POWER CONSUMPTION OF DOSMETIC REFRIGERATOR USING NANOPARTICLES Al_2O_3 LUBRICANT MIXTURE

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POWER CONSUMPTION OF DOMESTIC REFRIGERATOR USING NANOPARTICLES Al_2O_3 LUBRICANT MIXTURE

FANNY TEY PEI BOON

Report submitted in partial fulfillment of the requirements for the award of the degree of Bachelor of Mechanical Engineering

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JUNE 2013

UNIVERSITI MALAYSIA PAHANG

FACULTY OF MECHANICAL ENGINEERING

I certify that the project entitled “Power Consumption of Domestic Refrigerator using Nano-particles Al_2O_3 Lubricant Mixture” is written by Fanny Tey Pei Boon. I have examined the final copy of this project and in my opinion; it is fully adequate in terms of language standard and report formatting requirement for the award of the degree of Bachelor of Engineering. I herewith recommend that it be accepted in partial fulfilment of the requirements for the degree of Bachelor of Mechanical Engineering.

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Examiner Signature

SUPERVISOR'S DECLARATION

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I hereby declare that the work in this project is my own except for quotations and summaries which have been duly acknowledged. The project has not been accepted for any degree and is not concurrently submitted for award of other degree.

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Dedicated to my beloved parents

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ABSTRACT

This paper studied the performance of a domestic refrigerator operating with or without nanoparticles alumina (Al_2O_3) lubricant mixture with different level of refrigerant charged into the refrigeration system. The nanoparticles Al_2O_3 lubricant mixture by volume concentration of 0.2% was prepared via two step method. The quantity of refrigerant charged by pressure was ranged from 36 psi to 44 psi. The evaluation was carry out by comparing the overall performance in term of refrigeration capacity (Q_L), compressor work (W_{in}), coefficient of performance (COP) and power consumption for both sets of experimental testing. The refrigeration system of HFC134a/Al_2O_3/POE oil works normally and safely. From the results, the refrigeration performance for experiment operating with HFC134a/Al_2O_3/POE oil system was improved compared to HFC134a/ /POE oil system. Both refrigeration systems attained its optimum level at 42 psi charging pressure. The power consumption for HFC134a/Al_2O_3/POE oil system at optimum refrigerant charged which is 42 psi was the least with value 0.32 kW/h and the highest COP of 2.67. The percentage of increment in term of COP was between 2.81%, 3.59%, 3.89%, 4.55% and 5.91%. For the power consumption, the percentage of reduction was 1.47%, 1.61%, 1.80%, 2.07% and 1.77%. Thus, the addition of Al_2O_3 nanoparticles into the base fluid (POE) for better performance and reduction of power consumption was achievable.

ABSTRAK

Tesis ini mengkaji prestasi bagi suatu peti sejuk domestik yang beroperasi dengan atau tanpa campuran nanopartikel alumina (Al_2O_3) dan minyak pelincir ke atas prestasi tahap penyejukan yang berbeza. Campuran nanopartikel Al_2O_3 dan minyak pelincir dengan kepekatan sebanyak 0.2% telah disediakan melalui kaedah dua langkah. Kuantiti penyejukan yang dicaskan dengan tekanan antara 36 psi hingga 44 psi. Penilaian telah dijalankan dengan membandingkan prestasi keseluruhan dari segi kapasiti penyejukan (Q_L), kerja pemampat (W_{in}), COP dan penggunaan kuasa elektrik bagi kedua-dua set eksperimen ini. Sistem penyejukan dengan HFC134a/Al_2O_3/POE berfungsi dengan normal dan selamat. Daripada keputusan yang diperolehi, prestasi penyejukan untuk sistem HFC134a/Al_2O_3/POE telah bertambah baik jika dibandingkan dengan system HFC134a / POE. Kedua-dua sistem penyejukan mencapai tahap yang optimum pada 42 psi. Penggunaan kuasa electrik untuk sistem HFC134a/Al_2O_3/POE dengan kuantiti penyejukan yang optimun iaitu 42 psi adalah 0.32 kW / j dan COP yang tercapai adealah tertinggi iaitu 2.67. Peratusan kenaikan untuk COP adalah di antara 2.81%, 3.59%, 3.89%, 4.55% dan 5.91%. Untuk penggunaan kuasa elektrik, peratusan pengurangan adalah 1.47%, 1.61%, 1.80%, 2.07% dan 1.77%. Oleh itu, penambahan campuran nanopartikel Al_2O_3 ke dalam bendalir asas (POE) untuk prestasi yang lebih baik dan pengurangan penggunaan kuasa electric telah tercapai.

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LIST OF SYMBOLS

Al_2O_3	Aluminium (III) Oxide / Alumina
˚c	Celsius
ρ	Density
˚F	Fahrenheit
g	gram
H_2O	Water
kJ	kilojoule
kg	kilogram
Kw/h	Kilowatt/hour
m	mass
ml	millilitre
Q_L	Refrigeration Capacity
V	volume
W_{in}	Compressor Work

LIST OF ABBREVIATIONS

ASHRAE	American Society of Heating, Refrigeration and Air-Conditioning Engineer
COP	Coefficient of Performance
HFC134a	Refrigerant R-134a/ 1,1,1,2-Tetrafluoroethane
POE	Poly-ol-ester Oil

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND

The advancements of nanotechnology have allowed the development of nanoparticles as an additive for Tetrafluorethane (HFC134a)/POE lubricant oil system which might improve the performance of refrigeration system. The nano additives can be charged into the refrigeration system via mixing nanoparticles with the existing lubricant oil. The nano additives might reduce the wear, tear and friction better than the existing lubricant oil in the compressor. Besides, the addition of nanoparticles in mineral based polyester lubricant oil can be used as to enhance the overall heat transfer performance of refrigeration system in domestic refrigerator. Recently, many researchers emphasized on the importance of nanoparticles as additives as it shows greater promise for use in cooling system such as refrigeration system.

The refrigeration systems used in domestic refrigerators mostly are of compression type system. The working principle of the vapor-compression refrigeration cycle is modified from the reversed Carnot cycle. It is a steady state refrigeration system where the refrigerant flows in one direction steadily with continuous cooling effect. The continuous improvement and modifications of the reversed Carnot cycle have resulted in the introduction of the vapor-compression refrigeration cycle and until now, the household refrigeration and air conditioning working principle is based on this cycle. Jacob Perkins, an Englishman, built a prototype of closed cycle ice machine based on the vapor compression cycle in 1834; it was commercialized by Alex Twinning in 1850 then. There are many research and experiments have been done as to enhance the overall performance of the refrigerator and cost savings.

A basic refrigeration system consists of four fundamental elements which are the compressor, condenser, expansion valve and evaporator. In real condition, the theoretical performance of refrigeration system was deteriorated due to the internal and external reversibility in the system. The internal reversibility is due to the design and selection of compressors used in the refrigerator. For the external reversibility, it is due to losses of condenser and evaporator. Therefore, the effectiveness on the functionality for each component is influencing the performance of the refrigeration system directly.

For the refrigerant compressors, they are known as the heart of the vapor-compression systems. A refrigerant compressor controls the circulation of the refrigerant by adding pressure on it. The refrigerant is compress to high pressure and high temperature from the evaporator at this stage. However, for the refrigerant compressor to be maintained in a hermetic condition, lubricant oil is necessary in lubricate the moving part in a domestic refrigerator. The quality of lubricate oil is closely associated with the lifespan of the compressor. It is because the lubricant oil is used to lubricate the internal parts of the compressor by mixing with the refrigerants employed in the refrigeration system. Commonly, mineral based oil such as polyester (POE) lubricant oil is used as to reduce wear, tear and friction on metal parts of the compressor in industrial applications. When the compressor pumps the refrigerant around the circuit, the temperature will be increased. As the temperature increases, the internal metal parts of the compressor heating up. The lubricant oil functioned as to cool down and lubricate the hot metal parts so that the compressor able to operate efficiently.

Then, the refrigerant is discharged into condenser upon completion of compression process. The condensation process rejects heat to the surroundings. The condensed refrigerant will flow into the expansion device after condensation process. The refrigerant vapour expanded as the pressure reduced. At this state, it draws the energy from surroundings or medium in contact as to produce continuous refrigerating effect. The cycle is complete when the refrigerant flows into the evaporator, from the expansion valve, as a low-pressure, low-temperature liquid.

Despite of that, the refrigerant charged into the refrigeration system is another factor that influenced the performance of refrigeration system. This parameter can be altered easily as to optimize the performance of refrigerator. Anyhow, the refrigerant charged into the refrigeration system should be kept within the compressor’s limits. It is to avoid flooding or slugging due to excessive liquid refrigerant charge in the refrigeration system.

A lot of advancement happens in our world today, a need to reduce the power consumption to lubricate the moving parts of the refrigerator using nano additives in lubricant for refrigeration system was compromised. Through the employment of HFC134a/POE oil system with nano addtives, the flow and thermal characteristics might be improved. Therefore, a reduction in energy consumption of refrigerator is possible via this heat transfer enhancement technology.

The rate and amount at which the condenser discharges the heat to the environment and also the rate and amount which the evaporator absorbs heat from the compartment can be a measure of performance and efficiency of the refrigeration system. The measure on performance and efficiency of the refrigeration system is evaluated based on the coefficient of performance, COP computed. As the COP higher, the performance of the refrigeration system better. In addition, the power consumption of the refrigerator which operating with HFC134a/POE oil system with or without nanoparticles $Al_2O_3$was analyzed based on the data recorded in Wh. The power consumption of the refrigeration system operating with HFC134a/POE oil with nanoparticles Al_2O_3 will be highlighted.

This project report is to give the basic understanding on the potential of nanoparticles Al_2O_3 as an additive in conventional HFC134a/POE oil system with different refrigerant charged.

1.2 PROBLEM STATEMENT

A refrigerator can be one of the most costly household appliances. There are several methods for mankind to reduce the electricity bills by reducing energy consumption of the domestic refrigerator such as turn off the ice maker or water dispenser, unplug the refrigerator, regular checking on worn and cracked seals and minimize the amount of times on opening the doors of the refrigerator. However, this method cannot solve the problem permanently and yet inconvenient to other parties under the same roof.

Nanotechnology has the potential to revolutionize the way we live our daily lives. It can make our life much easier. The development of nano additive to the existing refrigeration HFC134a/POE oil system might enhance heat transfer; reduce the work load of compressor and thus the amount of power consumption in the refrigeration system as well. The study needed to analyze the actual performance of present refrigeration HFC134a/POE oil system and compare it with refrigeration HFC134a/Al_2O_3/POE oil system

1.3 OBJECTIVES

The main objective of this study is:

i) To conduct an experimental testing operating on HFC134a/POE oil system with or without addition of nanoparticles Al_2O_3 with different refrigerant charged into the system
ii) To prepare nanoparticles Al_2O_3 polyester lubricant mixture
iii) To analyse the performance of HFC134a/POE oil system with or without addition of nanoparticles Al_2O_3.
iv) To compare the performance on HFC134a/POE oil system with or without addition of nanoparticles Al_2O_3.

1.4 SCOPES

i) Fundamental study of refrigeration system HFC134/POE oil system with different refrigerant charged
ii) Potential of oxide based nanoparticles Al_2O_3 (alumina) as additives in lubricant
iii) Sample preparation of nanoparticles Al_2O_3 polyester lubricant mixture via two step method

1.5 LIMITATIONS

The limitations in conducting this study included the location of the experimental test rig. The experimental testing on refrigeration system is unable to carry out in control room. Therefore, there are several environmental factors that are neglected during this experimental testing such as the surrounding temperature and humidity. Besides, the experimental testing for refrigeration system should be run for a periods of 24 hours. The experimental testing can only run for ten hours due to the restricted accessibilities of students to laboratory. The range of temperature setting for thermostat in the refrigerator is fixed at less than half of its range.

CHAPTER 2

LITERATURE REVIEW

2.1 THERMODYNAMICS LAW

The First Law of Thermodynamics asserts that energy is a thermodynamics property whereas the Second Law of Thermodynamics asserts that energy has quality as well as quantity and actual processes occur in the direction of decreasing quality of energy. However, a process cannot take place in a refrigeration system unless it satisfied both the first and second laws of thermodynamics.

As we known, the heat is transferred in the direction of decreasing temperature. The heat is transferred form high temperature mediums to low temperatures ones. The processes occur in a certain direction but not in the reverse direction. The reverse process cannot occur by itself. Therefore, the heat transfer from a low temperature medium to a high temperature one which requires special devices are called refrigerator.

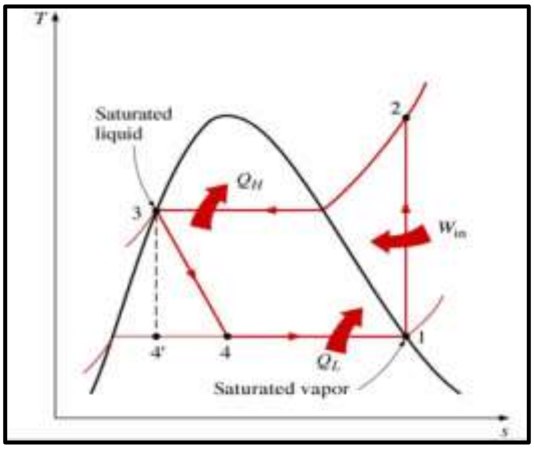
The Second Law of Thermodynamics is used in determining the theoretical limits for the performance of refrigerators. Under the Second Law of Thermodynamics, there are the Kelvin Planck and Clausius statement. For the Clausius statement which is related to refrigerators implies that it is impossible to construct a device that operates in a cycle and produces no effect other than transfer of heat from a lower temperature body to a higher temperature body (Cengel, 2007). Commonly, the heat does not naturally transfer on its own from a cold medium to a warmer one. Indirectly, the Clausius statement means if there is a cyclic device that able to produce net effect on others, the heat can be transferred from a cold medium to a warmer one. Therefore, a refrigerator must be operated with a compressor driven by an external power source. The net effect on the surroundings mentioned involves the consumption of energy in the form of work. The power consumption by the compressor work will be very useful on evaluation of the performance of refrigerator. To conclude, a domestic refrigerator is in complete compliance with the Clausius statement of the second law.

2.2 IDEAL VAPOR COMPRESSION REFRIGERATION CYCLE

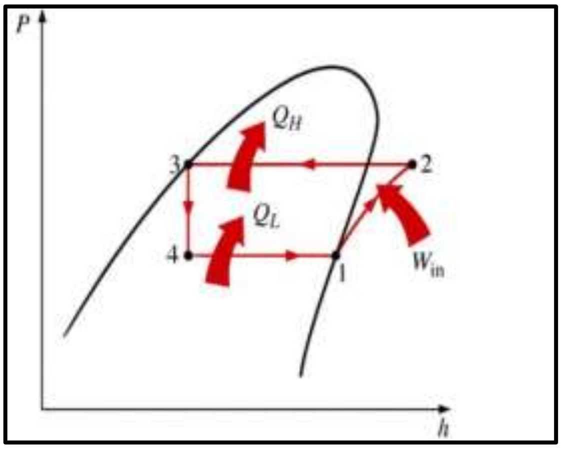
Commonly, the ideal vapour-compression refrigeration cycle is employed for most refrigeration system of refrigerator. In ideal vapour-compression refrigeration cycle, the refrigerant vaporized completely in the evaporator before compression. Then, an expansion valve or capillary tube which acts as the throttling device was used to replace the turbine. There are four fundamental processes in an ideal vapour-compression refrigeration cycle where the refrigerant has to go through inside the refrigeration system.

i) Isentropic Compression in a Compressor
ii) Constant Pressure Heat Rejection in a Condenser
iii) Throttling in an Expansion Device
iv) Constant Pressure Heat Absorption in an Evaporator

Figure 2.1 is the ideal vapour compression refrigeration cycle where (a) is the temperature-entropy (T-s) diagram of the cycle and (b) is the pressure-enthalpy (p-h) diagram of the cycle.



(a)



(b)

Figure 2.1: Ideal vapour compression diagram (a) T-s diagram, (b) p-h diagram

Source: Cengel 2007

In an ideal vapour compression refrigeration cycle, the refrigerant enters the compressor as saturated vapour at stage 1. Then, it is compressed isentropically to condenser pressure. During the isentropic compression process, there is an increase in the temperature of the refrigerant than that of the surrounding medium. The refrigerant enters condenser as saturated vapour at stage 2. The refrigerant leaves the condenser as saturated liquid at stage 3. The heat rejected to surroundings. The saturated liquid at stage 3 is passing through an expansion valve or capillary tube which acts as the throttling device. During this process, there is a decrease in the temperature of the refrigerant than that of the refrigerated space. At stage 4, the refrigerant enters as a low quality saturated mixture. It is completely evaporates by absorbing heat from the refrigerated space. The refrigerant will then leave the evaporator as saturated vapour and re-enters the compressor to complete the cycle.

2.3 VAPOR COMPRESSION REFRIGERATION SYSTEM

As mentioned above, in a basic vapour compression refrigeration system, there are four fundamental thermal processes take place. There are evaporation, compression, condensation and expansion. The schematic diagram of the basic vapour compression refrigeration cycle in refrigeration system is shown in Figure 2.2.

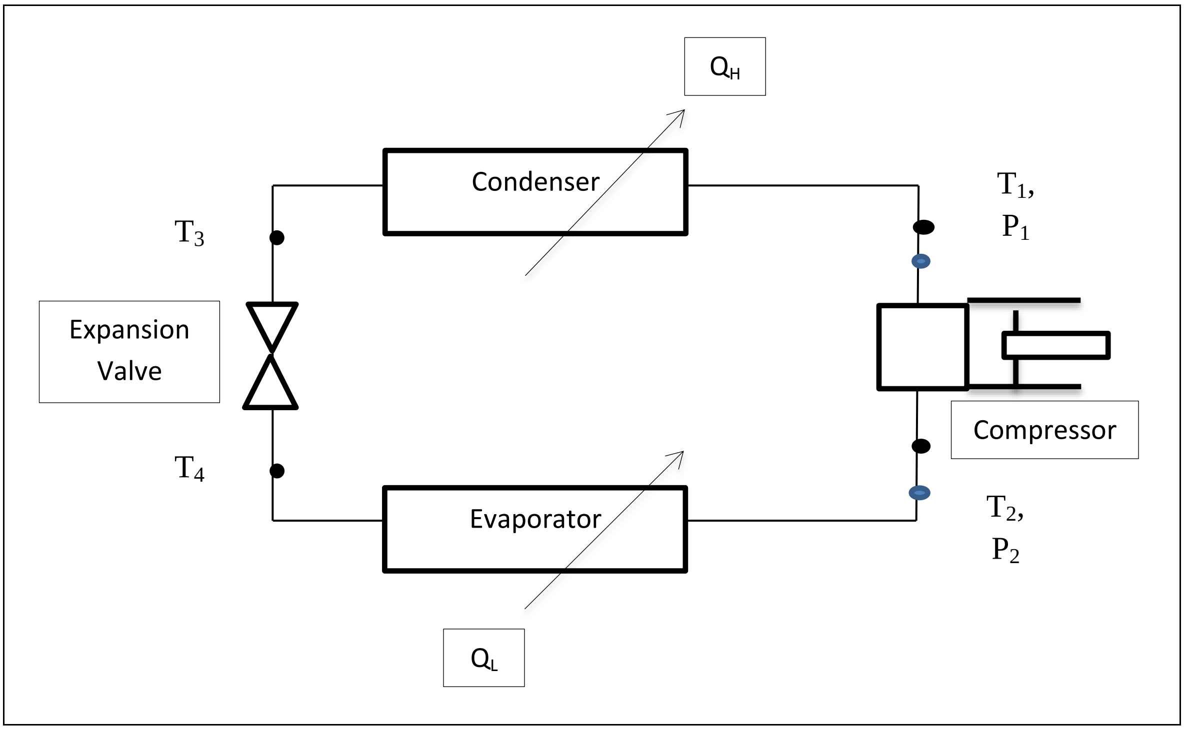


Figure 2.2: Schematic diagram of the basic vapor compression refrigeration cycle in refrigeration system

By applying steady-state flow according to the first law of thermodynamics, a vapour compression refrigeration cycle which comprises of a number of flow processes as illustrated below can be analysed (Dincer, 2003). Assume that the changes in kinetic and potential energies are negligible.

2.3.1 Evaporation Process

Evaporation is the gaseous escape of molecule from the surface of a liquid. It is accomplished by the absorption of a considerable quality of heat without any change in temperature. In the evaporator of a refrigeration system, a low pressure cool refrigerant vapor is brought into contact with the medium to be cooled, absorb heat, and hence boil, producing a low pressure saturated vapor. q_L is the heat taken from the low temperature environment to the evaporator per unit mass as expressed in Eq. (2.1). The refrigeration capacity is expressed as Eq. (2.2).

$$qL = (h1 - h4)kJ/kg \tag{2.1}$$

$$QL = m(h1 - h4)kW \tag{2.2}$$

2.3.2 Compression Process

The compressor raises the pressure of the refrigerant vapor obtained from the evaporator. The compressor work per unit mass is expressed as in Eq. (2.3). W_{in} is compressor power input, kW and expressed as in Eq. (2.4).

$$win = (h2 - h1)kJ/kg \tag{2.3}$$

$$Win = m(h2 - h1)kW \tag{2.4}$$

2.3.3 Condensation Process

Under condensation, the vapor is changed into a liquid by extracting heat. The refrigerant of high pressure and gaseous carries the heat energy absorbed in the evaporator and the work energy from the compressor into condenser. q_H is the heat rejection from the condenser to the high temperature environment per unit mass as expressed in Eq. (2.5). The heating capacity is expressed as Eq. (2.6).

$$qH = (h2 - h3)kJ/kg \tag{2.5}$$

$$QH = m(h2 - h3)kW \tag{2.6}$$

2.3.4 Throttling Process

The condensed refrigerant liquid is returned to the beginning of the next cycle. A throttling device such as expansion valve is used to reduce the pressure of the refrigerant liquid to the low pressure level and the boiling temperature of the refrigerant to below the temperature of the heat source. The enthalpy throughout the system remains as expressed in Eq. (2.7).

$$h3 = h4 \; kJ/kg \tag{2.7}$$

2.3.5 Coefficient of Performance

The coefficient of performance, COP of the refrigeration system is the ratio of refrigeration effect, Q_L to the compressor work per unit mass, W_{in}. The COP is expressed as in Eq. (2.8).

$$COP = \frac{Q_L}{W_{in}} \tag{2.8}$$

2.4 ACTUAL VAPOR COMPRESSION REFRIGERATION CYCLE

An actual vapor compression refrigeration cycle differs from the idealized model due to the irreversibility that occurs in various components such as fluid friction and entropy generation in the system. The Temperature-Entropy ($T - s$) diagram of actual vapor compression refrigeration cycle is shown in Figure 2.3.

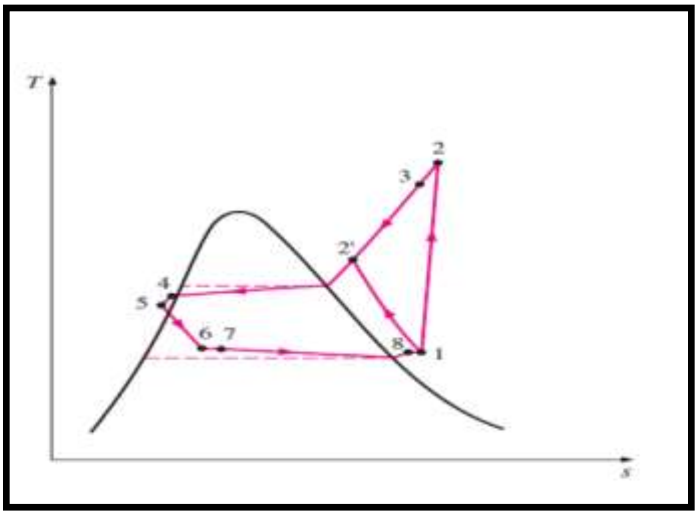


Figure 2.3: Temperature-Entropy (T-S) diagram of actual vapor compression refrigeration cycle

Source: Cengel 2007

The irreversibility that occurs in an actual vapor compression refrigeration cycle is divided into internal and external irreversibility in the system. For internal irreversibility, it is due to non-isentropic compression, friction and heat transfer to or from the surroundings. It is dependent on the selection and design of compressor used in the refrigerator. The external irreversibility is due to the losses on condenser and evaporator due to finite rate of heat exchange against finite values of temperature difference and heat capacities of external fluids. Therefore, a vapour compression refrigeration system can be optimized theoretically and balanced via finite time thermodynamics. However, a correct estimation on parameters involving in irreversibility is a real challenge.

For heat transfer coefficient on the external fluid in the evaporator and condenser, it can be optimized easily. However, for the heat transfer coefficient over the refrigerant R-134a in condenser, it is difficult. This is because they are closely associated with the phase change of refrigerant in the refrigeration system. The boiling heat transfer coefficient over the refrigerant R-134a in evaporator is hard to be estimated as well (J. K. Dabas, 2011).

2.5 COMPONENTS OF THE MECHANICAL VAPOR COMPRESSION REFRIGERATION SYSTEM

As mentioned earlier, there are four fundamental components employed in a refrigeration system. There are evaporator, compressor, condenser and capillary tube which acts the throttling device.

2.5.1 Evaporator

Evaporator divided into two main categories. There are direct cooler evaporators that cool air and in turn cool the products and indirect cooler evaporators that cool a liquid such as brine solution and in turn cool the product (Dincer, 2010).Commonly, the evaporators such as liquid coolers and air coolers or gas coolers used for cooling, refrigeration, freezing and air conditioning applications (Dincer, 2010).

2.5.2 Compressors

Generally, the compressor functioned as a mechanical device to compress and pump the refrigerant vapour from a low pressure region (the evaporator) to a high pressure region (the condenser) (M.Y.Taib). This is important in maintaining the desired temperature and pressure in the evaporator. Besides, the secondary function of the compressor within the refrigeration system is to increase the pressure and temperature of the refrigerant vapour.

There are several types of compressor used in refrigeration system such as hermetic compressors, semi-hermetic compressors, open compressors and displacement compressors.

Hermetic compressors are preferable due to its design for smaller range of temperatures required in air cooling applications primarily. Hermetic compressors are available for small capacities. The motor and drive of compressors are sealed in compact welded housing which is also where the refrigerant and lubricating oils are contained. Another advantage of hermetic compressors is it can work for a long period of time in a small-capacity refrigeration system without any maintenance requirement and without any gas leakage. Besides, it is sensitive to electric voltage fluctuations which may make the copper coils of the motor burn. The hermetic type compressors are widely used in domestic refrigerator, freezers and air conditioners (Dincer, 2010). Figure 2.4 is the cut-in view of the hermetic typed compressor.

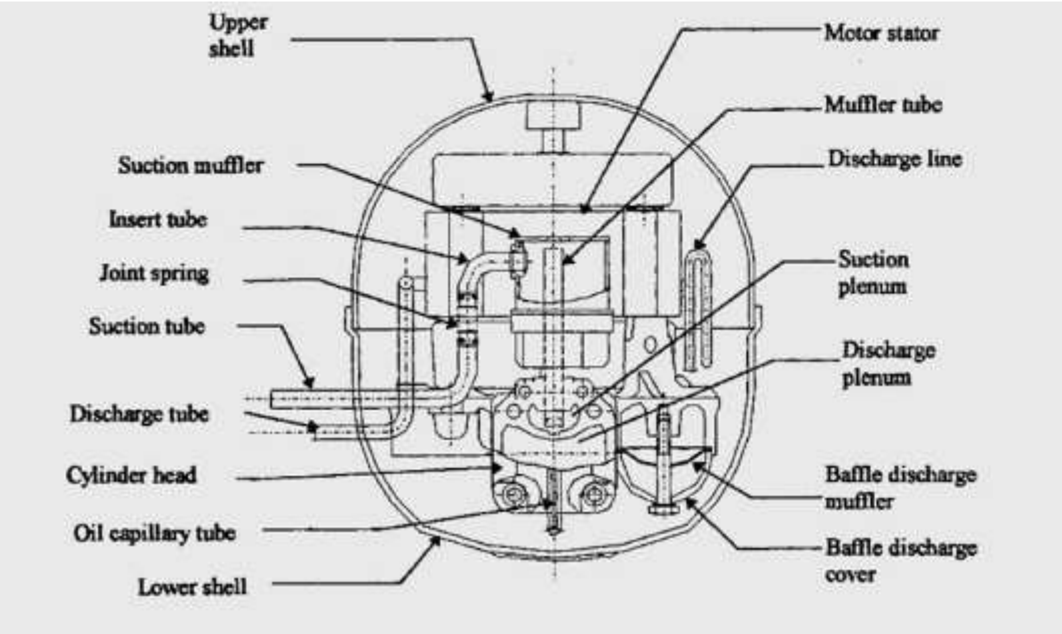


Figure 2.4: Cut-in view of the hermetic compressors

Source: Dincer (2010)

2.5.3 Condenser

The condenser is selected based on the criterion such as the size of the cooling load, refrigerant used, quality and temperature of available cooling water and amount of water that can be circulated. There are three types of condensers utilized in the refrigeration industry. There are water-cooled, air cooled and evaporative condensers. For domestic refrigerator, the air-cooled condensers are the most applicable with a common capacity of 20-120tons. The advantages of air cooled condensers are no water requirement, low installation cost and low maintenance and service requirement (Dincer, 2010).

2.5.4 Capillary Tube

There are many types of throttling devices such as thermostatic expansion valves, constant pressure expansions valves, float valves and capillary tubes. The simplest types of refrigerant flow control device employed in refrigeration system are capillary tubes (Dincer, 2010). The capillary tube controls the refrigerant flow from the condenser to the evaporator and separates the system, to high pressure and low pressure sizes (M.Y.Taib).

Capillary tube more widely used in a small hermetic-type refrigeration systems which up to 30kW capacity. It reduces the condensing pressure to the evaporating pressure in a copper tube with small internal diameter. These tubes are used to transmit pressure from the sensing bulb of some temperature control device to the operating element (Dincer, 2010).

2.6 IMPACTS OF REFRIGERANT CHARGED ON VAPOUR COMPRESSION REFRIGERATION SYSTEM

Generally, there are several factors influencing the performance of the refrigeration system. The seven factors that influencing the performance of a refrigerator are ambient temperature, compressor speed, refrigerant charge, humidity, door opening, thermal load and compartment internal temperature inside the refrigerator. All factors mentioned are essential for a steady operation on refrigeration system. They influenced the performance of a refrigerating system directly.

For the amount of refrigerant charged into the refrigeration system, it does give significant effect on the performance of the refrigeration system. The refrigerant charged is one of the factors that can be altered easily in the refrigerating system.

For low refrigerant charge levels, it can cause significant reduction in cooling and heating capacity. As the refrigerant charged increased, the overall performance of the refrigerating system increased. It is because the amount of refrigerant charged into the refrigeration system enhanced the overall heat transfer coefficient in the evaporator by increasing the part of space occupied by liquid refrigerant in the evaporator. The pool boiling and nucleate boiling conditions in evaporator attained maximal level on filling of evaporator with liquid. Thereby, the heat transfers increased multiple times (J. K. Dabas, 2011).

However, flooding or slugging could be occurred due to excessive liquid refrigerant charge in the refrigeration system.

2.7 HFC134a/POE OIL SYSTEM WITH AND WITHOUT NANOPARTICLES ALUMINA

Commonly, the working fluid employed in the refrigeration system is refrigerant of type R-134a with a polyester lubricant. The non-flammable R-134a is the universal choice as an HFC zero ozone depletion replacement for CFCS and HCFs. Refrigerant R-134a is non-toxic and colourless liquefied gas. R-134a has been developed for a number of refrigeration applications.

Refrigeration lubricant plays a vital role in the lifespan and performance of a refrigerator. It lubricates internal compartments of a refrigerator. Besides, it must be compatible with refrigerant used by the compressor and removes heat generated during compression. Lubrication is vitally important during compression as to maintain the components of compressor in a hermetic environment. It acts as a fluid seal and reduces power consumption of a refrigerator. As to improve the efficiency and reduce the power consumptions of refrigeration system, an evaluation on new lubricant oil has been considered. Therefore, a lubricant based nanofluid which is termed nanolubricant has been introduced.

The nanoparticles increase the thermal conductivity of the lubricant. Often, the “fluid” doing the lubrication is a combination of both lubricant and refrigerant called “working fluid”. The influence of nanoparticles in improving the working fluid properties is proved. The nanoparticles caused a heat transfer enhancement relative to the heat transfer of pure R-142a/polyester (99.5%/0.5) of between 50% and 275% for the nanolubricant mass fraction of 0.5% (M .A. kedzierski and M.Gong, 2007). The mineral lubricant included added nanoparticles Al_2O_3 of 0.01% by weight reduce power consumption by about 2.4% and the COP was increased by 4.4% (Jwo et al, 2009).

For the AL_2O_3 nanoparticles, they are many studies on this oxide particle based nanofluids due to its availability (Das et al., 2006). AL_2O_3 nanoparticles have the advantage of a well-established, successful dispersion technology and being relatively inert with respect to lubricate compressor parts. The AL_2O_3 nanoparticles caused a heat transfer enhancement relative to the heat transfer of pure R-134a/polyester was proved by M.A. Kedzierski (2011). Figure 2.5 indicated the influence of AL_2O_3 nanoparticles on R-134a.RL68H boiling heat transfer. From the Figure 2.5 below, there is heat transfer enhancement exists where the heat flux ratio is greater than one and the 95% simultaneous confidence intervals do not include the value one.

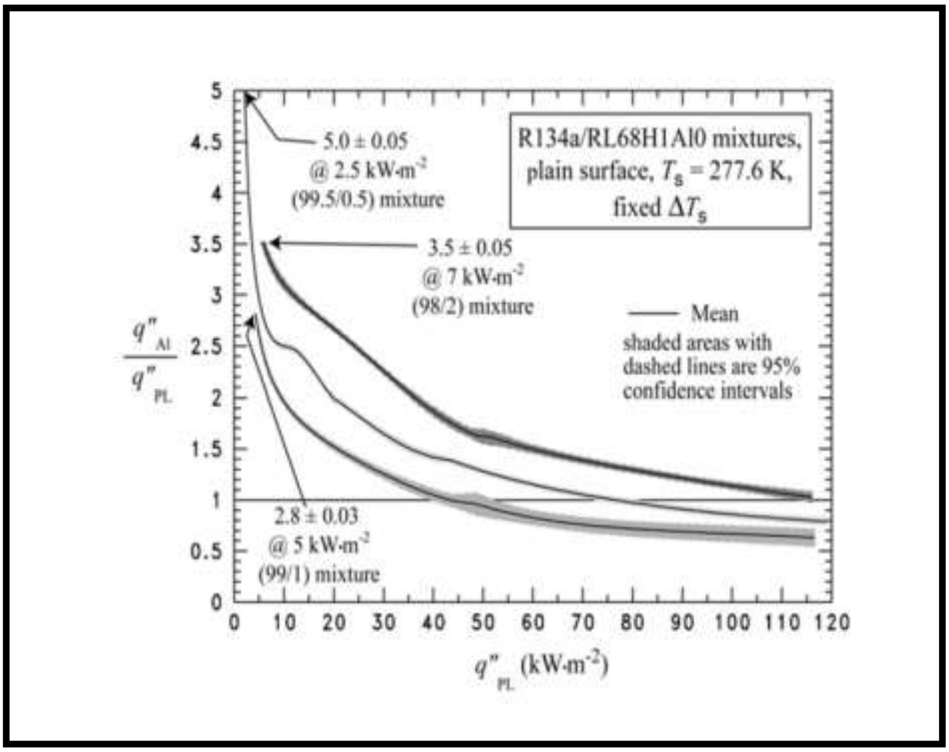


Figure 2.5: Boiling heat flux of R-134a/RL67H1ALO mixtures relative to that of R-134a/RL68H without nanoparticles

Source: M.A.Kidzierski (2011)

2.8 SAMPLE PREPARATION OF NANOFLUIDS

The nanofluids can be prepared via two methods which is the one-step and two-step method. The one step method has been carried out by makes and disperses the nanoparticles directly into a base fluid simultaneously. It is a non-commercial method. The two-step method produced the nanoparticles and dispersed them into a base fluid which normally used for research and industrial applications.

Zhu et al. presented a novel one step chemical method. The CuO nanofluid was synthesized with a wet chemical method (Zhu et al., 2011). The advantage of one-step method is the nanoparticle agglomeration can be minimized whereas the disadvantages is only a low vapor pressure fluids are compatible with such process (X.-Q Wang and Arun S. Mujumdar, 2008).

Commonly, the two-step method is used for preparing nanofluids. Firstly, the nanoparticles are produced as dry powders by chemical or physical methods. Then, the nanosized powder will be dispersed into a fluid in the second processing step with the help of intensive magnetic force agitation, ultrasonic agitation, high-shear mixing, homogenizing, and ball milling (W.Yu and H.Xie, 2012). The ultrasonic equipment is used to disperse the nanoparticles intensively and reduce the agglomeration of particles (X.Q. Wang, 2008). Often, ultrasonic agitation and addition of surfactant into the fluids is used as to minimized the particle agitation and improve dispersion behavior. The two-step method was used in this study due to its ability to work well for oxide nanoparticles (Eastman et al., 1999).

For the sample preparations of nanofluids with different volume concentrations, it can be carried out by using a sensitive mass balance with an accuracy of 0.1 mg. The volume fraction of the powder is calculated from the weight of dry powder using the density provided by the supplier and the total volume of the suspension. Eq. (2.9) is used for calculating the volume fraction of the powder (F.Duan).

$$vol\,\% = [\left(\frac{m}{\rho}\right) \div \left(\frac{m}{\rho} + Vbasefluid\right)] \tag{2.9}$$

where m = mass of the nanoparticles

ρ = density of the nanoparticles

2.9 ADVANTAGES AND DISADVANTAGES OF NANOFLUIDS

In recent years, nanofluids have attracted more and more attention. Nanofluids have been found to possess enhanced thermo-physical properties such as thermal conductivity, thermal diffusivity, viscosity, and convective heat transfer coefficients compared to those of base fluids like oil or water. It has demonstrated great potential applications in many fields. Generally, nanofluids possess the following advantages and disadvantages (Choi et al., 2007):

i. Nanofluids have enhanced ability to conduct heat transfer applications due to larger surface area. Therefore, there is more heat transfer surface between particles and fluids.
ii. The dispersion stability of nanofluids is higher due to predominant Brownian motion of particles.
iii. Nanofluids are ideal for micro-channel applications without clogging.
iv. Nanofluids reduce chances of erosion of components such as heat exchanger due to their small size. There is an increase in thermal conductivity with decreasing particle size.
v. Nanofluids reduced the pumping power due to an increase in thermal conductivity.

In addition, there are several literature review highlighted the advantages of the use of nanoparticles in refrigeration system. The nanoparticles enhance the solubility for the refrigerant of typed HFC134a (S.-S. Bi, 2008). S.-S. Bi et al. also reported the refrigerator performance with 9.6% less energy used with 0.5 g/L TiO_2-R600a nano-refrigerant (S.S.Bi, 2011). Besides, the freezing capacity is higher and the power consumption is reduces by 25% when PEO oil is replaces by a mixture of mineral oil and alumina nanoparticles (N.Subramani, 2011).

Another advantage of nanoparticles is their dispersion in the base oil did not affect significantly the solubility, suggesting the independence of the thermodynamics properties of the oil from the presence of nanoparticles (S.Bobbo, 2010). The heat transfer coefficient of refrigerant-based nanofluid is larger that of pure refrigerant (Peng et al., 2009)

Despite the advantages mentioned above, there are several disadvantages of nanofluids which hinder the application of nanofluids. One of the main challenges is their dispersion stability which may influences the properties of nanofluids for application. There is physical and chemical treatment which had been conducted to get stable nanofluids:

i. An addition of surfactant changes the wetting or adhesion behavior which helps in reducing their tendency to agglomerate (Aida Nasiri et al., 2011).
ii. Surface modification of the suspended particle
iii. Applying strong force on the clusters of the suspended particles

Besides, the dispersing agent and surface active agent can disperse fine particles of hydrophobic materials in an aqueous solution (Y.Hwang, 2006). J.L proposed a visual inspection of nanofluids over an extended period of time as to investigate the stability of nanofluid. Unfortunately, it found that there are some settlement and a concentration gradient in both 1% and 2% Al_2O_3 nanofluids taken shortly after initial mixing and 30days later. Such effects can cause long term degradation in thermal performance of nanofluids for application.

Secondly, the viscosity of nanoparticle-water suspensions increases in accordance with increasing particle concentration in the suspension. Therefore, the mass fraction of particle cannot be increase unlimitedly (S. Wu et al, 2009). The influence of viscosity is more profound on pumping power and heat transfer in comparison with other thermo-physical properties of nanofluids (P.K. Namburu et al., 2009).

Thirdly, the production cost of nanofluids inevitably is higher than that of conventional fluid (J. Lee and I. Mudawar, 2007). The equipment used to manufacture them is one-of-a-kind. Therefore, the fluid must be made affordable if they are ever to see widespread use.

The specific heat of nanofluids is lower than that of basefluid. When the specific heat of refrigerant employed in refrigeration system is low, the compression work will be high with increased discharge temperature and reduced thermal efficiency (R.Prapainop and K O Suen, 2012). The table 2.1 illustrated the specific heat of the base fluid and nanofluids of CuO, SiO2 and Al_2O_3 used in the numerical computations at inlet temperature of 293K. The specific heat of CuO, SiO_2 and Al_2O_3 nanofluid exhibits lower specific heat than that of basefluids (P.K. Nambru et al., 2009)..

Table 2.1: Specific heat of the nanofluids used in the numerical computations at inlet temperature of 293K

Type of Fluid	Volume Concentration (%)	Specific Heat (J/kg K)
EG/water(base fluid)	0	3084.00
CuO/ EGwater	1	2942.61
CuO/ EGwater	2	2813.63
CuO/ EGwater	3	2695.50
CuO/ EGwater	4	2586.89
CuO/ EGwater	5	2486.70
CuO/ EGwater	6	2393.99
SiO2/EG water (20nm)	6	2814.09
SiO2/EG water (50nm)	6	2814.09
SiO2/EG water (100nm)	6	2814.09
Al2O3/EG water	6	2645.35

Source: Praveen K. Namburu et al., (2009)

From the literature review, there is a significant increase of nanofluids pressure drop compared to basefluid. Jaeseon.L and Issam Nudawar reported that an increased in nanoparticle concentration caused an increase in single-phase pressure drop compared to pure fluids at the same Reynolds number.

2.10 DEVELOPMENT OF NANOPARTICLES IN REFRIGERATION SYSTEM

Table 2.2 illustrated the summary of previous research on development of nanoparticles in refrigeration system. The application of nanofluids in refrigeration system is compromised with enhanced heat transfer coefficient. The power consumption of a domestic refrigerator is expected to reduce with the presence of nano additives as a lubricant in refrigerant employed in refrigerator.

Table 2.2: Summary of previous research on development of nanoparticles in refrigeration system

Year	Author/s	Nano-particles	Size of Nano-particles	Refrigerant Type	% of Concentration	Preparation Method	Performance
2007	Kedzierski and Gong, NIST	CuO	30 nm	R134/ Polyolester	1% and 0.5%	By volume	For the 0.5% nanolubricant mass fraction, the nanoparticles caused a heat transfer enhancement relative to the heat transfer of pure R134a/polyolester (99.5/0.5) of between 50 % and 275 %.
2013	Sreejith. K	CuO	-	R134a/Polyolester & R-134a/SUNISO 3GS Oil	-	-	The energy consumption of the HFC134a refrigerator using CuO nanoparticle-lubricant mixture reduced the energy consumption of the household refrigerator system between 12% and 19% for air-cooled condenser and between 9% and 14% for water-cooled condenser and on using SUNISO 3GS mineral oil as the lubricant the energy consumption reduced between 11% and 15% for air-cooled condenser and between 8% and 11% for water-cooled condenser on various load conditions.
2011	Kedzierski	Al_2O_3	10 nm	R134a	0.5, 1 and 2%	By weight	The average heat flux improvement for heat fluxes less than 40kW/m2 was approximately 105%, 49%, and 155% for the 0.5%, the 1%, and the 2% mass fractions, respectively.

Table 2.2: Continued

Year	Author/s	Nano-particles	Size of Nano-particles	Refrigerant Type	% of Concentration	Preparation Method	Performance
2007	Park and Jung	CNT and TiO_2	20 nm OD	R123 and R134a	1%	By volume	Enhanced heat transfer coefficient up to 36.6%.
2009	Jwo et al.	Al_2O_3	-	Hydrocarbon/ mineral oil	0.01%	By weight	Power consumption is reduces about 2.4% increase COP was about 4.4%.
2009	Jiang et al	CNT	15 and 80 nm diameter, 1.5 and 10 nm length	R113	0.2, 0.4, 0.6, 0.8 and 1%	By volume	At volume fraction is 1.0%, the measured thermal conductivities of four kinds of CNT/R113 nano-refrigerants increase 82%, 104%, 43% and 50% respectively.
2009	Peng et al.	CuO	-	R113	0 – 0.5%	By weight	Maximum enhancement of heat transfer coefficient is 29.7% is obtained.
2010	Peng et al.	Diamond	10 nm	R113	0 – 5%	By weight	The nucleate pool boiling heat transfer coefficient of R113/oil mixture with diamond nanoparticles is increase by 63.4%.
2011	Peng et al.	Cu	20, 50 and 80 nm	R113	0 – 5%	By weight	The nucleate pool boiling heat transfer coefficient of R113/oil mixture with Cu nanoparticles is enhanced by a maximum of 23.8% with the decrease of nanoparticle size from 80 to 20 nm.
2009	Trisaksri and Wong-wises	TiO_2	Average 21 nm	R141b	0.01, 0.03 and 0.05%	By volume	Nucleate pool boiling heat transfer deteriorated with increasing particle concentrations.

2.11 CALIBRATION OF TEMPERATURE SENSOR

Commonly, the temperature sensors used is thermocouple which consists of two different metal wires connected together. The thermocouple need to be calibrated as to obtain an accurate temperature reading the ANSI/ASHRAE STANDARD 150-2000. The calibration can be carried out in a fixed-point environment. There are three types of fixed point baths which can be set as a reference standard. There are freezing point, boiling point and triple point baths (R.P.Benedict, 1966). Figure 2.5 is the triple point realization of phase diagram for H_2O. However, the temperature difference between the sensor readings and thermometer must be within the following limits as shown in the table 3.2 below.

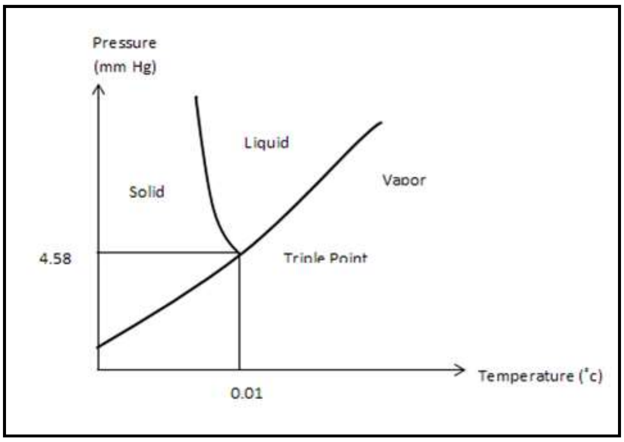


Figure 2.6: Phase Diagram for H_2O

Source: R.P.Benedict (1966)

Table 2.3: Accuracy of temperature sensors

Types	**Temperature**	**Temperature Difference**
Accuracy	±0.15˚C(±0.3˚F)	±0.10˚C(±0.2˚F)
Precision	±0.10˚C(±0.2˚F)	±0.075˚C(±0.15˚F)
Resolution	±0.05˚C(±0.1˚F)	±0.05˚C(±0.1˚F)

Source: ANSI/ASHRAE STANDARD 150-2000\

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

Under this chapter, the detail planning for this project will be proposed and discuss. The approach that will be implemented in this study was an experimental based. Firstly, there was calibration testing on thermocouple of model SR 630 which used for temperature measurement. Then, there was identification on location for the temperature and pressure measurement. This was essentially important on obtaining the most accurate data for optimum performance of refrigerator. The measuring equipment for power consumption was power quality analyzer of KEW 6310 was installed. Then, the method on preparing the refrigerant R-134a nano-polyester lubricant had been determined. The experimental setup with and without nano additive was proposed as below. The preliminary experiment was based on refrigeration HFC134a/POE oil system and the second one was based on refrigeration HFC134a/Al_2O_3/POE oil system. All the data obtained via this experiment was recorded manually in the table prepared. The experimental results obtained will be calculated and compared in term of COP and power consumption. A summary of methodology is as shown in Figure 3.1 below.

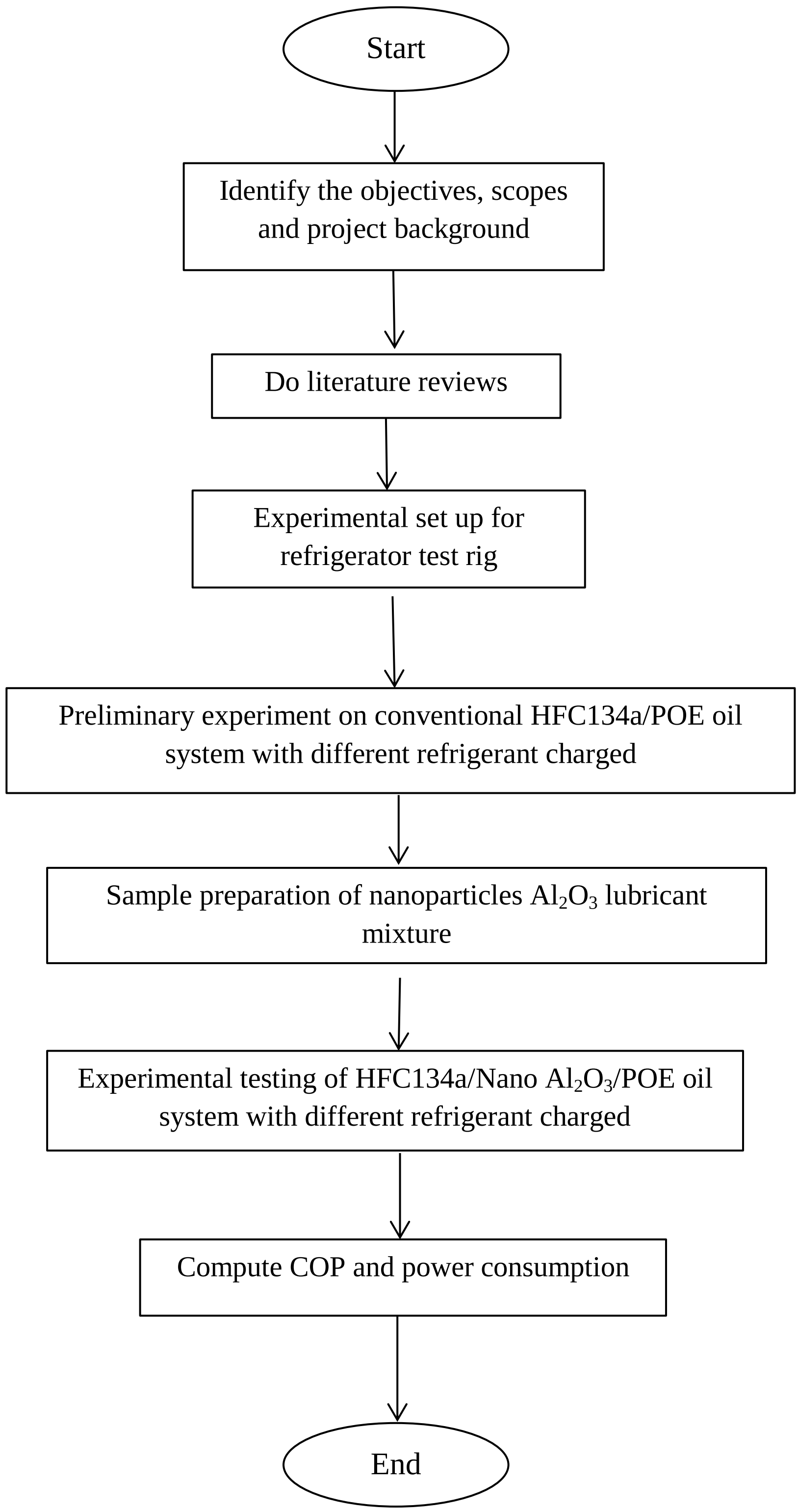


Figure 3.1: A flow chart showing a summary of research methodology

The importance of study in this project was to investigate the overall performance of refrigeration system operating with or without nano additives with different refrigerant charged.

3.2 EXPERIMENTAL SETUP OF TEST RIG

The refrigeration system test rig used in this research was a mini bar refrigerator of model UPSON. The Table 3.1 as shown below stated the technical specification of this refrigeration system test rig in details. The test rig was used for testing on refrigerant with and without nano additives. The Figure 3.2 as shown is the mini bar refrigerator which used as the test rig with completion of four thermocouple and two pressure gauges. For measurement on power consumption, power quality analyzer was used in this research studies.

Table 3.1: Technical specification of the refrigeration system test rig of a mini bar refrigerator

Properties	Specifications
Model	UPSON
Model number	URF-M5
Rated voltage	240V
Electricity consumption	0.5kWh/24h
Power	90W
Weight	19kg
Total effective volume	47L
Dimension	45cm x 45cm x 45cm
Refrigerant	R-134a
Expansion valve	TXV (Thermostatic Expansion Valve)
Compressor model	AS 30
Compressor motor type	RSIP
Compressor voltage	220-240 v/50HZ
Compressor oil charge ml	190
Place of origin	Guangzhou Refrigeration Co. LTD.

Figure 3.2: Experimental test rig of UPSON mini bar domestic refrigerator

3.2.1 Procedure for calibration testing

According to ANSI/ASHRAE STANDARD 150-2000, the temperature sensors used shall be calibrated. The calibration was carried out through freezing point, boiling and triple point baths with a reference standard. For the calibration of ice bath, the reference reading was 0 ˚c (32˚F). Figure 3.3 as shown below was the triple point bath.

a) For Triple Point bath:

1. An insulated container was filling up with water of a volume of at least 0.5 liter.
2. The temperature sensor was removed from its installed location on the experimental refrigerator test rig.3. The temperature sensor of copper wire typed was immersed in the water bath.
4. The thermometer was immersed in the water bath at the same time as well.
5. The temperature reading displayed on the thermocouple was allowed to stabilize at the temperature of the bath.
6. The sensor temperature reading was taken within an interval of one minute.
7. The data was recorded manually in the table prepared for a period of at least five minutes continuously.

(b) For Freezing Point Bath:

1. An insulated container was filling up with crushed of fake ice in water of a volume of at least 0.5 liter.
2. The temperature sensor was removed from its installed location on the experimental refrigerator test rig.
3. The temperature sensor of copper wire typed was immersed in the ice bath.
4. The thermometer was immersed in the water bath at the same time as well.
5. The temperature reading displayed on the thermocouple was allowed to stabilize at the temperature of the bath.
6. The sensor temperature reading was taken within an interval of one minute.
7. The data was recorded manually in the table prepared for a period of at least five minutes continuously.

(c) For Boiling Point Bath:

1. An insulated container was filling up with boiled hot water of a volume of at least 0.5 liter.
2. The temperature sensor was removed from its installed location on the experimental refrigerator test rig.

1. The temperature sensor of copper wire typed was immersed in the water bath.
2. The thermometer was immersed in the water bath at the same time as well.
3. The temperature reading displayed on the thermocouple was allowed to stabilize at the temperature of the bath.
4. The sensor temperature reading was taken within an interval of one minute.
5. The data was recorded manually in the table prepared for a period of at least five minutes continuously.

Figure 3.3: Triple Point Bath

3.2.2 Temperature and Pressure Measurement

Thermocouple as shown in the Figure 3.4(a) was used as the temperature measurement device in this study. The Table 3.2 as shown below stated the technical specification of this thermocouple in details.

Table 3.2: Technical specification of the thermocouple used for temperature measurement

Properties	Specifications
Model	Standard Research System
Model number	SR630
Channels	16
Thermocouple Types	B, E, J, K, R, S,T
Display Units	Degree C

The thermocouples were installed at three locations of the refrigerator compartment as shown in the Figure 3.6. The attachment of copper wire on three different locations for the temperature measurement was show in the Figure 3.4(b) below.

(a) (b)

Figure 3.4: Temperature measurement (a) Thermocouple of model SR630 (b) Copper wire attachment

For pressure measurement in the refrigeration system, bourdon tube typed pressure gauge as shown in the figure 3.5 was used. There were two pressure gauges attached on wooden panel of the test rig for measurement of pressure. The measured pressure at point 1 was suction pressure and point 2 was discharged pressure as shown in the schematic diagram of Figure 2.2. This was done as to avoid the pressure gauge from vibrating during experimental testing.

Figure 3.5: Pressure gauge

The sensing element of the pressure gauge will be placed on the line of suction and discharge of compressor as shown in the Figure 3.6 below. Bourdon tube pressure gauge was measuring the suction (P_s) and discharged (P_d) pressure of the compressor

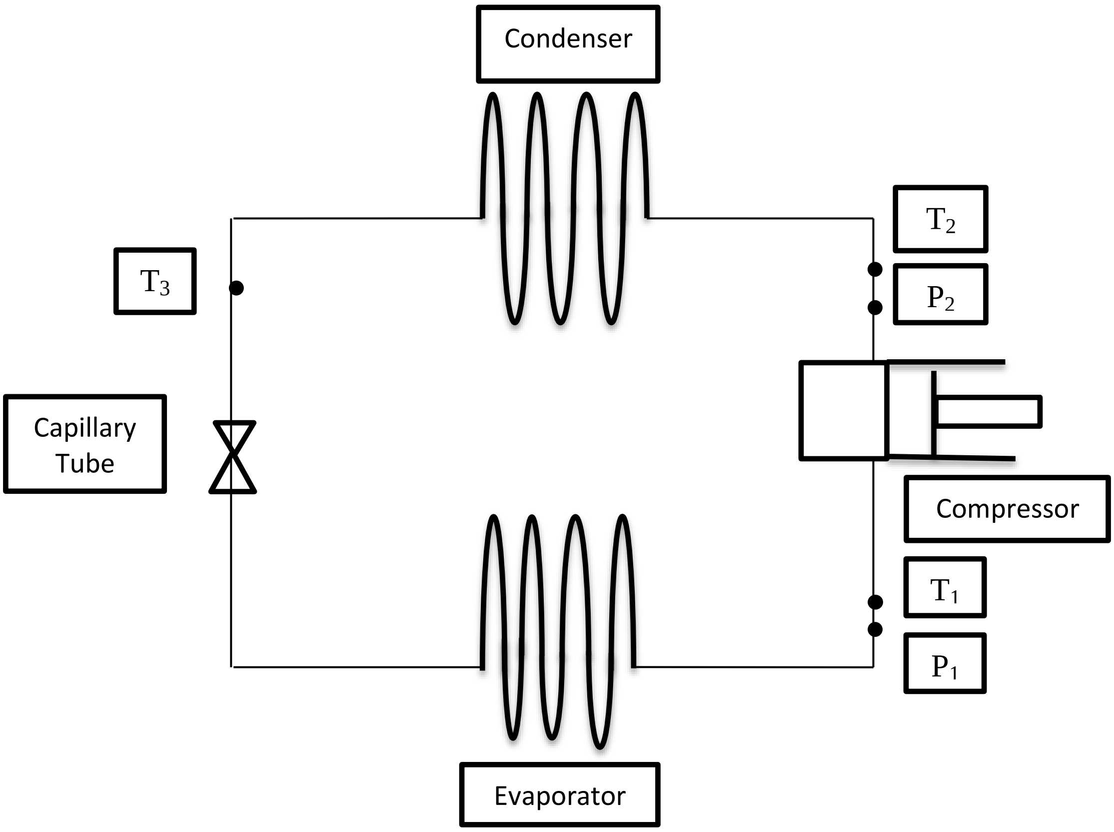


Figure 3.6: Schematic diagram of measurement point for temperature and pressure.

Where T_1 = Temperature measurement at point 1

P_1 = Pressure measurement at suction

T_2 = Temperature measurement at point 2

P_2 = Pressure measurement at discharge

T_3 = Temperature measurement at point 3

3.2.3 Measurement of Power Consumption

The power quality analyzer of KEW 6310as shown in the Figure 3.7(a) was used to measure the power consumption of the experimental refrigeration test rig. The wiring configuration was single phase 2 wire (1ch) which was 1P2Wx1. Figure 3.7(b) was the wiring configuration of single phase 2 wire (1ch).

(a) (b)

Figure 3.7: Power consumption measurement (a) Power quality analyzer (b) Wiring configuration (KEW 6310)

3.3 EQUIPMENT AND TOOLS

The necessary equipment for evacuation and charging process were charging kit, vacuum pump and refrigerant R-134a as shown in the Figure 3.8(a), (b) and (c). The tools which used for attachment of copper wire on thermocouple were screwdriver. Other necessary tools used were test pen, pliers, wire stripper, monkey wrench, adjustable spanner and insulation pipe.

(a)

(b)

(c)

Figure 3.8: Equipment and tools (a) Charging manifold (b) Vacuum pump (c) Refrigerant R-134a

3.4 SAMPLE PREPARATION OF NANOPARTICLES Al_2O_3 LUBRICANT MIXTURE

For the preparation of nanoparticles Al_2O_3 lubricant mixture, there were two techniques. There were single step and two step techniques. In this study, the preparation method employed for refrigerant R-134a polyester lubricant with nano additives is two step techniques.

3.4.1 Two-step method

In two step method, the nanoparticles are directly mixed in the base fluid and thoroughly stirred. After estimating the amount of nanoparticles required for preparation of nanoparticles Al_2O_3 for a given volume concentration, the nanoparticles are mixed in the base fluid of POE lubricant oil. The alumina (Al_2O_3) with an average diameter size of 13nm was used in this present experimental work.

The POE oil as shown in the Figure 3.9(a) was measured with a small beaker with volume of 190 milliliters (ml). For the nanoparticles Al_2O_3 in powder form, it was weighted at 1.4744 gram (g). Then, the nanoparticles Al_2O_3 was dispersed in the measured POE oil. Figure (a) was the base fluid which is the POE oil used for lubricity in this experiment.

Figure 3.9(b) was the photographic view of the nanoparticles Al_2O_3 as seen by the naked eyes. The preparation of nanoparticles Al_2O_3 lubricant mixture was an important stage. It was prepared in a systematic and careful manner. The nanoparticles Al_2O_3 was diluted with POE oil as a based fluid at concentrations, Ø of 0.2% by volume.

(a) (b)

Figure 3.9: Base fluid and nanoparticles (a) POE oil (b) Photographic view of Al_2O_3

A stable nanoparticles lubricant mixture with uniform particle dispersion was required. Therefore, the test samples of nanoparticles Al_2O_3 lubricant mixture used were subjected to magnetic stirring process followed by ultrasonic homogenization for about 2 hours. A magnetic stirrer was a laboratory device that employs a rotating magnetic field to cause a stir bar (also called flea) immersed in a liquid to spin very quickly and thus stirring it. Meanwhile, a hotplate at the bottom provides heating to catalyze the process when necessary. Ultrasonic homogenizer worked by generating intense sonic pressure waves in a liquid media. Figure 3.10 was the hotplate stirrer and Figure 3.11 was the ultrasonic homogenizer used in the process of homogenization.

Figure 3.10: Hotplate Stirrer

Figure 3.11: Ultrasonic Homogenizer

For the density of nanoparticles Al_2O_3 under investigation are obtained using the density correlation equation (Eq.2.9) developed by F.duan for nanofluids.

3.4.2 Experimental Step

1. Firstly, the polyester lubricant mineral based oil was poured into beaker as whon in the Figure 3.12(a).
2. Then, the nanoparticle which is the alumina, Al_2O_3 of an oxide based weighted 1.4744g (as shown in the Figure 3.12(b) was added in it.
3. The beaker which contained alumina nano particles and polyester lubricant oil was placed on the hotplate stirrer (as shown in Figure 3.12(c)) after 10 minutes.
4. The hotplate stirrer was press “ON” .
5. Then, it was followed by ultrasonic vibration as shown in the Figure 3.13(d) for about 1 hour.
6. The processor of ultrasonic homogenizer was pressed “ON”.
7. The probe size of value 1.5 was set as 1.5.
8. The total of time was set as 900 seconds (15 minutes).
9. The pulser on time which was the duration for ultrasonic processor to shoot the fluid was set as 1 seconds.
10. The pulser off time which was the resting point taken before the ultrasonic processor shoot back the fluid was set as 2 seconds.
11. The sample temperature can be withstand by polyester lubricant oil of R-134a was set at 10˚C.
12. The power used for ultrasonic homogenizer was 475 watts.
13. The beaker was then removed from ultrasonic homogenizer.
14. The nanoparticles Al_2O_3 lubricant mixture was ready.

(a) **(b)**

(c) **(d)**

Figure 3.12: (a): POE oil (b) Weighing Scale (c): Stirring (d) Ultrasonification

3.5 EXPERIMENTAL FLOW CHART

The Figure 3.13 was the experimental flow chart for this study.

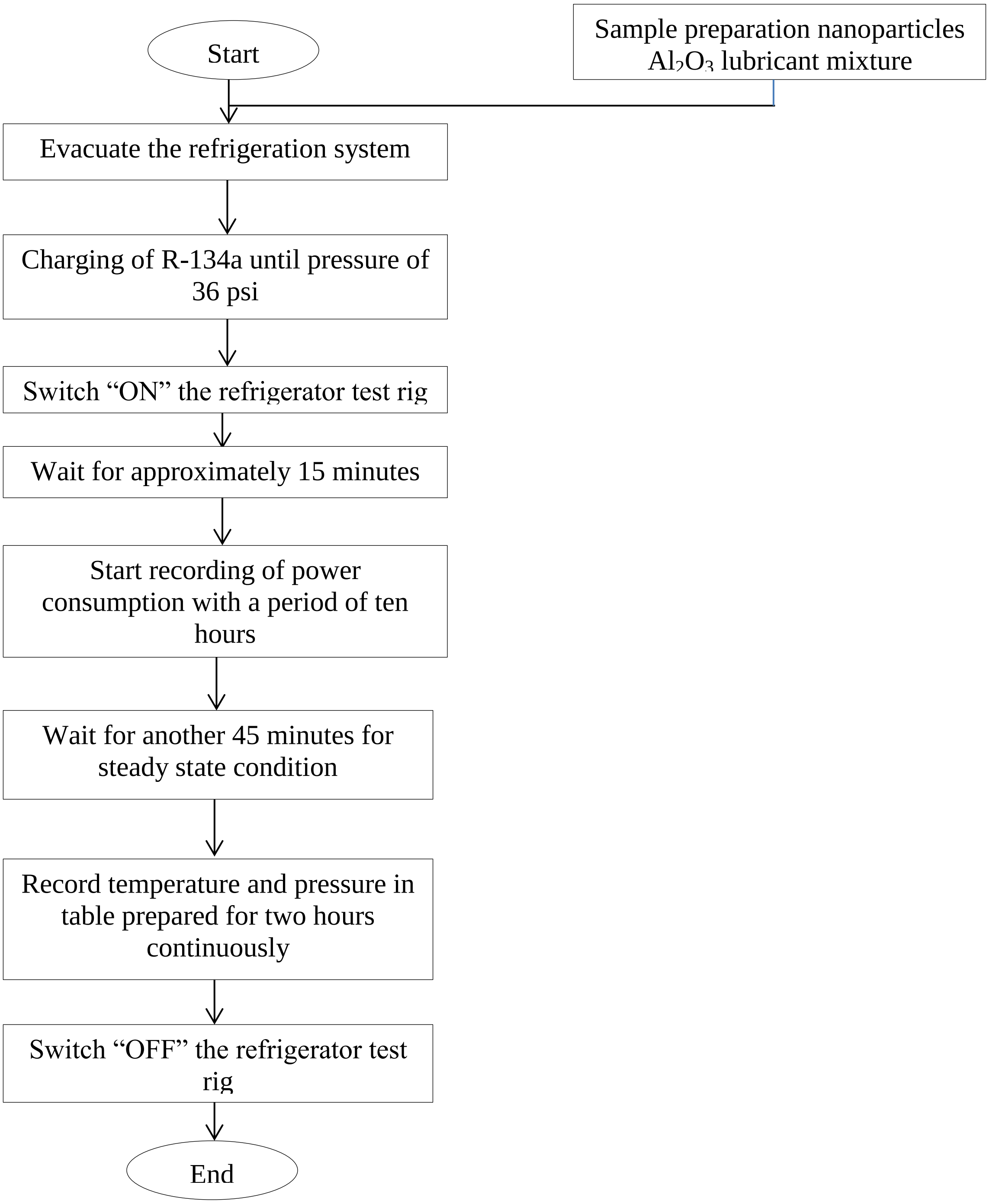


Figure 3.13: Experimental flow chart

3.6 EXPERIMENTAL PROCEDURE

The experimental testing will be carried out in two ways. Firstly, there was a preliminary experiment with existing refrigerant R-134a polyester lubricant oil. Secondly, there was another experimental testing with addition of nanoparticles in the refrigerant R-134a polyester lubricant oil. Both test setup was a necessary as to compare the performance of existing and newly developed testing. The first experimental testing will be used as the frame of reference for the newly developed experimental testing with nanoparticles Al_2O_3 lubricant mixture.

3.6.1 Experimental procedure without nanoparticles Al_2O_3

The experimental steps for refrigeration system operating without nano additives were illustrated in details as below.

1. Firstly, the pressure gauges were installed. The pressure sensor was attached on the point of interest, the suction, P_s (P_1) and discharged, P_d (P_2) side as shown in the schematic diagram of figure 3.5.
2. Secondly, the copper wire was fixed on the point of interest for temperature measurement at three different locations which is T_1, T_2 and T_3.
3. Another end of the copper wire was free and only attached on the thermocouple of model SR630 when the data for temperature at different points needed to be collected.
4. The power quality analyzer of model KEW 6310 was installed. The arrow mark on the clamp sensor should be points towards to load side. The wiring configuration employed for this experimental test rig was wiring method for single phase 2 wire (1 channel).
5. The evacuation process will be carried out with vacuum pump and charging manifold as to remove dust and moisture air in the refrigeration system. There are 3 ports at the charging manifold. Right port is connected to the charging port at the compressor, middle port is connected to the refrigerant cylinder and the left port is connected to the vacuum pump.
6. Open both the valves of the low and high pressure gauges on the left and right charging manifold while the valve of the refrigerant cylinder must be ensured closed at this state.

7. The vacuum pump was switched on until the suction pressure gauge shows a vacuum reading of 76 cmHg (1 std. atm).
8. Disconnect the vacuum pump from the system by disconnecting the left port of the charging manifold from the vacuum pump. The left port of charging manifold is free of connection now.
9. Next, the refrigerant R-134a cylinder will be connected to the middle port of the charging manifold as mentioned earlier.
10. Open the middle manifold valve and valve on the refrigerant cylinder while both the right charging manifold valves remain closed.
11. Slowly open the right side manifold valve until the pressure read about with refrigerant charge of 36 psi.
12. The leakage on tubing checked using bubble soap or snoop.
13. The experimental test rig was switched “ON”.
14. The power quality analyzer KEW6310 was pressed “ON”.
15. Under “save”, the date and time for starting and ending the experimental testing was set.
16. The power quality analyzer KEW6310 was starting the data recording after 15 minutes.
17. Another end of the copper wire was attached on the channel 1, 2 and 3 of thermocouple model SR630.
18. The experimental test rig was left running for at least 45 minutes before data collection on temperature and pressure.
19. The value displayed on the thermocouple model SR630 on channel 1, 2 and 3 was recorded in the table prepared for every 5 minutes.
20. The pressure on the pressure gauge was also recorded in the table prepared for every 5 minutes.
21. There were 24 sets of data were taken for each test run with an interval of 5 minutes. Then the average will be taken to for temperature and pressure reading.
22. For the power consumption of the experimental test rig, the data was recorded automatically according to the timer set previously.
23. The step 9-22was repeated for refrigerant charged of 38 psi, 40 psi, 42 psi and 44 psi.
24. The data collected was used as a frame of reference for another experimental testing with nanoparticles Al_2O_3.
25. The experimental test rig was switched “OFF”.

26. The conventional POE oil as shown in the Figure 3.14 below was poured out from the compressor upon completion for all experimental testing with different refrigerant charged.

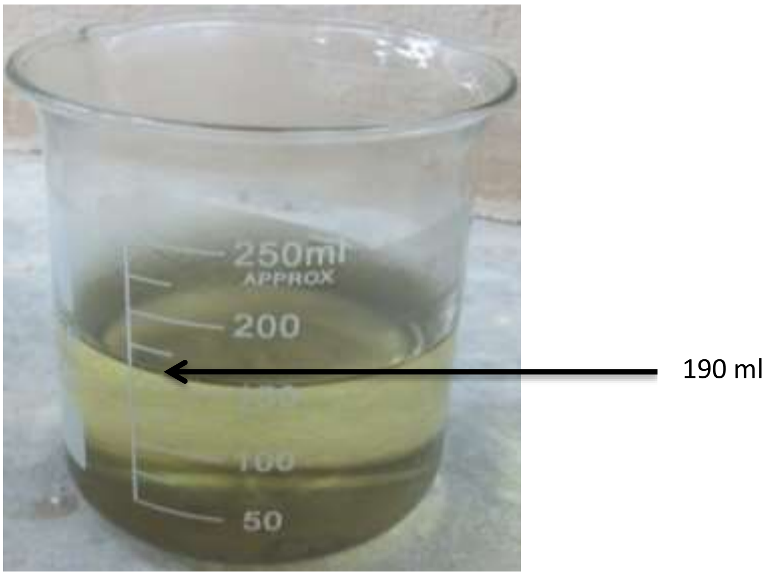


Figure 3.14: Conventional POE oil

3.6.2 Experimental procedure with nanoparticles Al_2O_3

The experimental steps for refrigeration system operating with nanoparticles Al_2O_3 lubricant mixture were illustrated in details as below.

1. Firstly, the pressure gauges were installed. The pressure sensor was attached on the point of interest, the suction, P_s (P_1) and discharged, P_d (P_2) side as shown in the schematic diagram of figure 3.5.
2. Secondly, the copper wire was fixed on the pint of interest for temperature measurement at three different locations which is T_1, T_2 and T_3.
3. Another end of the copper wire was free and only attached on the thermocouple of model SR630 when the data for temperature at different points needed to be collected.
4. The power quality analyzer of model KEW 6310 was installed. The arrow mark on the clamp sensor should be points towards to load side. The wiring configuration employed for this experimental test rig was wiring method for single phase 2 wire (1 channel).

5. The evacuation process will be carried out with vacuum pump and charging kit. This is done as to remove dust and moisture air in the refrigeration system. The evacuation was considered done when the suction pressure gauge shows a vacuum reading of 76 cmHg (1 std. atm).
6. Open both the valves of the low and high pressure gauges on the left and right charging manifold while the valve of the refrigerant cylinder must be ensured closed at this state.
7. The vacuum pump was switched on until the suction pressure gauge shows a vacuum reading of 76 cmHg (1 std. atm).
8. Disconnect the vacuum pump from the system by disconnecting the left port of the charging manifold from the vacuum pump. The left port of charging manifold is free of connection now.
9. Next, the refrigerant R-134a cylinder will be connected to the middle port of the charging manifold as mentioned earlier.
10. Open the middle manifold valve and valve on the refrigerant cylinder while both the right charging manifold valves remain closed.
11. Slowly open the right side manifold valve until the pressure read about with refrigerant charge of 36 psi.
12. The nanoparticles Al_2O_3 lubricant mixture as shown in Figure 3.16 was placed inside the compressor before experimental testing with a funnel.
13. The leakage on tubing checked using bubble soap or snoop.
14. The power quality analyzer KEW6310 was pressed “ON”.
15. Under “save”, the date and time for starting and ending the experimental testing was set.
16. The power quality analyzer KEW6310 was starting the data recording after 15 minutes.
17. Another end of the copper wire was attached on the channel 1, 2 and 3 of thermocouple model SR630.
18. The experimental test rig was left running for at least 45 minutes before data collection on temperature and pressure.
19. The value displayed on the thermocouple model SR630 on channel 1, 2 and 3 was recorded in the table prepared for every 5 minutes.
20. The pressure on the pressure gauge was also recorded in the table prepared for every 5 minutes.
21. There were 24 sets of data were taken for each test run with an interval of 5 minutes. Then the average will be taken to for temperature and pressure reading.

22. For the power consumption of the experimental test rig, the data was recorded automatically according to the timer set previously.
23. The step 9-22 was repeated for refrigerant charged of 38 psi, 40 psi, 42 psi and 44 psi.
24. The experimental test rig was switched "OFF".
25. The data collected will be analyzed and compare with the previous experimental testing operated without nanoparticles Al_2O_3.
26. The nanoparticles Al_2O_3 lubricant mixture as shown in the Figure 3.15 was poured out from the compressor upon completion for all experimental testing with different refrigerant charged.

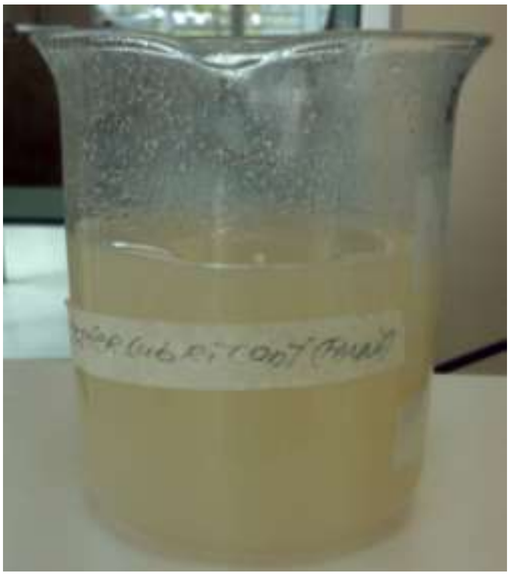


Figure 3.15: Nanoparticles Al_2O_3 lubricant mixture

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

In this chapter, the results of the experiments will be analyzed with the data collected throughout the experiments. This study will be focused on collecting the data in term of pressure, temperature and energy consumption of the refrigeration HFC134a/POE oil system with or without nanoparticles alumina (Al_2O_3). The quantity of refrigerant charged for optimum refrigeration performance for both experiments will be discussed and analyzed in details. In addition, the overall performance and power consumed by the HFC134a/POE oil system with or without nanoparticles Al_2O_3 will be compared. The dispersion stability of nanoparticles Al_2O_3 will be highlighted at the end of this chapter.

In this study, KLEA 134a, pressure-enthalpy p- h diagram has been adopted as the reference for the enthalpy value. The enthalpy (h) value of each point of the refrigeration process will be determined using p-h diagram. In order to determine the enthalpy value, the temperature and pressure value needed which was determined from the experiments that had been conducted. From the enthalpy value, basic calculations could be computed as to determine the cooling load, Q_L, compressor work, W_{in} and Q_H. The coefficient of performance, COP of the refrigeration system will be determined using equation (2.8). The power consumption was recorded by a power quality analyzer in unit of Wh.

There were three sets of data which were taken for each refrigerant charged ranged from 36 psi to 44 psi for both experiments operating with or without nanoparticles Al_2O_3. All of the data will be collected and tabulated in a table. The average of each data will be obtained in order to determine the enthalpy (h) value of each point and energy consumption for the vapor compression refrigeration system.

The experiments were conducted for a domestic mini bar refrigerator operating with or without nanoparticles Al_2O_3 in the HFC134a/POE oil system. The results and analysis of the subsequent finding will be discussed in details.

4.2 DATA COLLECTION

The experiment was conducted for HFC134a/POE oil system operating with or without nanoparticles Al_2O_3. There are three sets of data taken on sunny day for each refrigerant charged either with or without nanoparticles Al_2O_3.

The resulting data for pressure, temperature and power consumption were tabulated in Table 4.1 and Table 4.2 respectively. For pressure measurement, it was taken at the points of suction and discharge of the reciprocating typed compressor. However, for temperature measurements, the points of interest were at the suction, discharge and before the capillary tube.

The data was taken at approximately same time of the day in which the temperature of surroundings was expected to be most humid. This was to ensure the refrigeration system is operating at almost same ambient conditions with the surroundings.

4.2.1 Refrigeration operating without nano

The experiment was conducted with existing polyester lubricant oil of 190 ml. The refrigerant charged pressure ranged from 36 psi to 44 psi. Table 4.1 was the data collected for refrigeration HFC134a/POE oil system operating without nanoparticles Al_2O_3 in term of pressure, temperature and energy consumption.

Table 4.1: Data for refrigeration system operating without nano alumina

Refrigerant Charged (psi)	Temperature and Pressure						Energy Consumption (wh)
	Experiment	**Suction Pressure, P_1**	**Discharged Pressure, P_2**	**T_1**	**T_2**	**T_3**	
36	1	10	150	31.70	50.20	37.30	389.58
	2	10	150	31.50	50.70	37.50	394.62
	3	10	150	31.60	50.90	37.90	397.79
	Average	10	150	31.60	50.60	37.60	394.0
38	1	10	145	31.60	52.30	36.50	375.95
	2	12.5	147.50	31.50	52.50	36.80	386.35
	3	15	150.0	31.50	52.90	37.10	392.99
	Average	12.5	147.50	31.50	52.60	36.80	385.10
40	1	7.5	145	27.9	49.50	34.50	357.86
	2	7.5	145	27.9	49.40	34.90	363.14
	3	7.5	145	28.0	49.90	34.10	368.23
	Average	7.5	145	27.9	49.60	34.50	363.10
42	1	0	155	29.80	49.8	37.10	325.75
	2	2.5	157.5	29.7	49.9	37.50	326.16
	3	5	160	29.7	50.90	38.30	331.50
	Average	2.5	157.50	29.70	50.20	37.60	327.80
44	1	10	142.5	28.8	46.60	35.20	344.42
	2	10	145	28.90	46.80	35.60	344.58
	3	10	147.50	28.90	47.10	35.60	346.40
	Average	10	145	28.90	46.80	35.50	345.10

From data of Table 4.1, a sample was calculated to convert the pressure units at inlet and outlet of the compressor. The gauge pressure units were converted to absolute pressure in order to determine the enthalpy value at the points of interest.

Sample of calculations for refrigerant charged at 36 psi

Pressure inlet/suction, P1 = (0.76 – 0.10) m Hg

= 0.66 m Hg

Using the equations of pressure, P = ρgH

= (13560 kg/m^3)(9.81 m/s^2)(0.66 m Hg)

= 87.80 kPa

P_1 = 0.88 bar

Pressure discharge,P = 150 psi

= (150 psi) x (6.895×103 Pa)

= 1034.25 kPa

The absolute pressure discharge, P = 101.325 kPa + 1034.25 kPa

= 1135.60 kPa

P_2 = 11.40 bar

4.2.2 Refrigeration operating with nano alumina

The experiment was conducted with polyester lubricant of 190 ml with addition of nanoparticles Al_2O_3. The refrigerant charged pressure ranged from 36 psi to 44 psi. Table 4.2

was the data collected for refrigeration HFC134a/POE oil system operating with nanoparticles Al_2O_3 in term of pressure, temperature and energy consumption.

4.2: Data for refrigeration system operating with nano alumina

Refrigerant Charged (psi)	**Temperature and Pressure**						**Energy Consumption (wh)**
	Experiment	**Suction Pressure, P_1**	**Discharged Pressure, P_2**	**T_1**	**T_2**	**T_3**	
36	1	15	145.0	31.90	53.1	38.2	383.70
	2	15	145.0	32.0	53.2	38.1	388.80
	3	15	145.0	32.0	53.4	38.1	392.00
	Average	15	145.0	32.0	53.2	38.1	388.20
38	1	15	145.0	32.00	51.1	37.6	369.90
	2	15	142.5	32.0	51.3	37.5	380.10
	3	15	142.5	32.0	52.1	37.30	386.70
	Average	15	143.3	32.0	51.5	37.5	378.90
40	1	2.5	155.0	31.8	53.2	38.3	351.80
	2	2.5	155.0	31.8	53.1	37.9	357.00
	3	2.5	155.0	31.9	53.8	38.1	360.90
	Average	2.5	155.0	31.8	53.4	38.1	356.57
42	1	0	157.5	30.80	52.9	38.1	310.50
	2	0	157.5	30.7	53.3	37.9	322.30
	3	0	157.5	30.8	53.7	37.3	330.30
	Average	0	157.5	30.8	53.3	37.8	321.0
44	1	0	157.5	30.7	51.60	38.2	335.20
	2	0	157.5	30.7	51.80	38.0	344.25
	3	0	157.5	30.6	52.4	38.0	337.50
	Average	0	157.5	30.7	52.0	38.1	338.98

Following were the steps to convert the unit of the pressure obtained to International Standard (SI) unit. This was enabling easier referencing for enthalpy value in the latter stage. The gauge pressures of the outlet and inlet compressor were converted to absolute pressure.

Sample of calculations for refrigerant charged at 36 psi

Pressure inlet/suction, P1 = (0.76 – 0.15) m Hg

= 0.61 m Hg

Using the equations of pressure, P = ρgH

= (13560 kg/m^3)(9.81 m/s^2)(0.61 m Hg)

= 811.44 kPa

P_1 = 8.10 bar

Pressure discharge ,P = 145 psi

= (145 psi) x (6.895×103 Pa)

= 999.78 kPa

The absolute pressure discharge, P = 101.325 kPa + 999.78 kPa

= 1101.11 kPa

P_2 = 11.0 bar

4.3 DETERMINATION OF ENTHALPY VALUE

In order to calculate the coefficient of performance (COP) of the refrigeration HFC134a/POE oil system operating with or without nanoparticles Al_2O_3, the enthalpy value of the system at point of interest needs to be determined. According to Equation (2.1) and Equation (2.3), the value of enthalpy is needed in order to calculate the refrigeration cooling capacity and compressor work of the refrigeration system.

From Table 4.1, the pressure, temperature data, the enthalpy at points of interest could be determined. The values of enthalpy determined were tabulated on Table 4.3 for refrigeration HFC134a/POE oil system with nanoparticles Al_2O_3 and Table 4.4 for refrigeration HFC134a/POE oil system without nanoparticles Al_2O_3. The enthalpy value obtained will be used for calculation of refrigeration cooling capacity, compressor work and the COP.

Table 4.3: Enthalpy value for HFC134a/POE oil system with nanoparticles Al_2O_3

Type of Lubricant	**Refrigerant Charged (psi)**	**h_1, kJ/kg**	**h_2, kJ/kg**	**h_3, kJ/kg**	**Q_L, kJ/kg**	**W_{in}, kJ/kg**	**Q_H, kJ/kg**	**COP**
Nano-	36	277.0	335.0	152.50	124.5	58.0	182.5	2.15
alumina	38	279.0	333.0	154.0	125.0	54.0	179.0	2.31
Polyester	40	282.0	333.0	153.0	129.0	51.0	180.0	2.53
Lubricant	42	284.0	333.0	153.0	131.0	49.0	180.0	2.67
By con-centration of 0.2%	44	281.0	331.0	153.0	128.0	50.0	178.0	2.56

Table 4.4: Enthalpy value for HFC134a/POE oil system without nanoparticles Al_2O_3

Type of lubricant	**Refrigerant Charged (psi)**	**h_1, kJ/kg**	**h_2, kJ/kg**	**h_3, kJ/kg**	**Q_L, kJ/kg**	**W_{in}, kJ/kg**	**Q_H, kJ/kg**	**COP**
Polyester	36	277.0	336.0	157.5	119.5	59.0	178.5	2.03
Lubricant	38	276.5	332.0	152.5	124	55.5	179.5	2.23
	40	276.0	328.0	150.0	126.0	52.0	178.0	2.42
	42	283.0	332.5	156.0	127.0	49.5	176.5	2.57
	44	277.0	328.0	150.0	127.0	51.0	178.0	2.49

4.4 DETERMINATION OF COEFFICIENT OF PERFORMANCE (COP)

The coefficient of performance is the measure of performance of the refrigeration HFC134a/POE oil system. COP is the ratio of cooling capacity over the compressor work needed to run the system. It is the amount of refrigeration capacity the refrigerant system can remove at a certain rate of compressor work. The value of cooling capacity, compressor work and COP obtained from calculation for refrigeration HFC134a/POE oil system with or without nanoparticles Al_2O_3 will be tabulated in Table 4.5 and 4.6 for analysis.

Since COP is the ratio of refrigeration capacity divided by the amount of compressor work needed, the value of enthalpy of the cooling load need to be determine. From Figure 2.1 (b), the $p-h$ diagram of the ideal vapor compression refrigeration diagram, the cooling load of the refrigeration system can be obtained by subtracting the enthalpy value at point one with the enthalpy value at point four. Thus, the refrigeration capacity can be calculated using equation (2.1).

Sample of calculation for refrigerant charged at 36 psi

Without nano alumina

The refrigeration capacity, Q_L $= \mathbf{h_1 - h_4}$

$= (277.0 - 157.5)$ kJ/kg

$= 119.5$ kJ/kg

With nano alumina

The refrigeration capacity, Q_L $= \mathbf{h_1 - h_3}$

$= (277.0 - 152.5)$ kJ/kg

$= 124.5$ kJ/kg

Compressor work is the amount of work done required by the system to remove an amount of heat from the refrigerated space. Referring to Figure 2.1, which is the $p-h$ diagram of the ideal vapor compression refrigeration cycle, the compressor work can be determined by subtracting the enthalpy value at point two with the enthalpy value obtained at point one. The compressor work obtained is the amount of work required to remove heat from the refrigerated space. Thus the compressor work can be obtained by utilizing the equation (2.3).

Sample of calculation for refrigerant charged at 36 psi

Without nano alumina

The compressor work required, W_{in} $= \mathbf{h_2 - h_1}$

$= (336.0 - 277.0)$ kJ/kg

$= 59.0$ kJ/kg

Without nano alumina

The compressor work required, $\boldsymbol{W_{in}}$ $\mathbf{= h_2 - h_1}$

$$= (335.0 - 277.0) \text{ kJ/kg}$$

$$= 58.0 \text{ kJ/kg}$$

From the obtained cooling capacity, Q_L, and compressor work, W_{in}, the coefficient of performance (COP) of the refrigeration system can be determined. The COP of the refrigeration system can be calculated using equation (2.8). The COP obtained from the calculations is as follows.

Sample of Calculation for refrigerant charged at 36 psi

Without nano alumina

$$\boldsymbol{COP} = \frac{Q_L}{W_{in}}$$

$$= \frac{119.5}{59.0} \qquad \frac{kJ}{kg}$$

$$\mathbf{= 2.03}$$

Without nano alumina

$$\boldsymbol{COP} = \frac{Q_L}{W_{in}}$$

$$= \frac{124.5}{58.0} \qquad \frac{kJ}{kg}$$

$$\mathbf{= 2.15}$$

Table 4.5: Calculated value of the HFC134a/POE oil system with nanoparticles Al_2O_3

Refrigerant Charged (psi)	Q_L (kJ/kg)	W_{in} (kJ/kg)	Q_H (kJ/kg)	COP
36	124.5	58.0	182.5	2.15
38	125.0	54.0	179.0	2.31
40	129.0	51.0	180.0	2.53
42	131.0	49.0	180.0	2.67
44	128.0	50.0	178.0	2.56

Table 4.6: Calculated value of the HFC134a/POE oil system without nanoparticles Al_2O_3

Refrigerant Charged (psi)	Q_L (kJ/kg)	W_{in} (kJ/kg)	Q_H (kJ/kg)	COP
36	119.50	59	178.50	2.03
38	125.50	55.50	181	2.26
40	126	52	178	2.42
42	127	49.5	176.50	2.57
44	127	51	178	2.49

4.5 DATA ANALYSIS

The analysis of the results was based on the three main parameters that have been studied, mainly the compressor work, W_{in}, coefficient of performance, COP and power consumption.

4.5.1 Compressor work

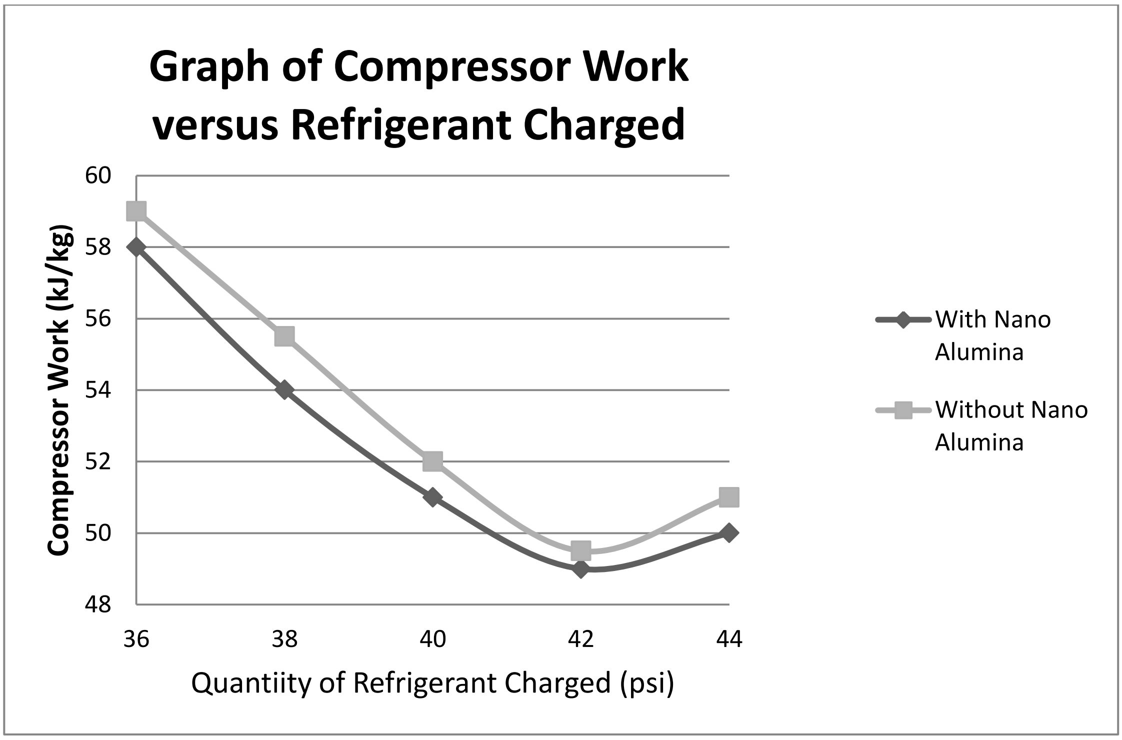


Figure 4.1: Graph of compressor work versus refrigerant charged (psi)

As shown in the graph named figure 4.1 above, the compressor work per unit mass decreased as the quantity of refrigerant charged increased. However, the compressor work per unit mass increased after the refrigerant charged by pressure at 42 psi. Therefore, the refrigeration system was attained its optimum point at refrigerant charged by pressure method at 42 psi.

The evaporation rate in evaporator increased as the refrigerant charged in the refrigeration system increased. This was due to more filling of liquid refrigerant R-134a in the evaporator for the HFC134a/POE oil system. As the liquid fraction in the evaporator increased, the heat transfer coefficient on refrigerant side increased, therefore, the evaporation rate in evaporator increased.

The compressor work per unit mass after the optimum point was increased due to high compression ratio. The liquid backed up in the condenser will flood some of the condensing surface area, causing high head pressures. The TXV (thermal expansion valve) metering devices will still try to maintain the superheating condition of evaporator. However, the evaporator's higher pressure would be caused by the decreased mass flow rate through the compressor from the high compression ratios causing low volumetric efficiencies. The evaporator would have a harder time keeping up with the higher heat loads from the warming entering-air temperature. The TXV will have a tendency to overfeed on its opening stroke due to the high head pressures.

If the system is overcharged more than 10 percent, liquid can enter the suction line and get to the suction valves. This will cause compressor damage and eventually failure.

The work done by the compressor was greater for the conventional HFC134a/POE oil system than the HFC134a/POE oil system with addition of nanoparticles Al_2O_3.This was due to high pressure difference within condenser and evaporator for the refrigeration system operating with conventional HFC134a/POE oil system than the HFC134a/POE oil system with addition of nanoparticles Al_2O_3. The AL_2O_3 nanoparticle-lubricant mixture enhanced the solubility of the HFC134a and POE mineral oil as indicated by the high oil return ratio. It could be reduce the wear, tear and friction in the compressor and thus the work done by the compressor will be lesser.

4.5.2 COP

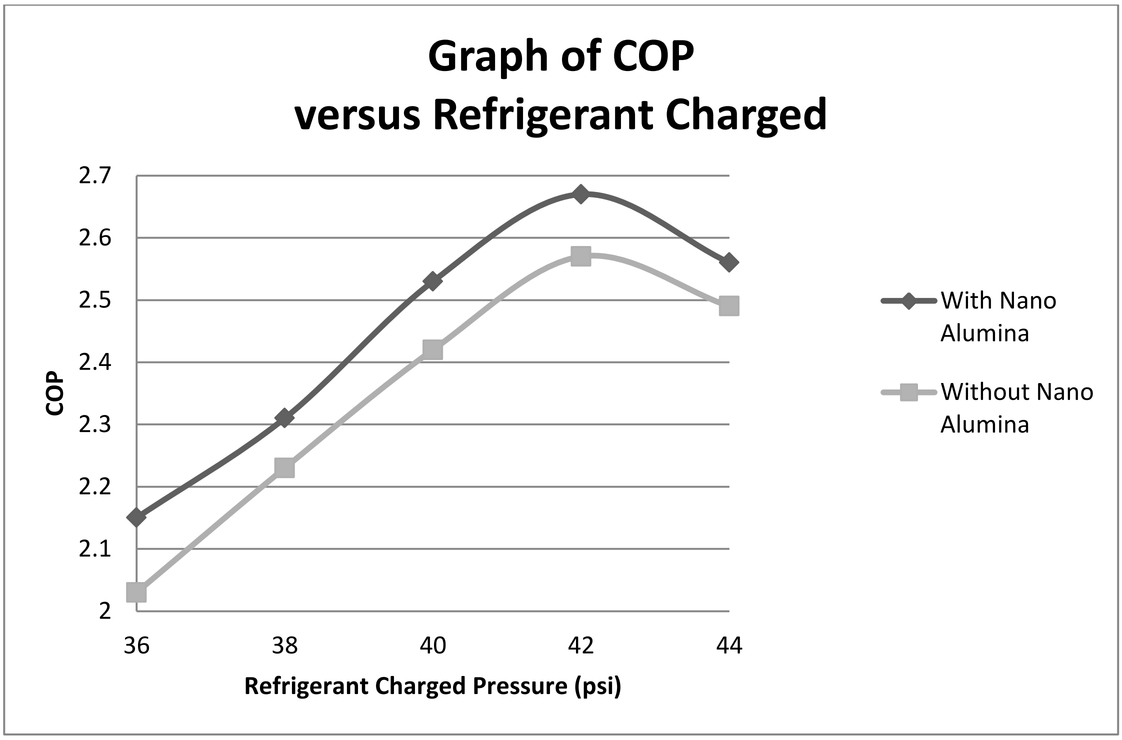


Figure 4.2: Graph of COP versus refrigerant charged (psi)

Figure 4.2 was the graph of COP versus quantity of refrigerant charged (psi). The value for coefficient of performance (COP) was increased as the quantity of refrigerant charged increased. However, the quantity of refrigerant charged for refrigeration system attained its optimum coefficient of performance (COP) at 42 psi. Therefore, the value of COP after the optimum point was dropped.

When the quantity of refrigerant charged into the compression-refrigeration system increased, the work done by the compressor decreased. Thereby, the overall refrigeration coefficient of performance (COP) better. The COP was inversely proportional to the work done by the compressor. The highest COP will be the desirable parameters to be achieved for a domestic refrigerator.

The role of refrigerant charged into the system was vitally important. More charge means more filling of evaporator with liquid, thereby; there are more refrigeration capacity until a limiting condition of liquid sucking by compressor was reached. The optimum charged quantity of refrigerant R-134a for this compression refrigeration system was proved to be at 42 psi for both sets of experiment.

For excessive liquid refrigerant charge in the refrigeration system, flooding or slugging could be occurred. When the condenser flooded with liquid during the overcharge, the condensing pressures will be increased. This causes high compression ratios and causes low volumetric efficiencies causing low refrigerant flow rates.

COP was an important parameter in compression refrigeration system which has to be optimized in a given refrigeration application for maximum conservation of energy. The COP was greater for the HFC134a/POE oil system with addition of nanoparticlesAl_2O_3 rather than the conventional HFC134a/POE oil system. This may be due to the inverse proportionality of COP to work done in refrigeration system. The higher the COP values, the better performance of refrigeration system.

4.5.3 Power consumption

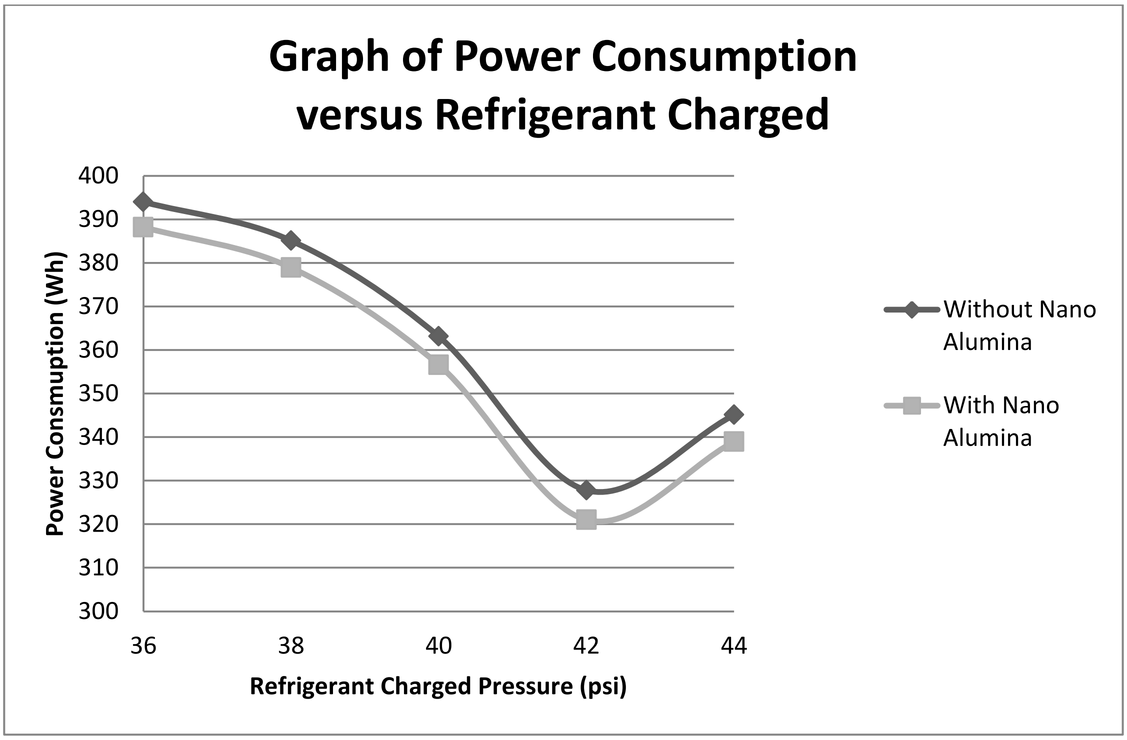


Figure 4.3: Power consumption versus refrigerant charged (psi)

As shown in the figure 4.3 above, the power consumption was reduced as the refrigerant charged into the refrigeration system increased. The power consumed by the HFC134a/POE oil system decreased as the work done by the compressor reduced. This was due to low compression ratio. A lower compression ratio caused higher volumetric efficiencies. A low compression ratio was resulted from high evaporator pressures. The evaporation rate was raised due to more fillings of liquid refrigerant in the evaporator. When the evaporator pressures increased, the work added to the compression stroke of the compressor decreased. This explained as the work done by the compressor reduced, the power consumption reduced as well.

The trend of graph for refrigerant charged ranged from 36 psi until 44 psi for both refrigeration systems was same. Both refrigeration systems attained optimum reduction in power consumption for refrigerant charged at 42 psi. It gave the least power consumption in term of kWh.

However, for the refrigerant charged at 44 psi, the power consumption increased. The compression ratio raised due to excessive refrigerant charged after the optimum point of 42 psi. The condenser flooded with liquid during the overcharge will run high condensing pressures. This causes high compression ratios and causes low volumetric efficiencies causing low refrigerant flow rates.

The power consumption of the refrigeration system was lesser for the HFC134a/POE oil system with addition of nanoparticlesAl_2O_3 than the conventional HFC134a/POE oil system. This was because nanoparticles enhanced the overall performance of HFC134a/POE oil system, thus, the energy consumed by the refrigerator became lesser.

Nanoparticles Al_2O_3 also enhanced the solubility of the HFC134a and POE mineral oil as indicated by the high oil return ratio. Thereby, the wear, tear and friction of the compressor reduced, the compressor work reduced; then the power consumption reduced as well.

4.5.4 Percentage of increment and reduction

The results obtained from the studies for refrigeration system operating with and without nano alumina will be compared and analyze. The comparison will be made base on the coefficient of performance (COP) and power consumption (wh). This is done to better reflect on the performance of refrigeration system that is operating under two conditions which is with and without nano alumina. The percentage of increment in term of COP and reduction in energy consumption for the refrigeration system operating with and without nano alumina was compare and analyze as in the Table 4.7 and Table 4.8.

As shown in the Table 4.7, the percentage of increment in term of COP for HFC134a/POE oil system with addition of nanoparticles Al_2O_3 was laid between 2.81% to 5.91 % compared to the conventional HFC134a/POE oil system. The percentage of reduction in term of power consumption for the HFC134a/POE oil system with addition of nanoparticles Al_2O_3 was laid within 1.47% to 2.07% compared to the conventional HFC134a/POE oil system. The addition of Al_2O_3 nanoparticles into the base fluid (POE) for better performance and reduction of power consumption was achievable.

Table 4.7: The percentage of difference for both refrigeration systems in term of COP

Refrigerant Charged (psi)	Coefficient of Performance (COP)		Percentage of increment (%)
	With nano alumina	Without nano alumina	
36	2.15	2.03	5.91
38	2.31	2.23	3.59
40	2.53	2.42	4.55
42	2.67	2.57	3.89
44	2.56	2.49	2.81

Table 4.8: The percentage of difference for both refrigeration systems in term of power consumption

Refrigerant Charged (psi)	Power Consumption		Percentage of Reduction (%)
	With Nano Alumina	Without Nano Alumina	
36	388.20	394.0	1.47
38	378.90	385.10	1.61
40	356.57	363.10	1.80
42	321.0	327.80	2.07
44	338.98	345.10	1.77

4.6 AGGREGATION AND SEDIMENTATION OF NANOPARTICLES AL_2O_3 LUBRICANT MIXTURE

The Al_2O_3 nano lubricant mixture sample prepared are kept for observation and no particle settlement was observed at the bottom of the glass containing Al_2O_3 nano lubricant mixture even after three hours. The stability of alumina (Al_2O_3) nanoparticle lubricant mixture was evaluated by observing the sedimentation pictures. The appearance of mixtures at various sedimentation times was compared. The time taken for observation on sedimentation of nano alumina was listed as in the Table 4.9 below.

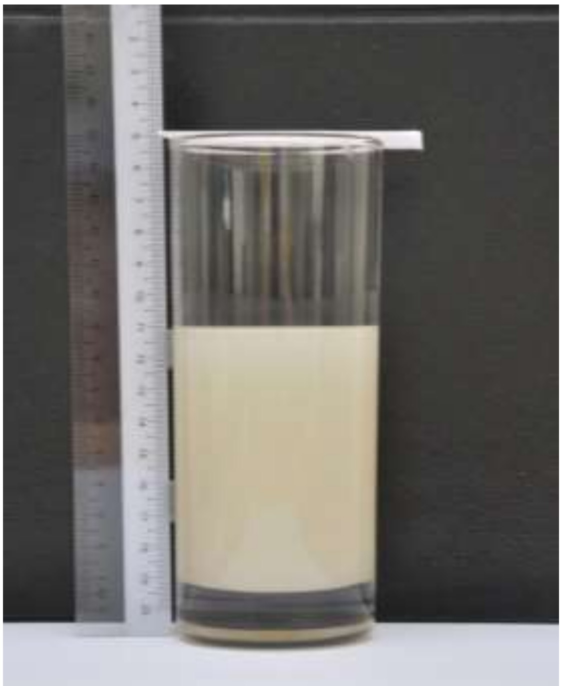
In the present experiments with Al_2O_3 nano lubricant mixture, the time taken to complete the experiment for property estimation is less than the time required for first sedimentation to take place and hence surfactants are not mixed in the Al_2O_3 nano lubricant mixture. The Al_2O_3 nano lubricant mixture prepared are assumed to be an isentropic, Newtonian in behavior and their thermo physical properties are uniform and constant with time all through the fluid sample.

Table 4.9: Time taken for observation on sedimentation of nano alumina

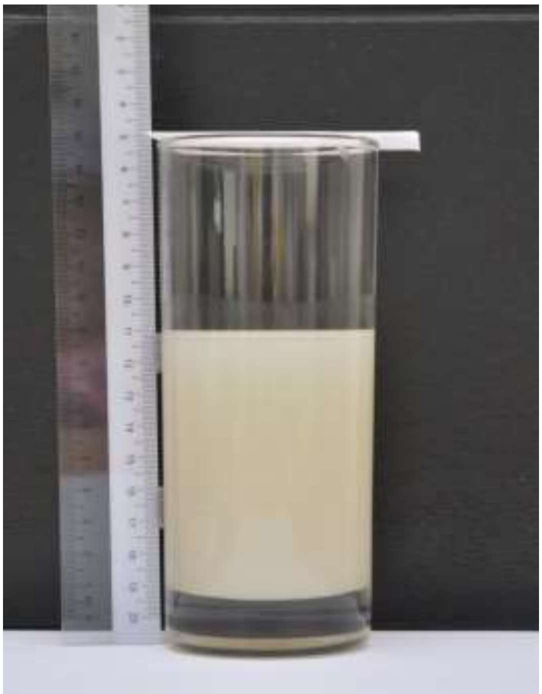
Date	**Time**	**Time Taken for Sedimentation (Hours)**	**Time Taken for Sedimentation (Cumulative Hours)**	**Initial Height of Nano Lubricant Mixture (cm)**	**Height of Nano Lubricant Mixture After De-gradation (cm)**
26 Feb 2013	10.00 am	0	0	9	9
26 Feb 2013	11.00 am	1	1	9	9
26 Feb 2013	12.00 am	1	2	9	9
26 Feb 2013	1.00 pm	1	3	9	9
26 Feb 2013	2.00 pm	1	4	9	8.70
26 Feb 2013	4.00 pm	2	6	9	8.60
26 Feb 2013	6.00 pm	2	8	9	8.50
26 Feb 2013	8.00 pm	2	10	9	8.40
26 Feb 2013	10.00 pm	2	12	9	8.30
27 Feb 2013	12.00 am	2	14	9	8.20
27 Feb 2013	6.00 am	6	20	9	7.80
27 Feb 2013	12.00 pm	6	26	9	7.60
27 Feb 2013	6.00 pm	6	32	9	7.40
28 Feb 2013	6.00 am	12	44	9	6.80
28 Feb 2013	6.00 pm	12	56	9	6.40
4 Mar 2013	10.00 am	88	144	9	5.40

The sedimentation pictures taken after an hours, 2 hours, 6 hours, 12 hours and 84 hours. Figure 4.5 (a) was the initial condition of Al_2O_3 nanoparticle-lubricant mixture after stirring. The bonding between nanoparticle and lubricant was still strong. There was no any change on precipitation for Al_2O_3 nanoparticle-lubricant mixture for first three hours. The precipitation slightly can be observable after four hours of stirring. The Al_2O_3 nanoparticle-lubricant mixture settles to the bottom gradually with times. The settlement of Al_2O_3 nanoparticle-lubricant mixture after one week (144 hours) was more than half of its initial condition. This can be seen clearly as in the Figure 4.5 (p) below.

(a) At 0 hours (10.00 am on 26 Feb 2013)



(b) After 1 hour (11.00 am on 26 Feb 2013)



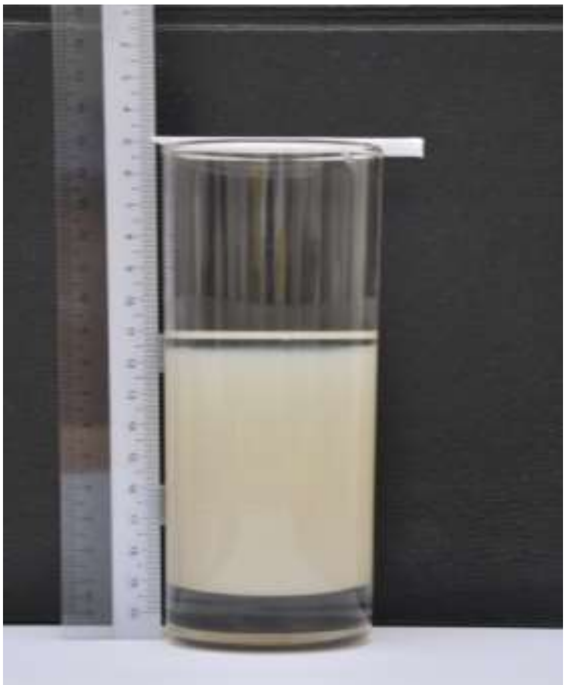
(c)After 2 hours (12.00 pm on 26 Feb 2013)

(d) After 3 hours (1.00 pm on 26 Feb 2013)

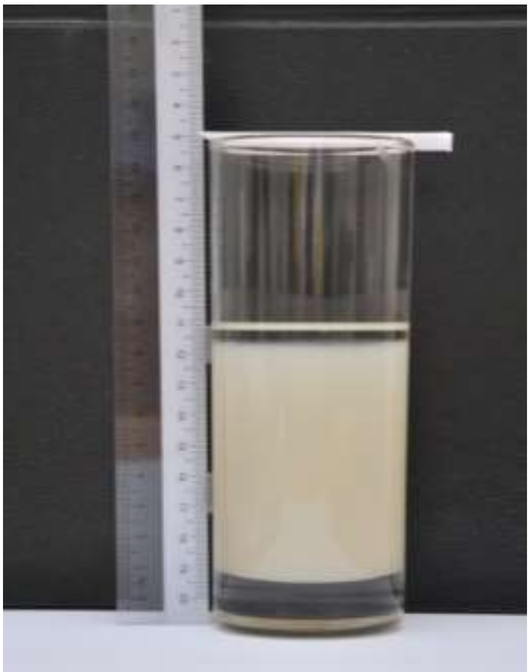
(e) After 4 hours (2.00 pm on 26 Feb 2013)

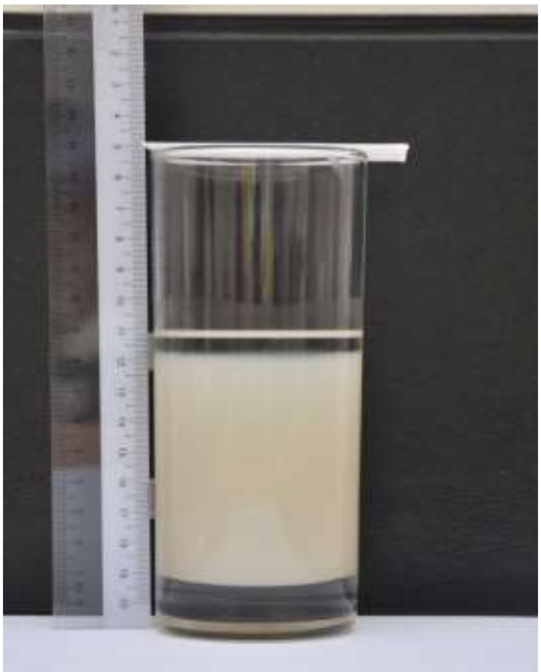
(f) After 6 hours (4.00 pm on 26 Feb 2013)

(g) After 8 hours (6.00 pm on 26 Feb 2013)



(h) After 10 hours (8.00 pm on 26 Feb 2013)

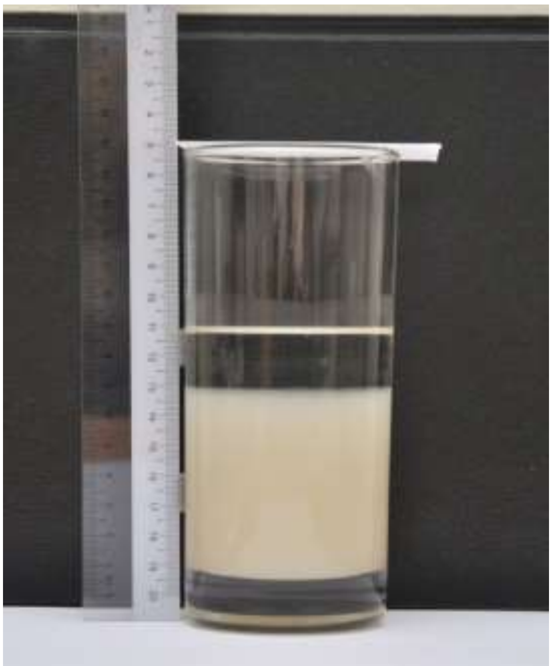




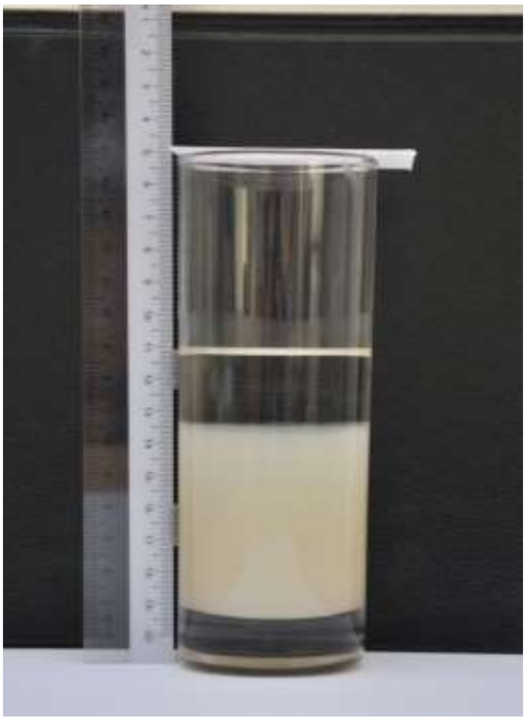
(i) After 12 hours (10 pm on 26 Feb 2013)   (j) After 14 hours (12 am on 27 Feb 2013)

(k) After 20 hours (6 am on 27 Feb 2013)   (l) After 26 hours (12pm on 27 Feb 2013)

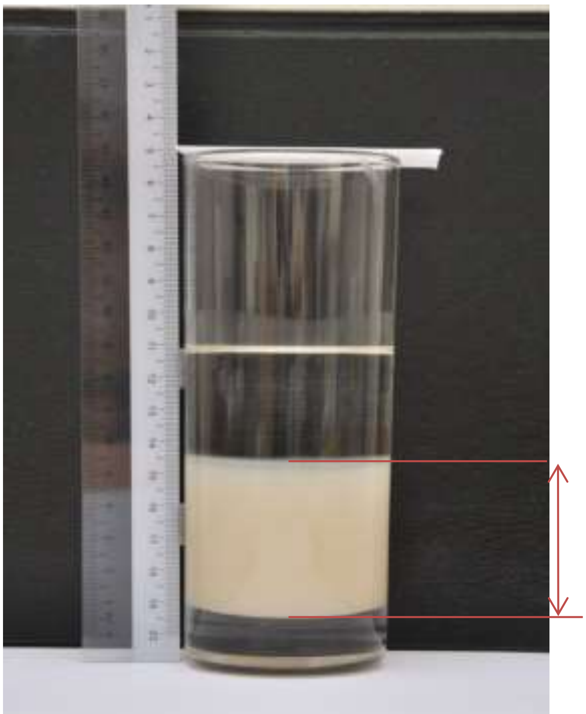
(m) After 32 hours (6 pm on 27 Feb 2013)



(n) After 44 hours (6 am on 28 Feb 2013)



(o) After 56 hours (6 pm on 28 Feb 2013)



(p) After 140 hours (6 am on 4 March 2013)

Figure 4.5: Stability of nanoparticles Al_2O_3 lubricant mixture

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

This study offers some insight into the role of nano additives and refrigerant charged over the performance characteristics of a simple vapour compression refrigeration system. The advantages of using nanoparticles Al_2O_3 lubricant mixture in the compressor of a domestic refrigerator was investigated experimentally. The conventional POE oil with addition of nano alumina was compatible with HFCR134a. The HFC134a/Al2O3/POE oil system was able to work normally and safely in domestic refrigerator.

The energy consumption for refrigeration system using HFC134a/POE oil with addition of nano alumina reduced the energy consumption for refrigeration system using HFC134a/POE oil between 1.47% for refrigerant charged at 36 psi, 1.61% for refrigerant charged at 38 psi, 1.80% for refrigerant charged at 40 psi, 2.07% for refrigerant charged at 42 psi and 1.77% for refrigerant charged at 44psi. At refrigerant charged at pressure at 42 psi, the energy consumption was at its minimal level which is 0.321 kW/h.

The overall performance of HFC134a/Al2O3/POE oil system was better than the conventional HFC134a/POE oil system. There was an improvement for the operating refrigeration system with optimum refrigerant charge which was 42 psi. The COP at optimum refrigerant charged which was 42 psi for HFC134a/Al2O3/POE oil system was 2.67 whereas the conventional HFC134a/POE oil system was 2.57. For HFC134a/Al2O3/POE oil system, the COP was between 2.15 to 2.56 for refrigerant charged ranged from 36 psi top 44 psi. Then, the

COP for HFC134a/ POE oil system was between 2.03 until 2.49. The percentage of increment in term of COP from refrigeration system using HFC134a/POE oil to refrigeration system using HFC134a/POE oil with addition of nano alumina was 5.91%, 3.59%, 4.55%, 3.89% and 2.81%.

The energy consumption reduced with Al^2O^3 nanoparticles-lubricant mixture due to the improvement in the overall performance of the refrigeration system. The conservation of energy was at its minimal level in correlation with optimum COP for the refrigeration system. When the compressor work reduced, the refrigerating effect increased, the coefficient of performance (COP) gave better performance. The AL_2O_3 nanoparticle-lubricant mixture enhanced the solubility of the HFC134a and POE mineral oil as indicated by the high oil return ratio. It could be reduce the wear, tear and friction in the compressor and thus the work done by the compressor will be lesser.

When the quantity of refrigerant charged increased to its optimum amount, the evaporation rate increased; therefore, the refrigerating effect increased too. This is due to the increment of overall heat transfer coefficient in the evaporator as there are more filling of liquid refrigerant HFC134a into the refrigeration system. The dry out point in the evaporator can be shifted downstream by allowing more liquid to stay in the evaporator by increasing the refrigerant charged. However, the optimum amount of refrigerant charged into the system attained when a limiting condition of liquid sucked by compressor was reached. Flooding or slugging could be occurred due to excessive liquid refrigerant charge in the refrigeration system. The compression pressure raised and the compressor may damage. Therefore, the refrigerant charged at optimum level should be emphasized. As from the results shown in chapter 4, the optimum performance attained at refrigerant charged at 42 psi for both sets of experiment.

In order to achieve the most desirable performance for the domestic refrigerator, nano alumina should be added into the conventional HFC134a/POE oil system at concentration by volume 0.20%. The refrigeration HFC134a/POE oil system with nanoparticles Al_2O_3 was more preferable and suitable for domestic refrigerator whose gave optimum refrigeration performance and maximum conservation of energy if compared to the conventional HFC134a/POE oil system.

5.2 RECOMMENDATIONS

The further study can be carried out to enhance the present investigation with several recommendations. The liquid typed nanoparticles were suggested for mixing in future studies. The liquid typed nanoparticles solution can be used for mixing directly. The stability for liquid type nanoparticles will be better than those of powder typed. The phase change of refrigerant with addition of liquid typed nanoparticles will be easier if compared to those of powder typed in the condenser.

Besides, there are many other types of nanoparticles can be used for investigation apart from the nanoparticles Al_2O_3. CNT was compatible with refrigerant typed of R113 and able to increase the thermal conductivities for refrigeration system too.

The themophysical properties of nanoparticles such as density should be determined via experimental method rather than taken from other previous researchers directly. The correlation via experimental method for density of nanoparticles enables us to obtain optimum accuracy for the results.

The conventional HFC134a/POE oil system was recommended to substitute by HFC134a/SUNISO 3GS mineral oil. The chemical stability of SUNISO 3GS mineral oil was excellent. Therefore, it might perform better than the POE oil in domestic refrigerator.

For more precise measurement on temperature, the thermocouple should be placed inside the tubing. For this research study, it was attach directly on the outer part of the copper tubing at the designated locations. The accuracy for the measurement on temperature will be higher with insertion method.

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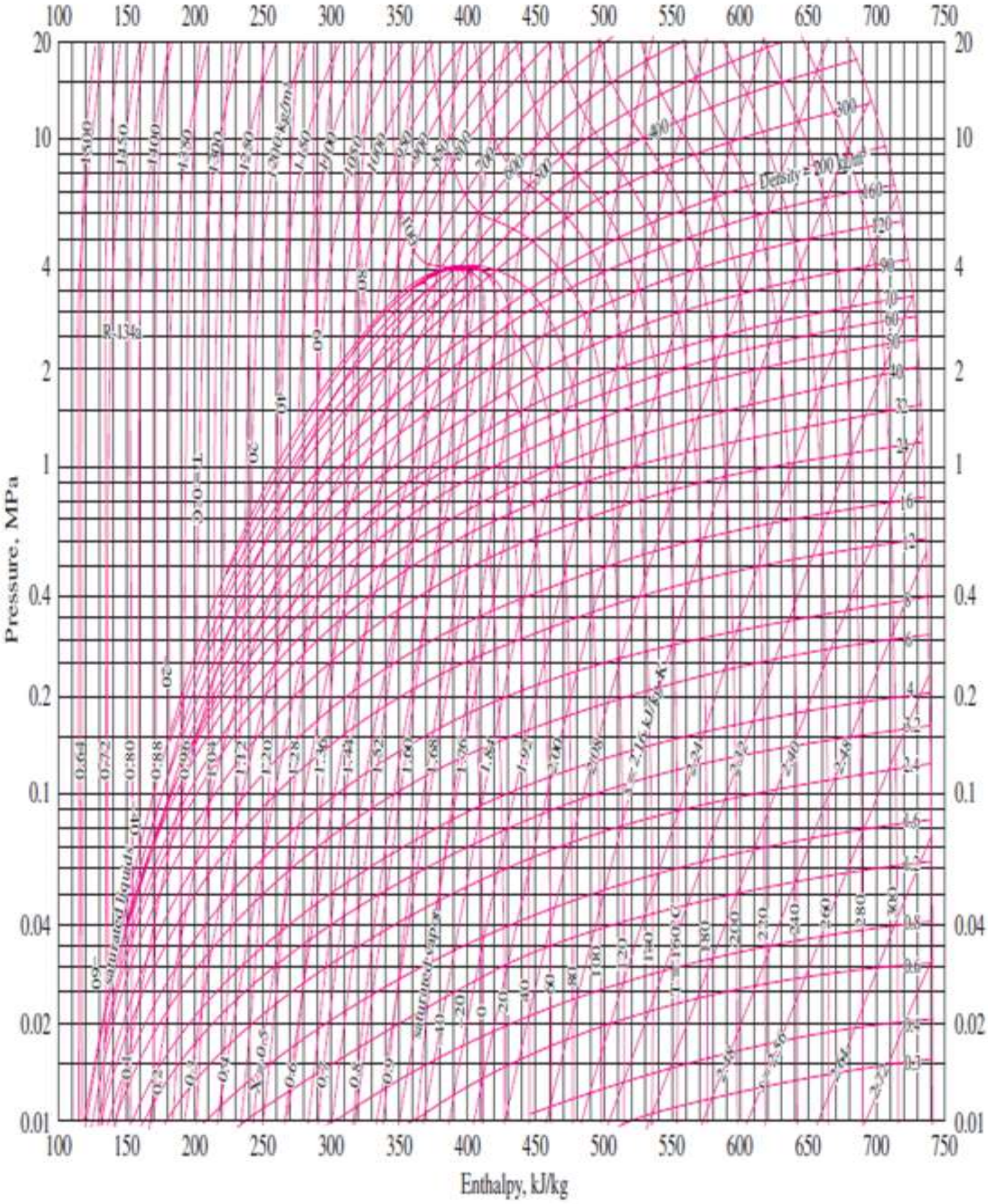
APPENDIX A

Pressure Enthalpy (p-h) diagram for refrigerant R-134a



APPENDIX B1

SAMPLE CALCULATION FOR ALUMINA BY VOLUME CONCENTRATION

Let $\rho_{alumina}$ = 3880 kg/m^3

$V_{polyester\ lubricant\ oil}$ = 190ml

Amount of Al_2O_3 needed for concentration by volume of 0.2%

$= 190\ ml\ x\ 0.002$

$= 0.38\ ml$

Mass of Al_2O_3 needed for concentration by volume of 0.2%

$$= 3880\frac{kg}{m^3}x\ 0.38ml\ x\ \frac{1\ m^3}{1000l}x\frac{1l}{1000ml}$$

$= 1.4744\ x\ 10^{-3}\ g$

$= 1.4744g$

APPENDIX B2

STANDARD EQUATION FOR ALUMINA BY VOLUME CONCENTRATION

$$vol\,\% = [\frac{\left(\frac{m}{\rho}\right)}{\left(\frac{m}{\rho} + Vbasefluid\right)}]$$

m – Mass of Al_2O_3

ρ – Density of Al_2O_3

$V_{base\ fluid}$ – Volume of base fluid

Let V_{np} = volume of nanoparticles

V_{bf} = volume of base fluid

$$\text{Ø} = \frac{Vnp}{Vnp+Vbf}\ x\ 100\%$$

$$\rho = \frac{m}{v}$$

Therefore:

$$\text{Ø} = \frac{\left(\frac{m}{\rho}\right)np}{\left(\frac{m}{v}\right)np + (V)basefluid}$$

For example,

$$\frac{0.2}{100} = \left[\frac{\frac{(m)np}{3880}}{\frac{(m)np}{38800} + 0.0019}\right]$$

$$0.002 = \frac{(2.5773x10^{-4})\ x\ (m)np}{[(2.5773x10^{-4}\ )\ x\ (m)np] + 0.00019}$$

$(m)np$ = 1.4744 g (PROVED)

APPENDIX C1

GANTT CHART FYP 1

No	Project Activities	Weeks (Semester 1)														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	Discuss on the title of FYP 1															
2	Discuss the objective and scopes															
3	Draft Chapter 1															
4	Literature study and find the related info															
5	Draft Chapter 2															
6	Discuss on the methodology and experiment set up															
7	Draft Chapter 3															
8	Experimental Setup															
9	Finalize Chapter 1, 2 and 3															
10	Draft Report submission															
11	Presentation															

Planning	
Actual Schedule	

APPENDIX C2

GANTT CHART FYP 2

No	Project Activities	Weeks (Semester 2)														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	1st Preliminary Experimental Testing															
2	2nd Preliminary Experimental Testing															
3	3rd Preliminary Experimental Testing															
4	Sample Preparation of Nano Alumina															
5	1st Experimental Testing with Nano															
6	2nd Experimental Testing with Nano															
7	3rd Experimental Testing with Nano															
8	Finalize Chapter 4															
9	Finalize Chapter 5															
10	Final Report Submission															
11	Final Presentation															

Planning	
Actual Schedule	