat transfer. It is a process when the heat is transfer by the movement of fluid. In analysing convective heat transfer, a dimensionless parameter known as prandtl number was introduced. The parameter that relates heat capacity, specific heat and fluid viscosity is shown in Equation (3.12),

$$Pr = \frac{C_p \mu}{k} \tag{3.12}$$

Since refrigeration system is the process where heat is transfer through refrigerant movement, so prandtl number is valid for the calculation.

Relationship between COP and specific heat

To relate COP with specific heat, prandlt number in Equation (3.12) is written in terms of specific heat, viscosity and thermal conductivity. Equation (3.12) is substitute into Equation (3.11) and another set of correlation that relates both COP and specific heat is,

$$COP = \frac{\left\{E\left[\frac{0.023}{D} {k_r}^{0.6} Re^{0.8} ({c_p}_r \mu)^{0.4}\right] + S\left\{\left[\frac{207}{(bd)}\right]\left(\frac{1}{k_r}\right)^{0.207}\left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_v}{\rho}\right)^{0.581}({c_p}_r \mu)^{0.533}\right\}\right\} A \times \Delta T}{W_{net,in}} \tag{3.13}$$

3.1.4 Density

(a) Mass flow rate

Short tubes are used in vapour compression cycles to regulate and meter refrigerant flow. The short tube is a constant area restriction with a sharp edge or a chamfered entrance. Inlet chamfering has a dramatic effect upon mass flow rate (Aaron and Domanski 1990, Kim 1993). Mass flow rate through a short tube can be modelled by the single-phase orifice equation:

$$\dot{m} = KA\sqrt{2g\rho(p_{up} - p_{down})} \tag{3.14}$$

where K is the orifice constant, A is the surface area, g is the gravitational acceleration, p_{up} is the upstream pressure and p_{down} is the downstream pressure. Payne and Neal (1981) has generalized a correlation for refrigerant mass flow through a short tube. The objective of their paper is to predict the mass flow rate of several refrigerants in refrigeration system.

<u>Relationship between COP and density</u>

Reduction in mass flow rate of refrigerant caused reduction in system capacity. It is proved by Hubert Bukac (2005),

$$COP = \frac{Capacity}{W_{net,in}} = \frac{\dot{m}\Delta h_E}{W_{net,in}} \tag{3.15}$$

where Δh_E is the change in the enthalpy in the evaporator. Capacity is basically the product of refrigerant flow rate and the change of enthalpy of the refrigerant. To obtain relationship between COP and density, the mass flow rate through a short tube in Equation (3.14) is substituted into COP equation of Equation (3.15),

$$COP = \frac{\Delta h_E K A \sqrt{2 g \rho_r (p_{up} - p_{down})}}{W_{net,in}} \tag{3.16}$$

3.3 Nanorefrigerant

Nano particles that is investigated in this study is Al_2O_3 particles with volume fraction of 0.05. The Al_2O_3 particles properties and its constant parameter is referring to Mahbubul et al. (2013).

Table 3.2 : Properties of Al_2O_3 nanoparticles and R-134a refrigerant

	Molecular mass (kg/kmol)	Density (kg/m³)	Thermal Conductivity (W/mK)	Specific heat (J/kg K)	Viscosity (N s/m²)
Al_2O_3 (d_p= 30 nm)	101	3880	40	729	-
R-134a refrigerant (T=300K)	102.03	1199.7	0.0803	1432	0.00019

Source : Mahbubul et al. (2013)

3.3.1 Viscosity

The viscosity of nanorefrigerant was calculated using Brinkman model and the equation is,

$$\mu_{nr} = \mu_r \frac{1}{(1 - \emptyset)^{2.5}} \tag{3.17}$$

where $\emptyset$ is the particle volume fraction which is 0.05 (5%). The relationship between COP and viscosity is referring to Equation (3.3), and the new COP equation for nanorefrigerant is,

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481L + 3.01 \times 10^{-4} L^2)\left(\frac{8\mu_{nr} l V}{\pi P_{evap} r^4}\right) \tag{3.18}$$

3.3.2 Thermal Conductivity

In determining thermal conductivity of Al_2O_3/R-134a nanorefrigerant, Mahbubul et al. (2013) used the method proposed by Sitprasert et al. (2009). The thermal conductivity for nanorefrigerant is calculated while taking into consideration the effects of particle volume fraction, particle size temperature and the interfacial layer of nanoparticles. The equation is,

$$k_{nr} = \frac{(k_p - k_l)\emptyset k_l[2{\beta_1}^3 - \beta^3 + 1] + (k_p + 2k_l){\beta_1}^3[\emptyset\beta^3(k_l - k_r) + k_r]}{{\beta_1}^3(k_p + 2k_l) - (k_p - k_l)\emptyset[{\beta_1}^3 + \beta^3 - 1]} \tag{3.19}$$

where k_{nr}, k_p are thermal conductivity of nanorefrigerant and solid particles, t is thickness of interfacial layer, C is the constant of of Al_2O_3 nanoparticles which is equals to 30. β, β_1, t, k_l are calculated using the following equations;

$$\beta = 1 + \frac{t}{r_p}, \beta_1 = 1 + \frac{t}{2r_p} \tag{3.20}$$

$$t = 0.01(T - 73){r_p}^{0.35} \tag{3.21}$$

$$k_l = C\frac{t}{r_p}k_r \tag{3.22}$$

The relationship between COP and thermal conductivity is referring to Equation (3.11), and the new COP equation for nanorefrigerant is,

$$COP = \frac{\left\{E\left[0.023\frac{k_{nr}}{D}Re^{0.8}Pr^{0.4}\right] + S\left\{207\frac{{k_{nr}}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}\left(\frac{\rho_v}{\rho}\right)^{0.581}Pr^{0.533}\right\}\right\}A \times \Delta T}{W_{net,in}} \tag{3.23}$$

3.3.3 Specific Heat

For a given particle volume fraction, specific heat for nanorefrigerant can be calculated using correlation from Pak and Cho (1998),

$$C_{p,nr} = \emptyset C_{p,p} + (1-\emptyset) C_{p,r} \tag{3.24}$$

where $C_{p,p}$ is the specific heat of Al_2O_3 nanoparticles and $C_{p,r}$ is the specific heat of R134a-refrigerant. The specific heat values for the base fluid which is in this study is R134a are taken from refrigerant property database (REFPROP 7.0) that is developed by the National Institute of Standard and Technology (NIST).

(a) Relationship between COP and specific heat

The relationship of COP and specific heat of nanorefrigerant is obtained by rearranging Equation (3.9). New set of equation containing $C_{p,nr}$ is,

$$COP = \frac{\left\{E\left[\frac{0.023}{D}{k_r}^{0.6} Re^{0.8} (c_{p_{nr}}\mu)^{0.4}\right] + S\left\{\left[\frac{207}{(bd)}\right]\left(\frac{1}{k_r}\right)^{0.207}\left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_v}{\rho}\right)^{0.581}(c_{p_{nr}}\mu)^{0.533}\right\}\right\} A \times \Delta T}{\dot{W}_{net,in}} \tag{3.25}$$

3.3.4 Density

The density of nanofluid is calculated by using density correlation shown in Equation (3.26) developed by Pak and Cho (1998)

$$\rho_{nr} = \emptyset \rho_p + (1-\emptyset)\rho_r \qquad 3.26$$

(b) Relationship between COP and density

The relationship of COP and density of nanorefrigerant was obtained by rearranging Equation (3.16). New set of equation containing ρ_{nr} is,

$$COP = \frac{\Delta h_E KA\sqrt{2g\rho_{nr}(p_{up} - p_{down})}}{W_{net,in}} \qquad 3.27$$

3.4 Sample Calculation

3.4.1 Viscosity

Below is the sample calculation to obtain COP values for R134a refrigerant at viscosity of 98.28 µ Pa.s, 188.3 µ Pa.s, 251.1µ Pa.s with constant pressure of 18 MPa.

a) *Viscosity equals to 98.28 µ Pa.s*

From Equation (3.3),

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481\ (140) + 3.01 \times 10^{-4}(140)^2)\left(\frac{8(98.28 \times 10^{-6}\mathrm{Pa.s})(4m)(0.00036\ \mathrm{m}^3)}{\pi(51.25Pa)(0.001)^4}\right) \qquad (3.3)$$

Where L= 140, μ_r=251.1 µPa.s, l= 4m, V= 0.00036 m³, P_{evap}= 51.25 Pa and r = 0.001m

$$\frac{\mathbf{COP}}{\mathbf{COP_{no\ pressure\ losses}}} = \mathbf{0.978}$$

b) *Viscosity equals to 188.3 µ Pa.s*

From Equation (3.3),

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481\ (140) + 3.01 \times 10^{-4}(140)^2)\left(\frac{8(188.3 \times 10^{-6}\mathrm{Pa.s})(4m)(0.00036\ \mathrm{m}^3)}{\pi(51.25Pa)(0.001)^4}\right) \qquad (3.3)$$

Where L= 140, μ_r=251.1 µPa.s, l= 4m, V= 0.00036 m³, P_{evap}= 51.25 Pa and r = 0.001m

$$\frac{\mathbf{COP}}{\mathbf{COP_{no\ pressure\ losses}}} = \mathbf{0.956}$$

c) *Viscosity equals to 251.1 μ Pa.s*

$$\frac{COP}{COP_{no\ pressure\ losses}} = 1 - (2.37 - 0.0481\,(140) + 3.01 \times 10^{-4}(140)^2)\left(\frac{8(251.1 \times 10^{-6}\text{Pa.s})(4m)(0.00036\ \text{m}^3)}{\pi(51.25Pa)(0.001)^4}\right) \tag{3.3}$$

Where L= 140, μ_r=251.1 μPa.s, l= 4m, V= 0.00036 m³, P_{evap}= 51.25 Pa and r = 0.001m

$$\frac{\mathbf{COP}}{\mathbf{COP_{no\ pressure\ losses}}} = \mathbf{0.944}$$

From the sample calculation above, it is proved that increase in refrigerant viscosity will decrease COP.

3.4.2 Thermal Conductivity

Below is the sample calculation to obtain COP values for R134a refrigerant at thermal conductivity of 82.67 kW/m K, 93.15 kW/m K and 104 kW/m K with constant pressure of 18 MPa.

a) *Thermal conductivity equals to 82.67 kW/m K*

From equation (3.11),

$$COP = \frac{\{E[H] + S\{I \times J\}\}A \times \Delta T}{W_{net,in}} \tag{3.11}$$

Where, $H = 0.023\frac{k_r}{D}Re^{0.8}Pr^{0.4}$, $I = 207\frac{{k_r}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}$ and $J = \left(\frac{\rho_v}{\rho}\right)^{0.581}Pr^{0.533}$

$$COP = \frac{\{8.497 \times 70.81 + 0.638\{1427 \times 5.49 \times 10^{-6}\}\}3.14 \times 10^{-6} \times 140}{1.81kW}$$

Where, A = 3.14×10^{-6}m², ΔT =140 and $W_{net,in}$= 1.81kW.

$\mathbf{COP = 0.146}$

b) *Thermal conductivity equals to 93.15 kW/m K*

$$COP = \frac{\{E[H] + S\{I \times J\}\}A \times \Delta T}{W_{net,in}} \quad (3.11)$$

Where, $H = 0.023\frac{k_r}{D}Re^{0.8}Pr^{0.4}$, $I = 207\frac{{k_r}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}$ and $J = \left(\frac{\rho_v}{\rho}\right)^{0.581} Pr^{0.533}$

$$COP = \frac{\{8.497 \times 76.057 + 0.638\{1392.34 \times 5.49 \times 10^{-6}\}\}3.14 \times 10^{-6} \times 140}{1.81kW}$$

Where, A = 3.14×10^{-6}m², ΔT =140 and $W_{net,in}$= 1.81kW.

$\mathbf{COP = 0.157}$

c) *Thermal conductivity equals to 104 kW/m K*

Where, $H = 0.023\frac{k_r}{D}Re^{0.8}Pr^{0.4}$, $I = 207\frac{{k_r}^{0.326}}{(bd)}\left[\frac{q(bd)}{T_s}\right]^{0.67}$ and $J = \left(\frac{\rho_v}{\rho}\right)^{0.581} Pr^{0.533}$

$$COP = \frac{\{8.497 \times 81.25 + 0.638\{1360.34 \times 5.49 \times 10^{-6}\}\}3.14 \times 10^{-6} \times 140}{1.81kW}$$

Where, A = 3.14×10^{-6}m², ΔT =140 and $W_{net,in}$= 1.81kW.

$\mathbf{COP = 0.168}$

From the sample calculation above, it is proved that increase in refrigerant thermal conductivity will increase COP.

3.4.3 Specific Heat

Below is the sample calculation to obtain COP values for R134a refrigerant at specific heat 1288.3 kJ/kg.K, 1310 kJ/kg.K and 1334 kJ/kg.K with increasing temperature and pressure

From Equation (3.13),

a) *Specific Heat equals to 1288.3 kJ/kg.K*

$$COP = \frac{\{E[U] + S\{X \times Y \times Z\}\}A \times \Delta T}{\dot{W}_{net,in}} \tag{3.13}$$

Where, $U = \frac{0.023}{D} {k_r}^{0.6} Re^{0.8} ({c_p}_{nr}\mu)^{0.4}$, X=$\left[\frac{207}{(bd)}\right]\left(\frac{1}{k_r}\right)^{0.207}$

$Y = \left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_v}{\rho}\right)^{0.581}$ and $Z = ({c_p}_{nr}\mu)^{0.533}$

$$COP = \frac{\{(8.497 \times 355) + 0.638\{15154176 \times 3.82 \times 10^{-6} \times 0.473\}\}3.14 \times 10^{-6} \times 140}{1.81\ kW}$$

$\boldsymbol{COP}$ **=0.737**

b) *Specific Heat equals to 1310 kJ/kg.K*

$$COP = \frac{\{E[U] + S\{X \times Y \times Z\}\}A \times \Delta T}{\dot{W}_{net,in}} \tag{3.13}$$

Where, $U = \frac{0.023}{D} {k_r}^{0.6} Re^{0.8} ({c_p}_{nr}\mu)^{0.4}$, X=$\left[\frac{207}{(bd)}\right]\left(\frac{1}{k_r}\right)^{0.207}$

$Y = \left[\frac{q(bd)}{T_s}\right]^{0.674}\left(\frac{\rho_v}{\rho}\right)^{0.581}$ and $Z = ({c_p}_{nr}\mu)^{0.533}$

$$COP = \frac{\{(8.497 \times 357) + 0.638\{15154176 \times 3.82 \times 10^{-6} \times 0.477\}\}3.14 \times 10^{-6} \times 140}{1.81\ kW}$$

$\boldsymbol{COP}$ **=0.747625**

c) *Specific Heat equals to 1334 kJ/kg.K*

$$COP = \frac{\{E[U] + S\{X \times Y \times Z\}\}A \times \Delta T}{\dot{W}_{net,in}} \tag{3.13}$$

Where, $U = \frac{0.023}{D} {k_r}^{0.6} Re^{0.8} ({c_p}_{nr}\mu)^{0.4}$, X=$\left[\frac{207}{(bd)}\right]\left(\frac{1}{k_r}\right)^{0.207}$

$Y = \left[\frac{q(bd)}{T_s}\right]^{0.674} \left(\frac{\rho_v}{\rho}\right)^{0.581}$ and $Z = ({c_p}_{nr}\mu)^{0.533}$

$$COP = \frac{\{(8.497 \times 360) + 0.638\{15154176 \times 3.82 \times 10^{-6} \times 0.482\}\}3.14 \times 10^{-6} \times 140}{1.81\ kW}$$

$\boldsymbol{COP}$ **=0.747625**

From the sample calculation above, it is proved that increase in specific heat of pure refrigerant will increase COP.

3.4.4 Density

Below is the sample calculation to obtain COP values for R134a refrigerant at density 1069 kg/m³, 1199 kg/m³and 1304 kg/m³.

From Equation (3.16)

a) *Density equals to 1069 kg/m³*

$$COP = \frac{143.69 \times 0.62 \times 0.000032\sqrt{2 \times 9.81 \times 1069(1176 - 427)}}{1.81}$$

Where $\Delta h_E = 143.69\, Joule$, K= 0.62, A= 0.000032m², ρ_r= 1199 kg/m³, p_{up}= 1176 Pa and p_{down}= 427 Pa

$\boldsymbol{COP = 11.3}$

b) *Density equals to 1199 kg/m³*

$$COP = \frac{143.69 \times 0.62 \times 0.000032\sqrt{2 \times 9.81 \times 1199(1176 - 427)}}{1.81}$$

Where $\Delta h_E = 143.69\, Joule$, K= 0.62, A= 0.000032m², ρ_r= 1199 kg/m³, p_{up}= 1176 Pa and p_{down}= 427 Pa

$\boldsymbol{COP = 11.97}$

c) *Density equals to 1304 kg/m³*

$$COP = \frac{143.69 \times 0.62 \times 0.000032\sqrt{2 \times 9.81 \times 1304(1176 - 427)}}{1.81}$$

Where $\Delta h_E = 143.69\, Joule$, K= 0.62, A= 0.000032m², ρ_r= 1199 kg/m³, p_{up}= 1176 Pa and p_{down}= 427 Pa

$\boldsymbol{COP = 12.5}$

From the sample calculation above, it is proved that increase in density of pure refrigerant will increase COP.

3.0 RESULT

Result that is obtained from calculation is compared to the graphical result that has been published in the literature for validation. The thermo-physical properties data and their COP values are presented in Tables A.1-A.6 in appendix A and in Figures 3.1-3.9.

3.1 Viscosity

Viscosity of the R-134a and Al_2O_3/R-134a nanorefrigerant is calculated within temperature of 290K and 430K at 18 MPa. The R134a refrigerant is assumed to be in liquid phase. The viscosity data is obtained from Krauss et al. (1993).

Figure (3.1) shows viscosity comparison between Al_2O_3/R-134a nanorefrigerant and R-134a refrigerant in the temperature ranging from 275 K to 425 K. The measured viscosity for both Al_2O_3/R-134a nanorefrigerant and R-134a refrigerant was observed to be decreasing exponentially with an increase in refrigerant temperature. From Figure (3.1), nanorefrigerant viscosity is observed to be slightly higher compared to pure refrigerant and the change in viscosity for both pure and nanorefrigerant are the same.

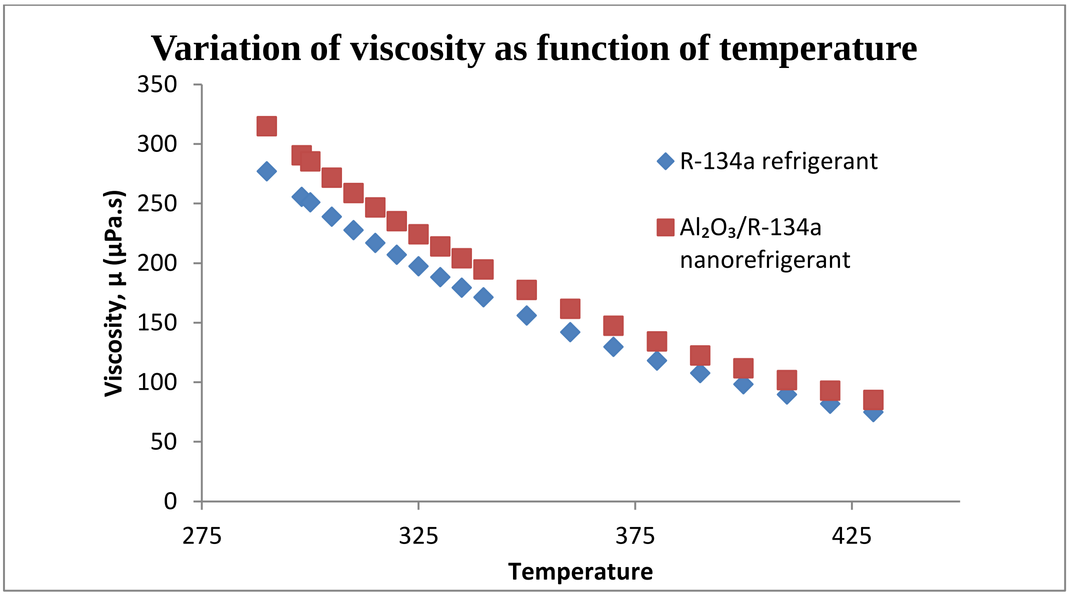


Figure 3.1 Variation of viscosity as a function of temperature

Figure (3.2) shows the change of COP ratio with respect to viscosity for R134a refrigerant and Al_2O_3/R-134a nanorefrigerant. COP ratio for both Al_2O_3/R-134a nanorefrigerant and R134a-refrigerant are observed to be decreasing with increasing viscosity. Graphs show that the change in viscosity for both refrigerants are similar.

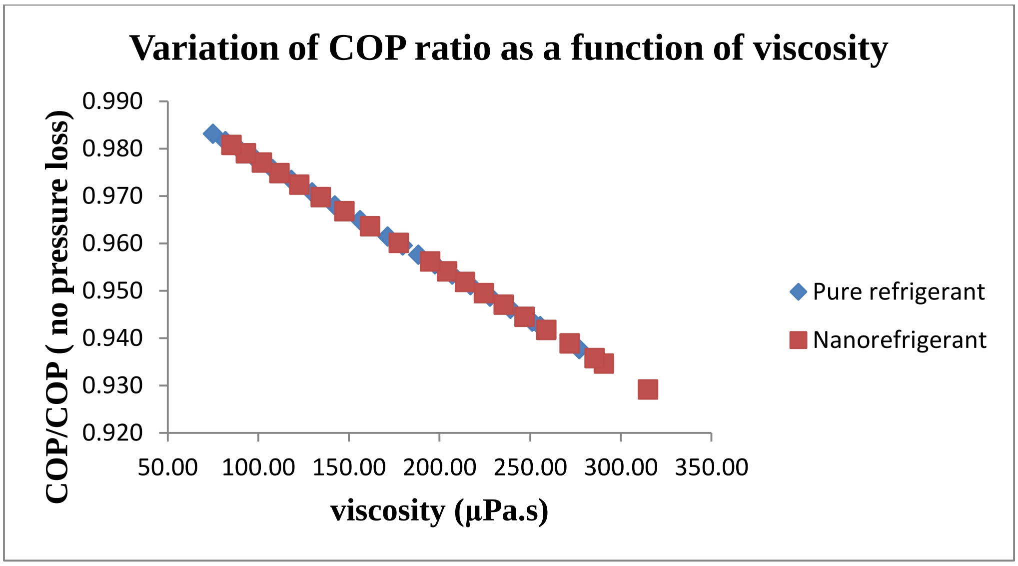


Figure 2.2 Variation of COP ratio of R134a refrigerant as function of viscosity

3.2 Thermal conductivity

Thermal conductivity values are obtained from Krauss et al. (1993) within temperature 240K and 410K at 18 MPa. The correlation of the thermal conductivity is based on the residual concept that combined three different types of approach. The thermal conductivity data is obtained by considering three different method; 1) using dilute gas function, the data is measured at atmospheric pressure or below, 2) using an excess or residual term, by subtracting the dilute-gas term and the critical enhancement term from the total quantity, 3) critical enhancement of the transport. Detail derivations of the said correlations are presented in appendix C, from Equation (C.2) until Equation (C.9).

From Figure (3.3), thermal conductivity of Al_2O_3/R-134a nanorefrigerant is exponentially increase with increasing temperature, while thermal conductivity for pure refrigerant is slightly decrease with increasing temperature.

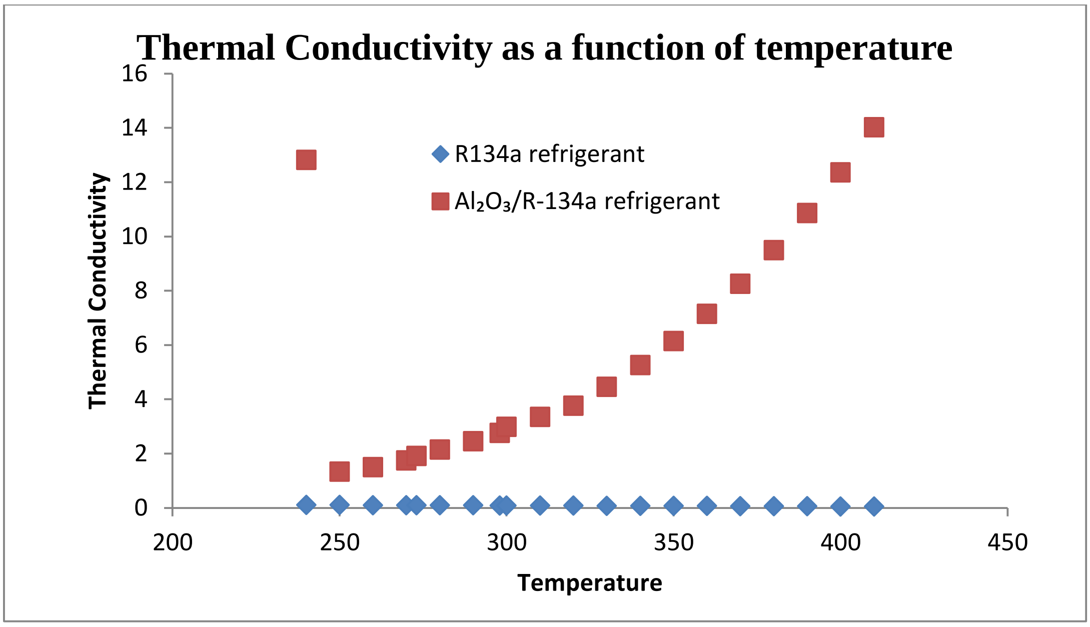


Figure 3.3 Variation of thermal conductivity as a function of temperature

In Figures 3.4 and 3.5, COP for both f Al_2O_3/R-134a nanorefrigerant and R134a-refrigerant are observed to be linearly increase with increasing thermal conductivity. The change in COP with respect to thermal conductivity for both pure and nanorefrigerant are observed to be similar.

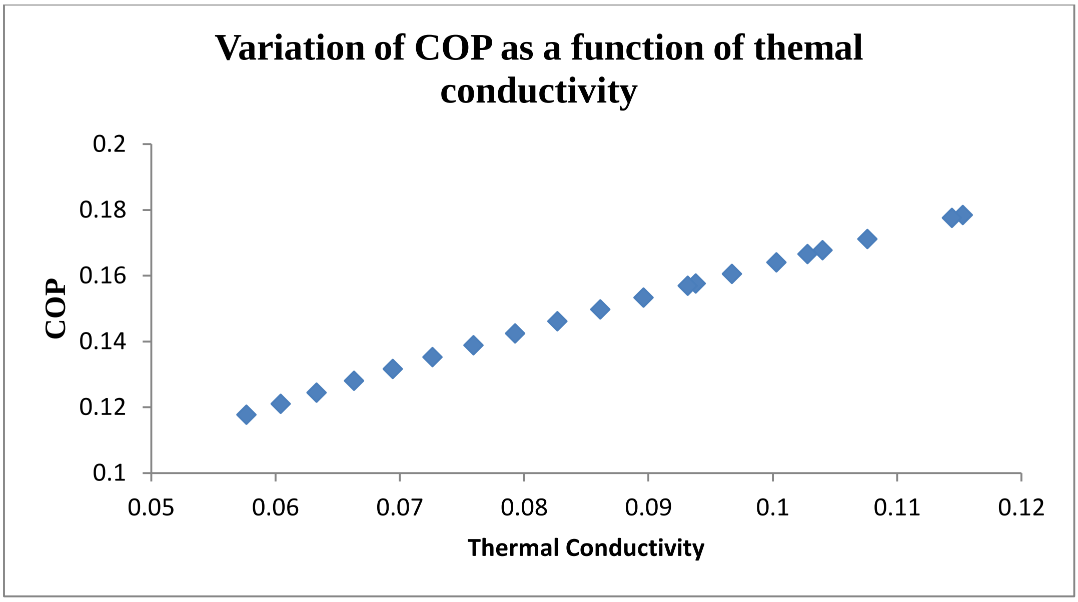


Figure 3.4 COP of R-134a refrigerant as a function of thermal conductivity

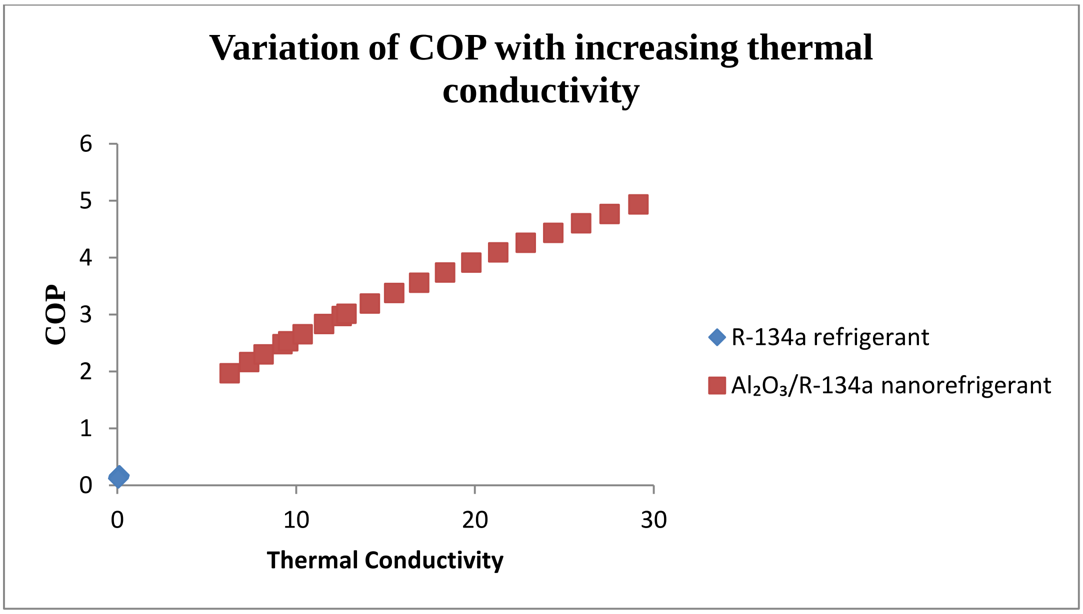


Figure 3.5 Variation of COP as a function of thermal conductivity

3.3 Specific heat

Specific heat is the one of the most important properties in refrigerant and it gives such a great influence towards heat transfer rate of refrigerant. Specific heat is amount of heat per unit mass required to raise temperature by one degree Celsius. For the calculation, specific heat data is obtained from refrigerant property database (REFPROP 7.0) that is developed by the National Institute of Standard and Technology (NIST) for

R-134a refrigerant. For this calculation, specific heat value is obtained from temperature between 240K until 270K.

From Figure 3.6, specific heat of both Al_2O_3/R-134a nanorefrigerant and R134a rerigerant is linearly increase with increasing temperature. Specific heat of R134a refrigerant is higher compared to Al_2O_3/R-134a nanorefrigerant. Fluid that contains nanoparticles (nanorefrigerant) are expected to give more specific heat over pure refrigerant. In Figure 3.7, COP for both Al_2O_3/R-134a nanorefrigerant and R134a-refrigerant are observed to be linearly increase with increasing specific heat. The trend in the change of COP with respect to specific heat for both pure and nanorefrigerant are observed to be similar.

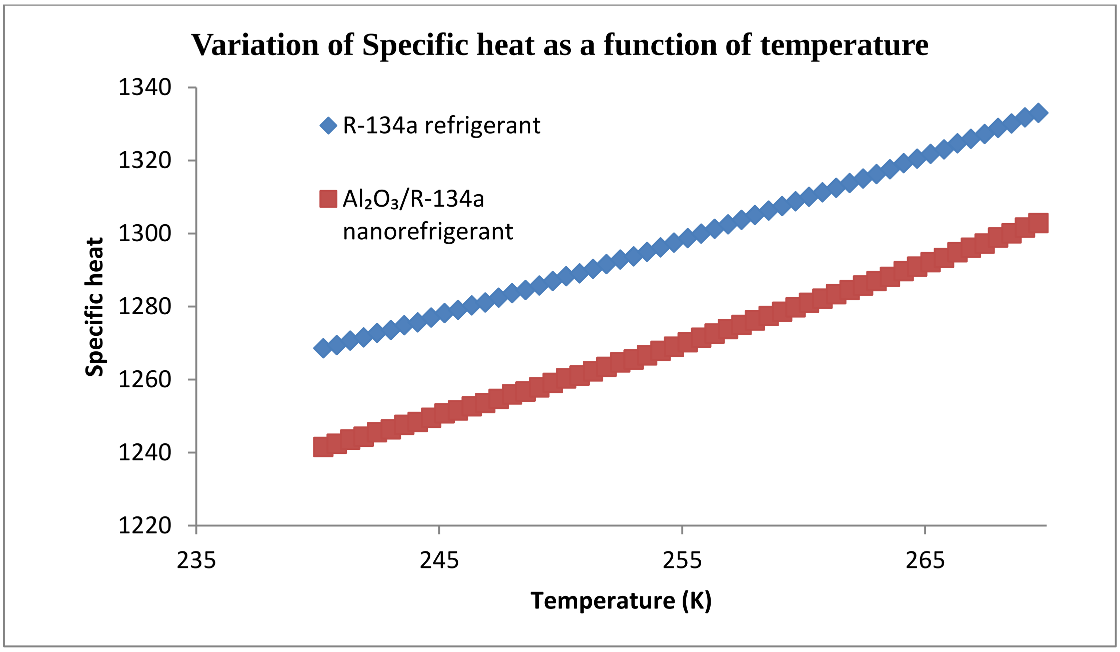


Figure 3.6 Variation of specific heat as a function of temperature

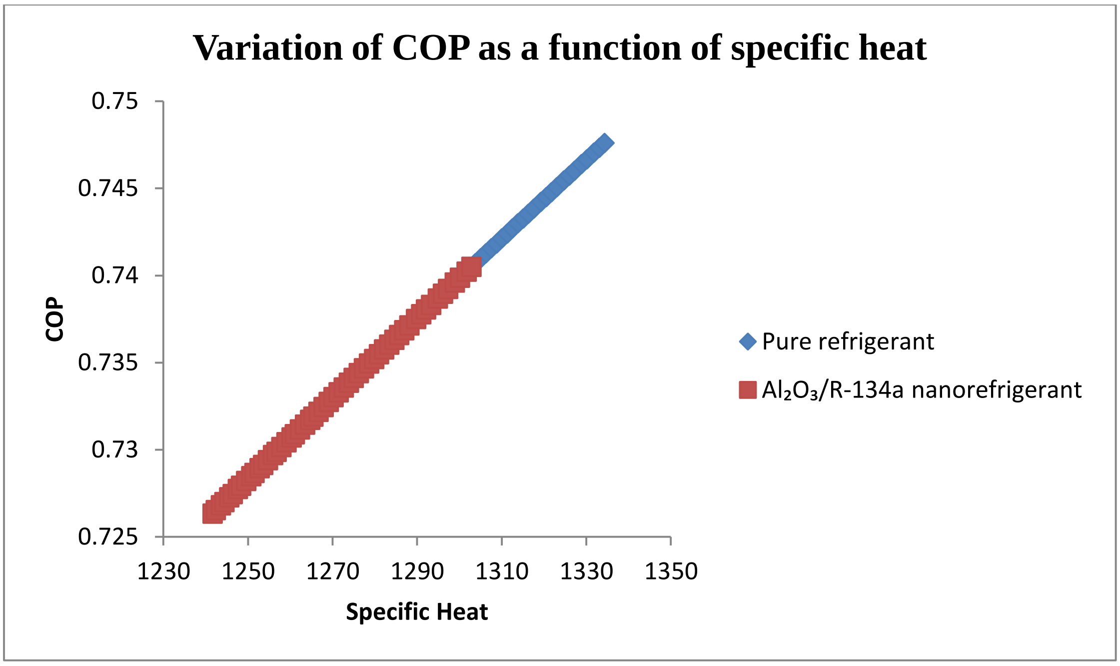


Figure 3.7 COP of R-134a refrigerant as a function of specific heat

3.3 Density

Density is one of the most important that effect heat transfer characteristics of the refrigerant. Density data is obtained from saturation properties of R134a from temperature of 240K until 374K. For refrigerant, a moderate density is in liquid form, a relative high density in gaseous form. In this calculation the moderate density is considered, so the refrigerant is assumed to be in liquid phase.

Figure 3.8 shows the change in density with respect to refrigerant's temperature. The density of Al_2O_3/R-134a nanorefrigerant and R134a-refrigerant are exponentially decrease with increasing temperature. Al_2O_3/R-134a nanorefrigerant exhibits more density compared to R134a-refrigerant. Figure 3.9 shows the change of COP with respect to viscosity for R134a refrigerant and Al_2O_3/R-134a nanorefrigerant. COP for both Al_2O_3/R-134a nanorefrigerant and R134a-refrigerant are observed to be increasing with increasing density. The change in density for both refrigerants are similar.

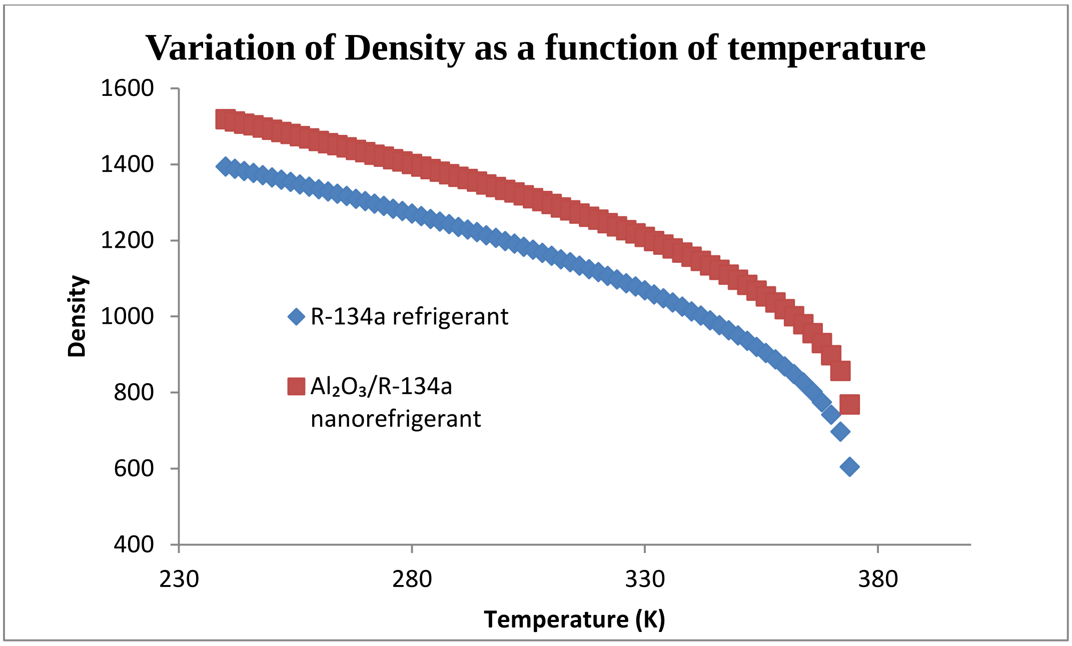


Figure 3.8 Variation of density as a function of temperature

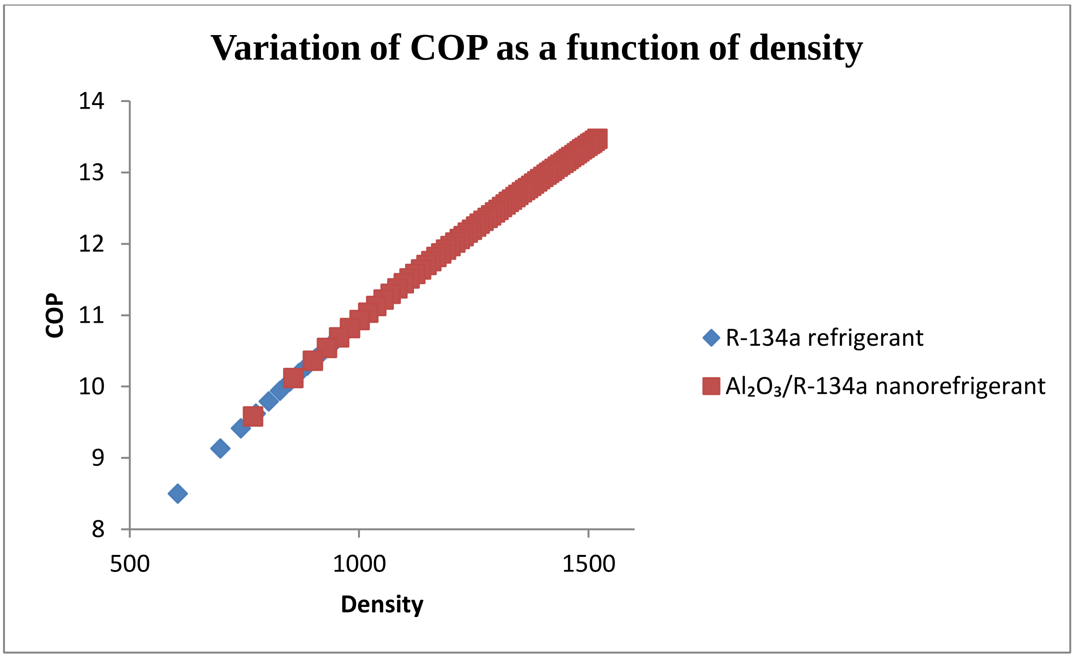


Figure 3.9 Variation of COP as a function of density

4.0 DISCUSSION

Figure 3.1 shows that both R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant is exponentially decrease with temperature. As temperature increase, the vibration between particles increase cause the molecular level of the liquid decrease. The more heat added to the liquid, the more particles will move and start to excite cause the liquid to be less viscous. From graph it shows that the fluid flow obeys Arrhenius equation, that is describe in Equation (C.1) Appendix C. The equation proves that fluid viscosity is decreased exponentially with increase temperature. From Figure 3.1, viscosity of Al_2O_3/R-134a nanorefrigerant is slightly higher than that of R-134a refrigerant due to the existence of the nanoparticles. Viscosity of nanofluid will increase with the increase of nanoparticle volume concentrations and decrease with increase in temperature (Stephen and Abdelsalam, 1980). High viscosity will affect system performance by effecting pumping power and pressure drop. Hence, in producing higher system performance, low viscosity is needed. High pumping power in heat exchanger is needed for high viscous fluid in order to encounter the frictional force of the fluid. The Prandtl number and Nusselt number of heat transfer fluids also depend on the viscosity property of fluids. Eventhough Al_2O_3/R-134a nanorefrigerant has higher viscosity compared to R-134a refrigerant, it is in fact doesn't give much effect on system performance since the viscosity value of nanorefrierant is still relatively low. Figure 3.2 shows that COP ratio for both R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant are decreasing with increasing viscosity. Higher values of liquid viscosity will cause the pressure drop in both evaporators and condensers to increase (Domanski and Didion, 1992). The amount of fluid friction is depends on the amount of the fluid viscosity and it shows how the fluid viscosity effect the flow of the fluid in a tube. Any type of obstruction, restriction or roughness in the system will contribute to increase pressure drop. The restriction in fluid flow is mainly due to high viscosity that causes resistance to fluid flow. This phenomenon is described by Equation (3.1). Increase in fluid viscosity will increase fluid flow resistance, hence increase pressure drop. Pressure drop is the difference in pressure between two point of fluid which is the different between discharge pressure and suction pressure. Pressure drop mainly occurs due to the frictional forces from the fluid resistance. Pressure drop increase proportionally with the fluid viscosity and decrease with increasing system temperature. As a result, the density of refrigerant entering the compressor will be decreased and the mass flow rate of refrigerant also decreases, which finally will cause the reduction in system capacity (Klein et al., 2000). Pressure drop will result in poor system performance and excessive energy consumption. In order produce a system with better performance, the working fluid (refrigerant) need to have relatively low viscosity.

From Figure 3.3, it shows that the thermal conductivity as a function of temperature of Al_2O_3/R-134a nanorefrigerant is much more higher compared to R-134a refrigerant. Since Al_2O_3 nanoparticles has high thermal conductivity and its aggregation in nanofluids cause the thermal conductivity of a nanofluid much higher than that of pure fluid. Mahbubul et al. (2013) introduced a thermal conductivity model of nanorefrigerant by considering the effect of particle volume fraction, particle size and temperature-dependent interfacial layer that is proposed by Sitprasert et al. (2009). Yu and Choi (2004) also take into consideration the effect of solid/liquid interfacial layers in thermal conductivity of nanofluid into their calculations. They renovated the Maxwell model and compute new set of correlations to determine thermal conductivity. The thermal conductivity values from these three methods are compared in order to verify the result. Figure 3.3 shows that thermal conductivity of Al_2O_3/R-134a nanorefrigerant increases with temperature while thermal conductivity of R134a refrigerant decreases with temperature. As the temperature increases, the Brownian motion of the nanoparticles will intensify and the nanorefrigerant viscosity will decrease. As the intensification of the Brownian motion increases, the contribution of microconvection in heat transport will increase hence increase thermal conductivity (Mahbubul et al. 2013). Based on the theory, it is proved that thermal conductivity of nanorefrigerant can be increased by increasing temperature. For pure refrigerant, thermal conductivity decreases with increasing temperature. This is because as temperature increases, the liquid is heated caused the atoms to positively charged and vibrate with greater amplitude. Since the molecules are vibrated farther apart, thermal conductivity of the liquid will be reduced. Figure 3.4 and 3.5 show that COP for both R-134a refrigerant and Al_2O_3/R134a nanorefrigerant are increasing with increasing thermal conductivity. Higher thermal conductivity give rise to higher heat transfer coefficient. Heat transfer coefficient is the amount of heat transferred from warmer region to another cooler region, thermal conductivity is how quickly the liquid able to absorb heat and heat capacity is how much heat can be absorbed. These three properties are proportional towards each other. As a result, the COP of the system will increase due to increase in heat capacity (Prapainop and Suen, 2012). Since higher energy levels are more easily accessed, the heat transfer rate should be increased. Wen and Ho (2005) computed mathematical model that compare heat transfer coefficient of R134a, R290, R600 and R600/R290. Based on their calculation, that concluded that with an equal mass flux, heat transfer coefficient of R290, R600 and R600/R290 are higher compared to R134a since these refrigerants have higher thermal conductivities than R134a. The paper proves that, liquid with higher thermal conductivity has higher heat transfer coefficient and that resulted to higher system performance.

Figure 3.6 shows that specific heat for both both R-134a refrigerant and Al_2O_3/R134a nanorefrigerant is proportional with temperature for increasing pressure. It is because increase in heat capacity will increase internal energy of the system. Increase in temperature will cause the liquid to fluctuates about its equilibrium value to a higher extent, then the heat capacity of the system will increase as more energy level will be filled up. Specific heat of Al_2O_3/R134a nanorefrigerant is slightly lower compared to R134a refrigerant. COP of R-134a refrigerant and Al_2O_3/R134a nanorefrigerant is increasing with increasing temperature, as shown in Figure 3.7. Lower heat capacity will cause high compression work due to increase in discharge temperature and as a result reduced thermal efficiency (Prapainop and Suen, 2012).. Increase in heat capacity will increase thermal conductivity since higher energy levels are more easily accessed. As a result, heat transfer rate will increase as well and eventually increase system performance. The graphs from Figures 3.3 and 3.6 proved that fluid containing nanoparticles are expected to give more thermal conductivity and lower specific heat compared to pure refrigerant.

According to Figure 3.8, both R -134a refrigerant and Al_2O_3/R134a nanorefrigerant density is decreasing with temperature since the volume of the liquid increase when the liquid is heated. Density is mass divided by volume. As the refrigerant is heated, the volume is increases because atoms start to vibrate cause the volume to increase, so the density will eventually decrease. Figure 3.8 also shows that density of Al_2O_3/R134a nanorefrigerant is slightly higher than that of R -134a refrigerant due to high density of nanorefrigerant that is extracted to the nanorefrigerant. From Figure 3.9, it can be seen that COP for both R-134a refrigerant and Al_2O_3/R134a nanorefrigerant increases with density. In centrifugal compressors, pressure rise is related to the density of the refrigerant. A high value of density results in high pressure rise which indirectly decrease pressure drop. Thus, increase system performance.

5.0 CONCLUSION

Understanding and research on role of refrigerant thermo- properties on the thermal performance of refrigeration system is crucial in order to obtain the most optimum cooling operation. Thermophysical properties, pressure drop, heat transfer performance and COP for both R -134a refrigerant and Al_2O_3/R134a nanorefrigerant have been investigated and the relationships between thermo-properties and COP have been evaluated. The following conclusions can be drawn from this study.

a) COP for both pure refrigerant and nanorefrigerant increases with increase thermal conductivity, specific heat, density and temperature
b) COP for both pure refrigerant and nanorefrigerant decreases with increase viscosity.
c) From the equation obtained, the calculation showed and concluded that Al_2O_3/R-134a refrigerant mixture has higher thermal conductivity, higher density, and higher viscosity compared to R-134a. Hence it is proved that COP of Al_2O_3/R-134a is higher than R-134a.

In general, to obtain high COP, combinations of high values of liquid thermal conductivity. liquid density, specific heat of liquid and low value of liquid viscosities are required.

4.1 Future Recommendation

The most important thermo-physical properties that influence refrigeration system performance include thermal conductivity, viscosity, specific heat and density. The relationship between each of the thermo-physical properties with COP (coefficient of performance) are presented in this report, in terms of mathematical models and graphs.

The mathematical model that relates COP with the four parameters are presented individually according to its thermo-physical property. For instance, the mathematical model that relates viscosity with COP is presented differently with the mathematical model that relates thermal conductivity with COP. In the future, another research need to be conducted in order to relate COP with the thermal conductivity, viscosity, specific heat and density all in one equation instead of four different correlations as presented in this paper. By combining all the equations into single equation, the more accurate result might be obtained.

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APPENDICES

APPENDIX A

Table A.1 : Viscosity and COP ratio of R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant

Temp. (K)	R-134a refrigerant: Viscosity, µ (µPa.s) @ 0.1 Mpa	R-134a refrigerant: COP/COP_no pressure losses	Al_2O_3/R-134a nanorefrigerant: Viscosity (Pa.s)	Al_2O_3/R-134a nanorefrigerant: COP/COP_no pressure losses
290	277.10	0.938	315.02	0.93
298	255.60	0.943	290.57	0.93
300	251.10	0.944	285.46	0.94
305	239.10	0.946	271.81	0.94
310	227.80	0.949	258.97	0.94
315	217.10	0.951	246.80	0.94
320	207.00	0.953	235.32	0.95
325	197.40	0.956	224.41	0.95
330	188.30	0.958	214.06	0.95
335	179.60	0.960	204.17	0.95
340	171.40	0.961	194.85	0.96
350	156.10	0.965	177.46	0.96
360	142.20	0.968	161.66	0.96
370	129.70	0.971	147.45	0.97
380	118.20	0.973	134.37	0.97
390	107.80	0.976	122.55	0.97
400	98.28	0.978	111.73	0.97
410	89.68	0.980	101.95	0.98
420	81.91	0.982	93.12	0.98
430	74.97	0.983	85.23	0.98

Table A.2 : Thermal conductivity and COP of R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant

Temp. (K)	**Thermal Conductivity, k (W.m^-1.K^-1)**		**COP**	
	Pure Refrigerant	**Nanorefrigerant**	**Pure Refrigerant**	**Nanorefrigerant**
240	0.1153	6.2909	0.1785	1.9666
250	0.1144	7.3811	0.1776	2.1646
260	0.1076	8.1789	0.1712	2.3020
270	0.1040	9.2424	0.1678	2.4772
273	0.1028	9.5642	0.1666	2.5286
280	0.1003	10.3633	0.1642	2.6534
290	0.0967	11.5573	0.1606	2.8328
298	0.0938	12.5473	0.1577	2.9760
300	0.0932	12.8102	0.1570	3.0132
310	0.0896	14.1186	0.1534	3.1943
320	0.0861	15.4797	0.1498	3.3756
330	0.0827	16.8855	0.1462	3.5563
340	0.0793	18.3305	0.1425	3.7359
350	0.0759	19.8069	0.1389	3.9137
360	0.0726	21.3113	0.1353	4.0894
370	0.0694	22.8392	0.1317	4.2629
380	0.0663	24.3830	0.1281	4.4335
390	0.0633	25.9428	0.1245	4.6016
400	0.0604	27.5298	0.1211	4.7685
410	0.0577	29.1394	0.1178	4.9338

Table A.3 : Specific heat and COP of R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant

Temp. (K)	**Specific Heat (J/kg.K)**		**COP**	
	Pure Refrigerant	**Nanorefrigerant**	**Pure Refrigerant**	**Nanorefrigerant**
240.2222	1268.5999	1241.6199	0.7326	0.7264
240.7778	1269.4372	1242.4154	0.7328	0.7265
241.3333	1270.6933	1243.6086	0.7331	0.7268
241.8889	1271.5306	1244.4041	0.7333	0.7270
242.4444	1272.7866	1245.5973	0.7336	0.7273
243.0000	1273.6240	1246.3928	0.7338	0.7275
243.5556	1274.8800	1247.5860	0.7341	0.7278
244.1111	1275.7174	1248.3815	0.7343	0.7279
244.6667	1276.9734	1249.5748	0.7346	0.7282
245.2222	1278.2295	1250.7680	0.7349	0.7285
245.7778	1279.0668	1251.5635	0.7351	0.7287
246.3333	1280.3229	1252.7567	0.7353	0.7290
246.8889	1281.1602	1253.5522	0.7355	0.7291
247.4444	1282.4163	1254.7455	0.7358	0.7294
248.0000	1283.6723	1255.9387	0.7361	0.7297
248.5556	1284.5097	1256.7342	0.7363	0.7299
249.1111	1285.7657	1257.9274	0.7366	0.7302
249.6667	1287.0218	1259.1207	0.7369	0.7304
250.2222	1288.2778	1260.3139	0.7372	0.7307
250.7778	1289.1152	1261.1094	0.7374	0.7309
251.3333	1290.3712	1262.3026	0.7377	0.7312
251.8889	1291.6272	1263.4959	0.7379	0.7315
252.4444	1292.8833	1264.6891	0.7382	0.7317
253.0000	1293.7206	1265.4846	0.7384	0.7319
253.5556	1294.9767	1266.6778	0.7387	0.7322
254.1111	1296.2327	1267.8711	0.7390	0.7325
254.6667	1297.4888	1269.0643	0.7393	0.7327
255.2222	1298.7448	1270.2576	0.7396	0.7330
255.7778	1300.0008	1271.4508	0.7399	0.7333
256.3333	1301.2569	1272.6440	0.7401	0.7336
256.8889	1302.5129	1273.8373	0.7404	0.7339
257.4444	1303.7690	1275.0305	0.7407	0.7341
258.0000	1305.0250	1276.2237	0.7410	0.7344
258.5556	1306.2810	1277.4170	0.7413	0.7347
259.1111	1307.5371	1278.6102	0.7416	0.7350

Table A.3, continued

259.6667	1308.7931	1279.8035	0.7419	0.7352
260.2222	1310.0492	1280.9967	0.7421	0.7355
260.7778	1311.3052	1282.1899	0.7424	0.7358
261.3333	1312.5612	1283.3832	0.7427	0.7361
261.8889	1313.8173	1284.5764	0.7430	0.7363
262.4444	1315.0733	1285.7696	0.7433	0.7366
263.0000	1316.3294	1286.9629	0.7436	0.7369
263.5556	1317.5854	1288.1561	0.7439	0.7371
264.1111	1319.2601	1289.7471	0.7442	0.7375
264.6667	1320.5161	1290.9403	0.7445	0.7378
265.2222	1321.7722	1292.1336	0.7448	0.7381
265.7778	1323.0282	1293.3268	0.7451	0.7383
266.3333	1324.7029	1294.9178	0.7455	0.7387
266.8889	1325.9590	1296.1110	0.7457	0.7390
267.4444	1327.2150	1297.3043	0.7460	0.7392
268.0000	1328.8897	1298.8953	0.7464	0.7396
268.5556	1330.1458	1300.0885	0.7467	0.7399
269.1111	1331.8205	1301.6795	0.7471	0.7402
269.6667	1333.0765	1302.8727	0.7473	0.7405
270.2222	1334.3326	1304.0660	0.7476	0.7408

Table A.4 : Density and COP of R-134a refrigerant and Al_2O_3/R-134a nanorefrigerant

Temp. (K)	**Density (kg/m^3)**		**COP**	
	Pure Refrigerant	**Nanorefrigerant**	**Pure Refrigerant**	**Nanorefrigerant**
240	1395.00	1519.25	12.9076	13.4702
242	1389.00	1513.55	12.8798	13.4449
244	1383.00	1507.85	12.8520	13.4195
246	1378.00	1503.10	12.8287	13.3984
248	1372.00	1497.40	12.8008	13.3730
250	1366.00	1491.70	12.7727	13.3475
252	1360.00	1486.00	12.7447	13.3220
254	1354.00	1480.30	12.7165	13.2964
256	1348.00	1474.60	12.6883	13.2708
258	1342.00	1468.90	12.6600	13.2451
260	1335.00	1462.25	12.6270	13.2151
262	1329.00	1456.55	12.5986	13.1893
264	1323.00	1450.85	12.5701	13.1635
266	1317.00	1445.15	12.5416	13.1376
268	1310.00	1438.50	12.5082	13.1073
270	1304.00	1432.80	12.4795	13.0813
272	1297.00	1426.15	12.4460	13.0509
274	1291.00	1420.45	12.4171	13.0248
276	1284.00	1413.80	12.3834	12.9943
278	1278.00	1408.10	12.3545	12.9681
280	1271.00	1401.45	12.3206	12.9374
282	1264.00	1394.80	12.2866	12.9067
284	1257.00	1388.15	12.2525	12.8759
286	1250.00	1381.50	12.2184	12.8450
288	1243.00	1374.85	12.1841	12.8140
290	1236.00	1368.20	12.1498	12.7830
292	1229.00	1361.55	12.1153	12.7519
294	1222.00	1354.90	12.0808	12.7207
296	1214.00	1347.30	12.0411	12.6850
298	1207.00	1340.65	12.0064	12.6537
300	1199.00	1333.05	11.9665	12.6177
302	1192.00	1326.40	11.9315	12.5862
304	1184.00	1318.80	11.8914	12.5501
306	1176.00	1311.20	11.8512	12.5139
308	1168.00	1303.60	11.8108	12.4776
310	1160.00	1296.00	11.7703	12.4412
312	1151.00	1287.45	11.7246	12.4001
314	1143.00	1279.85	11.6837	12.3634

Table A.4, continued

316	1134.00	1271.30	11.6376	12.3220
318	1125.00	1262.75	11.5914	12.2805
320	1117.00	1255.15	11.5501	12.2435
322	1107.00	1245.65	11.4983	12.1971
324	1098.00	1237.10	11.4514	12.1552
326	1088.00	1227.60	11.3992	12.1084
328	1079.00	1219.05	11.3519	12.0662
330	1069.00	1209.55	11.2992	12.0191
332	1058.00	1199.10	11.2409	11.9670
334	1048.00	1189.60	11.1877	11.9195
336	1037.00	1179.15	11.1288	11.8671
338	1026.00	1168.70	11.0696	11.8144
340	1014.00	1157.30	11.0047	11.7566
342	1003.00	1146.85	10.9448	11.7034
344	990.30	1134.79	10.8753	11.6417
346	977.60	1122.72	10.8054	11.5796
348	964.30	1110.09	10.7316	11.5143
350	950.50	1096.98	10.6545	11.4461
352	936.00	1083.20	10.5730	11.3740
354	920.80	1068.76	10.4868	11.2979
356	904.70	1053.47	10.3947	11.2168
358	887.60	1037.22	10.2960	11.1300
360	869.20	1019.74	10.1887	11.0358
362	849.20	1000.74	10.0708	10.9325
364	827.30	979.94	9.9401	10.8183
366	802.80	956.66	9.7918	10.6890
368	775.00	930.25	9.6208	10.5404
370	742.00	898.90	9.4137	10.3613
372	697.90	857.01	9.1297	10.1170
374	604.90	768.66	8.4996	9.5813

APPENDIX B

Table B.1 : Variations in refrigeration performance due to changes in thermophysical properties and derived parameters in sensitivities analysis.

Properties	Sensitivities	References
T_{crit}, T_b	T_{crit} and T_b $\uparrow$ 2 % => Q_{heat} $\downarrow$11-14 %, COP $\uparrow$ 0.3-1.8 % (vary depending on refrifegerants); T_{crit} and T_b $\downarrow$ 2 % => Q_{heat} $\uparrow$ 11-14 %, COP $\downarrow$ 0.7-2.3 % (vary depending on refrigerants)	Hogberg and Vamling (1996)
c_p	$c_{p,vap}$ $\uparrow$ 5% => Q_{heat} $\downarrow$ 1-4 %, COP $\downarrow$ 0.5%-1.2%, $c_{p,vap}$ $\downarrow$ 5% => Q_{heat} $\uparrow$ 1-4 %, COP $\uparrow$ 0.5-1.2%	Hogberg and Vamling (1996)
μ	μ_{liq} $\uparrow$ 15% => Q_{heat} $\downarrow$1-4%, COP $\downarrow$ 0.4-0.5% μ_{vap} $\uparrow$ 15% => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	μ_{liq} 50% => Q_{cool} $\downarrow$ 4 %, W $\downarrow$ 1%	
k	k_{liq} $\uparrow$ 10% => Q_{heat} $\uparrow$0.5-0.6 %, COP $\uparrow$ 0.5-0.6 % k_{vap} $\uparrow$ 15% => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	k_{liq} $\uparrow$ 50% => Q_{cool} $\downarrow$ 3 %, W_{com} $\uparrow$ 0.8 % k_{liq} $\downarrow$ 50% => Q_{cool} $\downarrow$ 7 %, W_{com} $\downarrow$ 1.6 %	Domanski and Didion (1987)
v	v_{liq} $\uparrow$10% => Q_{cool} $\downarrow$ 0.5 %, W_{com} ~ no change v_{vap} $\uparrow$ 10% => Q_{cool} $\uparrow$ 0.3 %, W_{com} $\downarrow$ 6% v_{vap} $\downarrow$ 10% => Q_{cool} $\downarrow$ 0.5 %, W_{com} $\uparrow$ 7%	Domanski and Didion (1987)
HTC_{evap}	HTC_{evap} $\uparrow$30% => Q_{heat} $\uparrow$2.2 %, COP $\uparrow$ 1.8% HTC_{evap} $\downarrow$30% => Q_{heat} $\downarrow$4 %, COP $\downarrow$ 2.5%	Hogberg and Vamling (1996)
	HTC_{evap} $\uparrow$50% => Q_{cool} $\uparrow$2.5 %, W_{com} $\uparrow$ 1% HTC_{evap} $\downarrow$50% => Q_{cool} $\downarrow$6 %, W_{com} $\downarrow$ 2.5%	Domanski and Didion (1987)
HTC_{cond}	HTC_{cond} $\uparrow$ 30% or $\downarrow$30% => Q_{heat} and COP ~ no significant change	Hogberg and Vamling (1996)
	HTC_{cond} $\uparrow$ 50% => Q_{cool} $\uparrow$ 0.5%, W_{com} $\downarrow$0.5% HTC_{cond} $\downarrow$ 50% => Q_{cool} $\downarrow$ 0.5%, W_{com} $\uparrow$1.5%	Domanski and Didion (1987)
$(\Delta P/\Delta z)_{evap}$	$(\Delta P/\Delta z)_{evap}$ $\uparrow$ 20% => Q_{heat} $\downarrow$ 0.6%, COP $\downarrow$ 0.2% $(\Delta P/\Delta z)_{evap}$ $\downarrow$ 20% => Q_{heat} $\uparrow$ 0.6%, COP $\uparrow$ 0.1%	Hogberg and Vamling (1996)
$(\Delta P/\Delta z)_{cond}$	$(\Delta P/\Delta z)_{cond}$ change from -50 to 100 % Q_{cool} and W_{com} ~ no significant change	Domanski and Didion (1987)

Source: Prapainop and Suen (2012)

Table B.2 : Refrigerant and refrigerant system parameters value.

Parameter	Value	Unit
L (temperature lift)	140	K
l (tube length)	4	m
V, volumetric flow rate	0.00036	m^3
G, gravitational acceleration	9.81	m/s^2
P_{evap}	51.25	Pa
A, orifice area	0.000032	m^2
$\dot{m}$, refrigerant mass flow rate	0.076	Kg/s
Q, heat output	1.81	kW

APPENDIX C

Additional Equations;

Arrhenius equation :

$$\mu(T) = \mu_0 exp\left(\frac{E}{RT}\right) \tag{C.1}$$

where T is temperature, μ_0 is the coefficient, E is the activation energy and R is the universal gas constant.

Dilute gas function;

$$\mu_0 = \frac{5}{16}\sqrt{\frac{MkT}{\pi N_A}}\frac{10^{24}}{\varsigma^2\Omega_\mu(T^*)} = \frac{0.2696566\sqrt{T}}{\varsigma^2\Omega_\mu(T^*)} \tag{C.2}$$

where M is the molecular weight equals to 0.10203 kg.mol^{-1}, k is the Boltzmann's constant (J.K^{-1}), N_A is the avogadro's number in mol^{-1}. The length scaling factor ς in nm. T^* is the dimensionless temperature defined as

$$T^* = kT/\varepsilon \tag{C.3}$$

For the collision integral Ω_μ as a function of T^*, using the formula presented by Bch et al. [38],

$$ln\Omega_\mu(T^*) = \sum_{i=0}^{4} a_i(lnT^*)^i \tag{C.4}$$

Table C.1 : Constant for Equations (C.2) and (C.4)

Variable	Value
ε/k	279.86 (K)
ς	0.50768 (nm)
a_0	0.4425728
a_1	-0.5138403
a_2	0.1547566
a_3	-0.02821844
a_4	0.001578286

Source: Krauss et al. (1993)

$$\lambda_0 = -16.5744 + 0.124286T - 0.761796 \times 10^{-4}T^2 \tag{C.5}$$

where λ_0 is the thermal conductivity of the dilute gas function.

Excess Function;

$$\frac{\Delta_R \mu}{H_c} = \sum_{i=1}^{3} e_i \left(\frac{\rho}{\rho_c}\right)^i + \frac{e_4}{\rho/\rho_c - e_5} + \frac{e_4}{e_5} \tag{C.6}$$

where ρ_c is critical density which is equal to 515.25 kg.m^{-3}, H_c is pseudo-critical viscosity defined as

$$H_{c=} \frac{M^{1/2}P_c^{2/3}}{R^{1/6}N_A^{1/3}T_c^{1/6}} = 25.21\mu Pa.s \tag{C.7}$$

where critical pressure P_C is equals to 4.065 MPa, critical pressure T_c is equals to 374.274 and R is the universal molar gas constant. The coefficients in equation (34) are tabulated in Table C.2

Table C.2 : Constant for Equation (C.6)

Coefficient	Value
e_1	-1.89758
e_2	0.256449
e_3	-0.301641
e_4	-23.1648
e_5	3.44752

Source: Krauss et al. (1993)

$$\frac{\Delta_R \lambda_t}{\Lambda_c} \sum_{i=1}^{4} l_i \left(\frac{\rho}{\rho_c}\right)^i \qquad (C.8)$$

where Λ_c is the pseudo-critical thermal conductivity defined as

$$\Lambda_c = \frac{R^{5/6} P_c^{2/3}}{T_c^{1/6} M^{1/2} N_A^{1/3}} = 2.055 mW.m^{-1}.K^{-1} \qquad (C.9)$$

The coefficients in Equation (C.8) is tabulated in Table (C.3)

Table C.3 : Constant for Equation (C.8)

Coefficient	**Value**
l	0.0579388
l_2	-0.254517
l_3	-3.52171
l_4	-0.371906

Source: Krauss et al. (1993)