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EFFECT OF REFRIGERANTS BLEND ON THE PERFORMANCE OF A REFRIGERATION SYSTEM.

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ABSTRACT :

This study explores the fabrication and testing of a straight forward, yet efficient two-refrigerant blending plant. Motivated by the need for a cost-effective and versatile solution in refrigeration systems, the design focuses on simplicity while maintaining optimal performance. The blending plant combines two refrigerants, carefully selected for their compatibility and performance characteristics. This study unfolds in several key stages, beginning with a detailed conceptual design that prioritizes ease of fabrication and operational functionality. Utilizing computer-aided design (CAD) tools, the team develops 3D models and engineering drawings to guide the fabrication process. Material selection considers the thermodynamic properties required for effective refrigeration, ensuring durability and efficiency. The chosen materials undergo precision machining, welding, and assembly, guided by the overarching principle of facilitating ease of manufacturing and scalability. The fabrication involves precision machining, welding, and assembly to bring the components together into a cohesive system. Following fabrication, the blending plant undergoes rigorous testing procedures to validate its performance under varying conditions. Tests include assessments of refrigerant mixture ratios, pressure levels, and thermal efficiency. The results were analyzed against design specifications to ensure the plant meets industry standards and is suitable for diverse refrigeration applications. This study not only contributes a practical solution to refrigeration challenges but also serves as a foundational exploration into the efficient blending of refrigerants. The simplicity of the design enhances its accessibility for broader applications, making it a promising candidate for small-scale refrigeration systems. The objective is to offer a versatile solution for refrigeration systems that balances efficiency, affordability, and adaptability to diverse applications.

Keywords: Refrigerants, Coefficient of Performance, Energy Analysis, Blend, Refrigeration Cycle.

INTRODUCTION:

Refrigerant are working fluids used in the refrigeration cycle of air conditioning systems and heat pumps where in most cases they undergo a repeated phase transition from a liquid to a gas and back again. Refrigerants are heavily regulated due to their toxicity, flammability and the contribution of CFC and HCFC refrigerants to ozone depletion and that of HFC refrigerants to climate change. Refrigerants are used in a direct expansion (DX) system to transfer energy from one environment to another, typically from inside a building to outside (or vice versa) commonly known as an "air conditioner" or "heat pump. Air conditioners use refrigerants to carry heat between the indoor and outdoor units. In late 1970s it was discovered that hydro-chlorofluorocarbons (HCFC) and chlorofluorocarbon (CFC) gases which were the most common refrigerants including air and water.

All refrigerants use Rl1 as a datum reference and thus RII has an ozone Depletion potential (ODP) of 1.0. The ozone depletion potential (ODP) is the ratio of the potential impact or ozone of a chemical compared to the impact of the same mass of chlorofluorocarbon-12 (CFC-12). The less the value of the Ozone Depletion potential (ODP), the better the refrigerant is for the ozone layer, because the ozone layer have a potential for a single molecule of the refrigerant to destroy it. Chlorofluorocarbon (CFC) has high ozone depletion potential because of the absence of hydrogen in its molecule, this means there cannot be broken down easily until they reach the stratosphere and thus have a high ozone Depletion potential (ODP).

Hydro-chlorofluorocarbon (HCFC) has low ozone Depletion potential (ODP) because it contains hydrogen in its molecule and can be broken down respectively easily in the atmosphere and thus has a low ozone Depletion potential (ODP). Hydro-fluorocarbon (HFC) has zero ozone depletion potential (ODP) because it contains no chlorine or bromine in its molecule. Furthermore, these refrigerants used in our air conditioners and refrigerators also cause Global warming is the rise in the average temperature of earth’s atmosphere and oceans.

In other words, Global warming refers to a phenomenon in which infra-red rays are absorbed by molecules in CO_2 and methane as well as air conditioner refrigerants like chlorofluorocarbons (CFC), hydro-chlorofluorocarbons (HCFC) and hydro-fluorocarbons (HFC), this prevents heat from escaping the earth surface. Since the late 19th century and early 20th century, earth’s main surface temperature has increased by about 0.8°C (1.4°F) with about two-thirds of the increasing occurrence since 1980. Global warming of the climate system is unequivocal, and scientists are more than 90% certain that it is primarily caused by human activities such as green-house gases produced by human activities, burning of fossils fuels refrigerants and deforestation. These findings are recognized by the national science academies of all major industrialized nations.

Global warming potential (GWP) is a measurement of how much effect the given refrigerant will have on global warming relatives to carbon dioxide, where CO_2 has a global warming potential (GWP) of 1. In this case the lower the value of global warming potential (GWP) the better the refrigerant is for the environment. In developing country like Nigeria, most of the vapor compression-based refrigeration, air conditioning and heat pump systems continue to run on halogenated refrigerants due to its excellent thermodynamic and thermo-physical properties apart from the low cost. However, the halogenated refrigerants have adverse environmental impacts such as ozone depletion potential (ODP) and global warming potential (GWP). Hence, it is necessary to look for alternative refrigerants to full fill the objectives of the international protocols (Montreal and Kyoto) and to satisfy the growing worldwide demand. This study reviews the various experimental and theoretical studies carried out around the globe with environment friendly alternatives such as hydrocarbons (HC), hydro-fluorocarbons (HFC) and their mixtures, which are going to be the promising long-term alternatives. In addition, the technical difficulties of mixed refrigerants and future challenges of the alternatives are discussed. The problems pertaining to the usage of environment friendly refrigerants are also analyzed.

Greenhouse gas (GHG) emissions from fossil fuel combustion for power generation and emission of halogenated refrigerants from vapor compression-based refrigeration, air conditioning and heat pump systems contribute significantly to the global warming. A reduction in GHG emissions can only be achieved by using environment friendly and energy efficient refrigerants. The high environmental impacts due to halogenated refrigerant emissions lead to identifying a long-term alternative to meet all the system requirements including system performance, refrigerant—lubrication interaction, energy efficiency, safety and service. Halogenated refrigerants have dominating the refrigeration and air conditioning industries over many decades due to its excellent thermodynamic and thermo-physical properties.

As per the Montreal protocol 1987, developing countries like India, with a per capita consumption of less than 0.3 kg of ozone depletion substance have been categorized as Article-5 countries. These countries are required to phase out all chlorofluorocarbons (CFCs) by 2010 and all hydro-chlorofluorocarbons (HCFCs) by 2040 (Powell, 2002). Johnson (1998) has reported that HFC refrigerants are considered as one among the six targeted greenhouse gas under Kyoto protocol of United Nations Framework Convention on Climate Change (UNFCCC) in 1997. Most of the developed countries reduced the production and consumption of halogenated refrigerants, which demands for suitable alternatives. HC and HFC based refrigerants with zero ODP and low GWP are considered to be long-term alternatives. On the other hand, HC refrigerants have flammability issues, which restrict the usage in existing systems. However, the reduction in flammability can be achieved by blending HC refrigerants with HFC refrigerants (Yang et al., 2004). Formeglia et al. (1998) reported that, it is possible to mix HC refrigerants with other alternatives such as HFC refrigerants. The miscibility of HC/HFC mixtures with mineral oil has been reported to be good (Avinash et al., 2005). The GWP of HC/HFC mixtures is less than one third of HFC, when it is used alone. Since the phase out of CFCs more than 10 years ago, refrigerant blends have become commonplace in the market for both retrofit refrigeration and new installations. Equipment that traditionally used R-12 or R-502 is now running on one of approximately 13 commercially available blends. When you consider the pending phase out of R-22, another three or four blends get thrown into the mix. In addition, contractors and service technicians must know the pitfalls of refrigerant blends. Fortunately, we have learned much about blend performance during the last 15 years.

Each refrigerant blend has its own unique properties that are somewhat different from the original product they are intended to replace. By understanding how blends differ from single-component refrigerants, contractors and technicians can better identify or avoid blend-related problems when installing or servicing equipment. Fractionation and temperature glide will affect system operation, control settings and service/troubleshooting practices. Different blends will show different amounts of fractionation or temperature-glide effects, though the impact on a system will be similar for all blends. Refrigerant blends are mixtures of refrigerants that have been formulated to provide a match to certain properties of the refrigerants originally used. These blends have been researched and developed since the issue of the ODS phase-out emerged and are being produced by many chemical companies. Blends can have 2-3 or even 4 components, and S can have a major component of a HCFC, HFC or HC; in most cases they will consist of a combination of these chemicals.

The refrigerant blends have their own trade names. The well-known ASHRAE (American Society of Heating, Refrigerating and Air Conditioning Engineers) refrigerant number also applies to blends.

The blends that are currently widely used around the world are HFC based such as R-407C and R-410A to replace R-22. However, due to the increase of the prices of R-134a, the HCFC-based blends such as R-406A and R-415B have entered into the regional and world servicing market for the replacement of R-12, and even R- 134a. This study is aimed at investigating the negative effects of the refrigerants used in air conditioners and refrigerators has on the ozone layer and the globe at large and how new refrigerants can be produced that will have very little ozone Depletion potential (ODP) value and global warming potential (GWP) value. Also due to the environmental impact of refrigerants to the ozone layer, continues manufacturing of air conditionals and refrigerators that uses refrigerants like chlorofluorocarbons (CFC) and hydro-chlorofluorocarbon (HCFC) will lead to further depletion of the ozone layer. There is need to improve the performance of the air conditioner and refrigeration system by reducing the heat emission to the atmosphere and increases the cooling effect, by blending R-12 and R-134a.

This study will help improve the coefficient of performance of the air conditioner and refrigeration system through the blend of R-12 and R-134a, thereby preventing further destruction of the ozone layer which allows more harmful ultraviolet rays to reach the earth surface, resulting in the increase of skin cancer and other problems. This study will also prevent further starvation, malnutrition and increased death due to food and crop shortages because of global warming.

MATERIALS AND METHOD

3.1.3 Description of the Blending Rig.

The Refrigerant Blend

The blending rig is a machine that can combine two or more component (in this case refrigerant) to yield a desired product. Refrigerant blending rigs combine two or more refrigerants in order to produce an improved refrigerant with better thermodynamic properties and more environmentally friendly.

The rig is supposed to blend (mix) two refrigerants dichlorodifluoromethane (R-12) with a molecular formula of CCl_2F_2 and tetrafluoroethane (R-134a) with a molecular formula of CH_2FCF_3 in balancing the molecular formula we have

$CH_2FCF_3 + CCL_2F_2 \rightarrow CH_2FCFCF_3 + CL_2$

The rig for this study has two refrigerants cylinders containing the refrigerants R-12 and R-134a on each side. The refrigerant pass through a hose fixed to the cylinders. The hose are connected to the refrigerant pipelines which collect and distribute the refrigerants. On the pipe network is the solenoid valve on either sides which maintain the flow of the refrigerant. Similarly, on either side of the pipe is the sight glass which enables proper viewing of the refrigerant flow. After the sight glass is the pressure gauge also on either side of the pipe network. This pressure gauge measures the pressure of each refrigerant. The refrigerant from each section enters the muffler (mixer) in which the actual blending (mixing) takes place. A hose is connected to the muffler which distributes the refrigerant blend into the collector, this collector is more of a final product cylinder that harbors the mixed refrigerant. A switch is used to open or close the circuit for powering the system. For this design, cylinder R12 is at the left – hand side and it was worm during the experiment. While cylinder R134a is at the right hand side which was cold during the experiment. This is shown alongside with the various components of the blend in the figure.

3.2 Methods

3.2.1 Construction of the Blending Rig

The blending rig was constructed by first; the angle iron was measured at 3ft using a tape and was placed on a machine vice, hack saw was used to cut the angle iron. The angle iron was welded together forming a rectangle L=3ft and W=1.2ft. At the extreme end of the side of the rectangle, 10mm was measured apart (using scriber to mark out) and a hand drilling machine was used to drill the hole where the roller can fit in. The hole was drilled using drilling bit size of 17 and the roller that was fit in was marked iut, cut at the width of 3ft and length of 4.5ft, which was later welded to the angle iron. The surface of the welded part was rough and a circular machine (filling machine) was used to smoothen the surface. From the top 1.6ft was marked out on both sides and an iron was welded on both sides for a firm stand.

A sheet metal was marked cut from both ends and the angle of the cylinder was cut out, then the sheet metal was welded to the top of the rig. From both sides 10.9inch was measured and a hole was drilled that the electric cable can pass through. From both sides, 3inch was marked out (i.e. 1.5ft from the top), and a hole was drilled using an electric hand drilling machine where the pipe lines can fit in properly. From the bottom 1ft at the centre was measured where the hole was drilled so that a muffler pipe can fit in.

Along the pipe lines are two solenoid valve, two sight glass and two pressure gauges. Beneath the cylinder seater on the left hand side is an electric connector where electric cables are connected to a switch and once it is ON the response will be from both side of R134a and R12 lines.

3.2.2 Theoretical Mixing of R-12 and R134a

Basic adiabatic equation of mixing of ideal gases was adopted and the properties of the two refrigerants were obtained from Johnson Tomczyk refrigeration and air conditioning technology 5Th edition. These were used for obtaining the mixture temperature and pressure. The assumption made included.

1. Assume 0.5kg of each refrigerants.
2. Assume $V_m = V_{R12} + V_{R134a}$ (No volume loss)

3.3 Refrigeration Cycles

3.3.1 Vapour Compression Refrigeration Cycle

The principle of operation of vapour refrigeration originated from a reversed Carnot power cycle. In the vapour compression cycle, the turbine in the Carnot cycle is replaced with a throttling device which can be an expansion valve, an expansion engine or capillary tube. It is cheaper to use an expansion valve or capillary tube than an expansion engine due to the high cost of the engine required to operate with two-phase flow. Components of an ideal vapour compression refrigeration cycle are illustrated in Figure 3.3. This cycle finds wide application in refrigerators, heat pumps and air conditioning systems. Figure 3.3 shows work and energy transfer when the system is operating in a steady state. A brief description of the units in the cycle is given below.

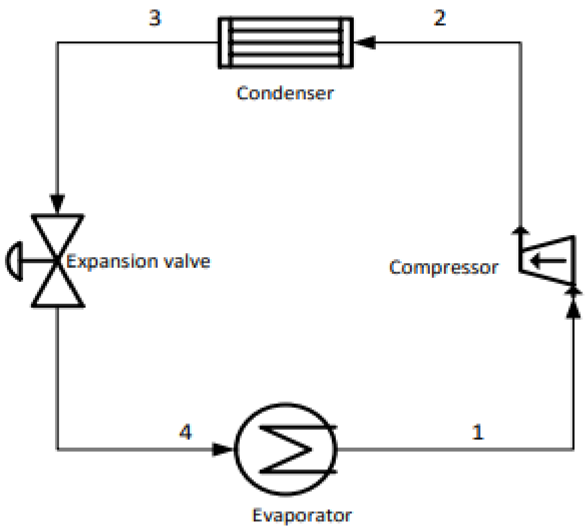

Figure 3.3: Components of Vapour-Compression Refrigeration Cycle.

Energy Analysis

All the major components in the vapour compression cycle shown in Figure 3.3 are internally reversible except for the throttling process. Since all the four components in the vapor compression refrigeration cycle are steady-flow units, therefore, it is referred to as an ideal cycle. For this reason, the analyses of all the cycle components processes can be done under steady-flow conditions (Moran and Shapiro, 2006)

$$\dot{Q}_{CV} = \sum_{0}\left(h+\frac{u^2}{2}+gz\right)_{o}\dot{m}_o - \sum_{i}\left(h+\frac{u^2}{2}+gz\right)_{i}\dot{m}_\iota + \dot{W}_{CV} \qquad \text{(i)}$$

Where Q_{cv} is the rate of heat transfer between the control volume and its surroundings [J/s] , h is the specific enthalpy [J/kg] , W_{cv} is the energy crossing the boundary of a closed system [J/s], $\dot{m}o$ and mi˙ are inlet and outlet mass flow rates, is the kinetic energy term, and gz the potential energy term. The potential and kinetic energy changes of the refrigerant across the cycle's components are small and thus can be neglected. Considering only work and heat transfer terms;

$$\dot{Q}_{CV} = \dot{h}_o\dot{m}_o - \dot{h}_i\dot{m}_i + \dot{W}_{CV} \qquad \text{(ii)}$$

Then the steady-state equation on a unit mass basis assuming constant mass flow rate in the system reduces to:

$$q - w = h_o - h_i \qquad \text{(iii)}$$

Where q is the heat transfer per unit mass [J/kg], $\dot{Q}$ is the rate of heat transfer between the control volume and its surroundings [J/s], $\dot{m}$ is the mass flow rate of the refrigerant [kg/s] and w is the work done per unit mass of the system.

Evaporator

Considering the refrigerant side of the evaporator as the control volume, denoted by 4-1 in Figures 3.3. The energy and mass rate balances (Equation ii) gives the rate of heat transfer per unit mass of the refrigerant flowing as:

$$\frac{\dot{Q}in}{\dot{m}} = h_1 - h_4 \quad \text{(iv)}$$

Compressor

Supposing no heat exchange occurs between the compressor and its surroundings, the energy and mass rate balances for a control volume encircling the compressor gives:

$$\frac{\dot{W}c}{\dot{m}} = h_2 - h_1 \quad \text{(v)}$$

Condenser

Considering the refrigerant side of the condenser as the control volume, the heat transfer rate from the refrigerant per unit mass flowing is:

$$\frac{\dot{Q}o}{\dot{m}} = h_2 - h_3 \quad \text{(vi)}$$

Expansion Valve

A two-phase liquid-vapour refrigerant mixture exits the valve at the state 4 (Figure 3.3). The pressure decrease of the refrigerant is an adiabatic process which is not reversible, accompanied by an increment in specific entropy:

$$h_4 = h_3 \quad \text{(vi)}$$

The coefficient of performance (COP) of the refrigeration system is given by:

$$\text{COP} = \frac{Refrigeration\ Effect}{Work\ Input} \quad \text{(vii)}$$

$$\text{COP} = \frac{h_1 - h_4}{h_2 - h_1} \quad \text{(viii)}$$

COP is higher for refrigerants with higher critical temperatures. Furthermore, it decreases as the temperature of the condenser reaches the refrigerant’s critical temperature (Venkatarathnam and Murthy, 2012). In summary, to obtain a high COP, vapour density, liquid thermal conductivity, and latent heat should have high values. Whereas molecular weight and liquid viscosity values should be low (Prapainop and Suen, 2012).

3.3.3 Procedure in carrying out the experiment using the Air conditioner and Refrigeration unit

Refrigerant R134a, R12 and blend of R134a/R12 was selected for use in the commissioning stage due to its widespread usage in refrigeration systems and the availability of vast amount of data published on it. Mollier charts were used to analyse the experimental data to obtain the derived results such as COP, compressor work, cooling effect and compressor efficiency. Test runs were undertaken to confirm that the unit was fit for use on a simple vapor compression cycle. To accomplish this objective, three test runs were carried out at set conditions to assess the functionality of the unit and to produce repeatable and consistent readings within experimental uncertainty. In achieving this, the critical operating parameters were kept constant. These parameters were as follows: mass of the refrigerant, compressor power setting, expansion valve setting, evaporator water flow rate, condenser cooling water temperature, and condenser water flow rate. The operating parameters selected to conduct the experiments were a refrigerant mass (charge) of 3.115 kg, condenser water flow rate of 2.75 L/min and a bath temperature of 20 °C. The evaporator flow rate was 0.58 L/min and the 0.128-inch/3.25 mm orifice Swagelok expansion valve was set in the half open position to achieve the desired level of superheating at the evaporator outlet. The compressor was set to 2.20 kW, the voltage to 400 V, current to 3 Amps and an operational speed of 1420 rpm. Whilst conducting the experiments, the room temperature was maintained at approximately 22^{o}C at all times, with the aid of an air-conditioning unit. This temperature control was implemented to eliminate the effects of external temperature variations to ensure a fair test environment. However, ambient temperature fluctuations were experienced during the experiments due to the malfunctioning of the air-conditioning system. The system was assumed to have reached steady-state when temperature and pressure fluctuations had ceased at the condenser and evaporator refrigerant entry and exit points. The system took approximately 40 minutes to stabilize. Once the system had stabilized, the temperature and pressure readings were logged every minute, for more than 30 minutes on the LabVIEW data logging system connected to the computer.

RESULTS AND DISCUSSION

The refrigerant blending rig was tested and gave reasonable performance, inspection by touching and sighting reviewed different temperature characteristics at refrigerant lines and the mixing chamber (muffler). TPS-3955 basic refrigeration and TPS-3954 professional air conditioning testing rig in the laboratory was used to determine the performance of the blended refrigerants.

Table 4.1 presents the thermodynamic properties of the refrigerants used in this study. As discussed by Venkatarathnam and Murthy (2012), the critical temperature and normal boiling point are fundamental thermodynamic properties of a refrigerant which influence the vapour pressure and the latent heat of vaporization. The refrigerants selected in this study have comparable critical pressures and temperatures, boiling points and molecular weights. Furthermore, their selection was influenced by their availability. However, while their GWP is in an acceptable range, the new European Union Fluorinated greenhouse gases (EU F-Gas) regulation states that from 2020, refrigerants with GWP value less than 2500 will be acceptable in refrigeration applications (Bitzter, 2014).

Table 4.1 Thermodynamic properties of the refrigerants

Properties	R134a	R12
Molecular Weight (g/mol)	102	120
Critical Temperature (^{o}C)	101.1	112
Critical Pressure (Mpa)	4.06	4.14
Bubble Point (^{o}C)	-26.1	-29.8
Dew Point (^{o}C)	-101	-
Temperature Glide (^{o}C)	0	0
ODP	0	1
GWP	1300	1810

4.2 Analysis of Result

The results obtained from the two pure refrigerants and two refrigerant blends are presented in Table 4.2.

Table 4.2: Comparison of experimental COP results for refrigerants and refrigerant blends.

Refrigerant/Refrigerant Blend (wt %)	Compressor Work (kW)	Refrigeration Effect (kW)	COP
R134a	0.9	0.43	4.78
R12	0.07	0.29	4.41
R134a\R12 (50/50)	0.05	0.24	4.80
R134a\R12 (66/34)	0.07	0.34	4.86

The refrigerant blend of R134a\R12 (66/34) had the highest COP while R12 had the lowest COP value. R134a, a pure refrigerant had the highest refrigeration effect while R134a/R12 (50/50) had the lowest refrigeration effect as well as the compressor work. The high COP value for R134a\R12 (66/34) was a result of both relatively high refrigeration effect and low power consumption at the compressor. Moreover, the high COP might have been due to high volumetric refrigeration capacity of the R134a\R12 (66/34) refrigerant as a result of the high pressure of its vapour since it had the highest pressure in the study (Prapainop and Suen, 2012). R134a/R12 (50/50 wt %) a laboratory synthesised blend in this study had the second highest COP value. The overall performance of refrigeration blends was better than that of pure refrigerants in the experiment, with R12 a pure refrigerant gave the lowest COP value.

Table 4.3: Comparison of R134a, R12, and R134a/R12 blends

Variables	R134a	R12	R134a\R12 (wt %)	R134a\R12 (wt %)

			(66/34)	(50/50)
COP	4.78	4.41	4.86	4.80
Refrigeration Effect (kW)	0.43	0.29	0.34	0.24
Compressor Efficiency (%)	0.69	0.69	0.75	0.75
Compressor Power (kW)	0.09	0.07	0.05	0.07
Vapour Fraction @ Evaporator Inlet	0.21	0.32	0.23	0.23
Compression Ratio	3.13	3.25	3.16	3.24
Discharge Temperature (ºC)	74.0	68.34	68.3	67.6
Refrigerant Evap Inlet Temp (ºC)	-6.5	-15.35	-12.7	-12.7
Refrigerant Evap Outlet Temp (ºC)	23.5	20.0	22.0	20.3
Refrigerant Cond Outlet Temp (ºC)	25.0	21.0	19.0	18.3

Considering pure refrigerants, from table 4.3, the refrigeration effect and compressor power of the R134a was higher than that of R12 therefore overall the coefficient of performance for R134a was greater than that of R12. Furthermore, R134a had a higher discharge temperature which was due to a high-power consumption at the compressor. For the refrigerant blends, (R134a/R12) blends had low discharge temperatures. Hence they are safe for the operation of the compressor and the compressor life. In general, the blends had superior compressor properties which are low compressor work and low discharge temperature. The efficiency values used in the experiment were obtained from the experimental runs hence the blends operate at a higher compressor efficiency. It is evident from the analysis in this section that blend formation improves the performance of pure refrigerants. The R134a/ R12 blend in the ratio studied (66/34 by wt %) had a high refrigeration effect and the low compressor power consumption consequently a high-value COP. The refrigeration effect of the (50/50 by wt %) R134a/ R12 had the lowest refrigeration effect. However, its compressor power was relatively low thus its COP was higher than that of R134a and R12.Refrigerant R134a expanded most favorable at the expansion valve as it existed with the largest percentage of the liquid phase in the two-phase mixture whereas R12 had the lowest proportion of liquid. However, R12 gave the lowest temperature at the evaporator inlet than R134a thus it is an excellent low-temperature refrigerant. From the analysis carried out above, the R134a/R12 (66/34 by wt %) blend is the best performing blend in the refrigerant unit operating conditions investigated in this study and can be utilized in medium temperature applications.

The results for these refrigerant blends display improved refrigeration performance in the experimental study over the pure refrigerant counterparts. The blends had higher COP values than the pure refrigerants as can be seen in Table 4.3. This result was to a large part due to lower compression power rather than the cooling effect.

4.3 Effect of Compressor speed on the Coefficient of Performance

Table 4.4 and figure 4.1 shows clearly that compressor speed have great impact on the coefficient of performance of the refrigeration system. It was observed from figure 4.1 that as the compressor speed reduces the coefficient of performance increases. It was seen that R134a/R12 (66/34 %wt) had the highest coefficient of performance for all varied compressor speed and R12 had the least coefficient of performance.

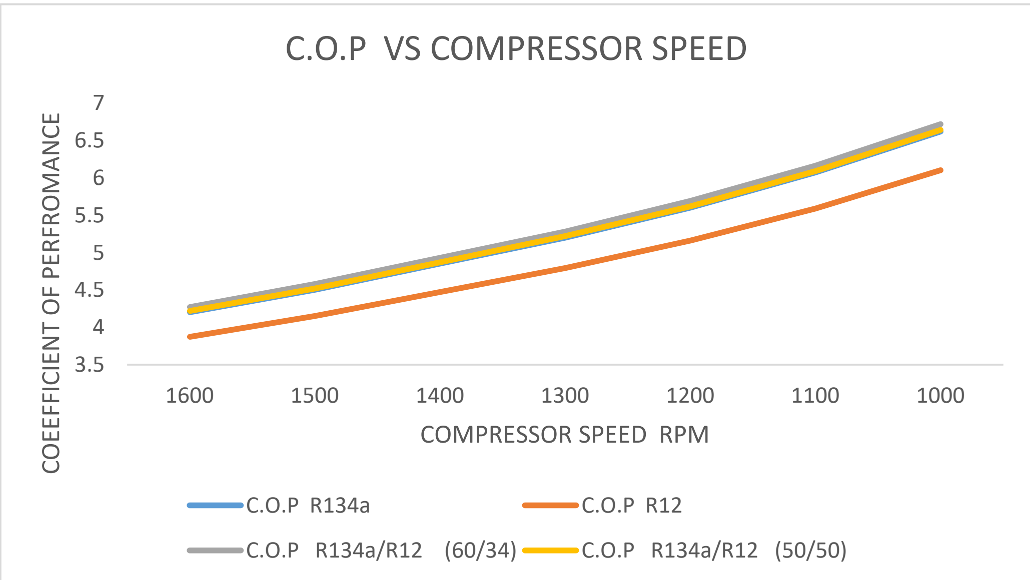

Figure 4.1: Graph of Coefficient of Performance (C.O.P) of refrigerants vs compressor speed

4.4 Effect of Ambient Temperature on the Coefficient of Performance

It is seen from figure 4.2 that ambient temperature is inversely proportional to the coefficient of performance of the refrigeration system. That is increase in ambient temperature results to reduction in the coefficient of performance of the refrigeration system. Table 4.5 shows that R134a/R12 (66/34 %wt) has the highest coefficient of performance of 5.3 at ambient temperature of 20oC, followed by R134a/R12 (50/50 %wt), R134a and then R12 respectively at same ambient temperature of 20oC and the R12 has the least coefficient of performance of 1.61 at ambient temperature of 50oC. This means that irrespective of the ambient temperature variation, both weight percentages of the refrigerant blends (R134a/R12) has higher coefficient of performance compare to the pure refrigerants (R134 and R12).

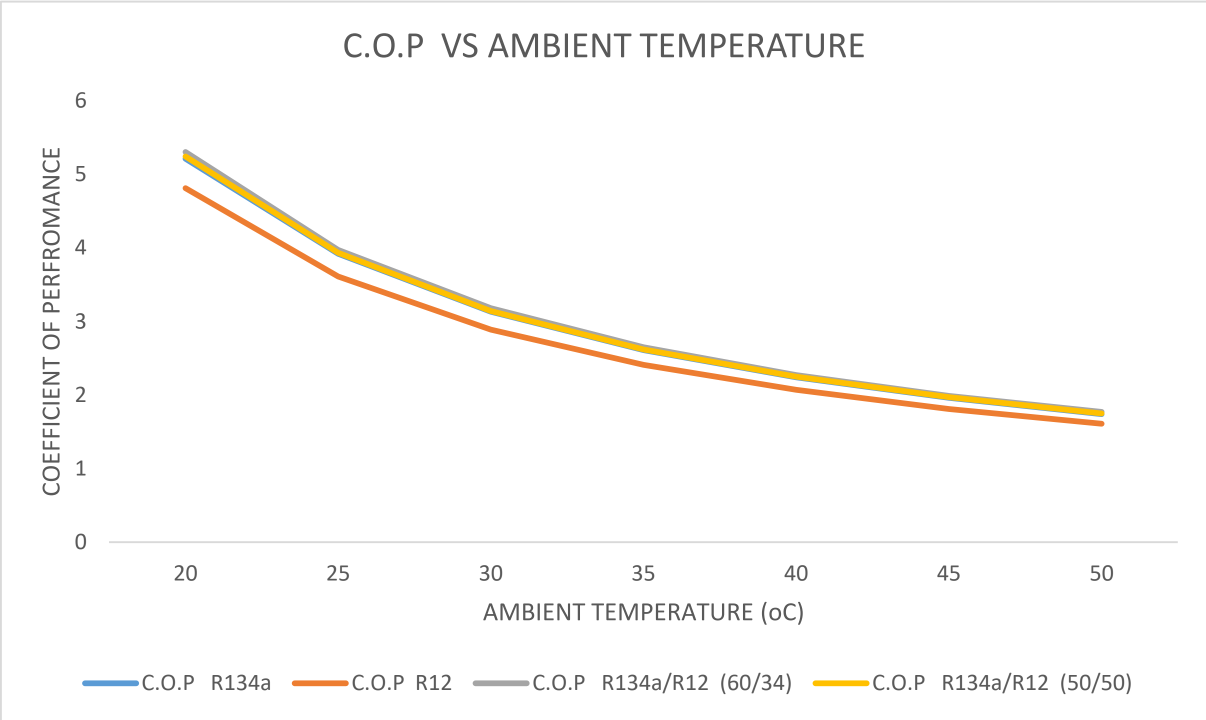

Figure 4.2: Graph of Coefficient of Performance (C.O.P) of refrigerants vs Ambient Temperature.

4.5 Effect of evaporator temperature on the Coefficient of Performance

Figure 4.3 shows that as the evaporator temperature increases the coefficient of performance of the refrigeration system also increases. Table 4.6 shows that R134a/R12 (66/34 %wt) has the highest coefficient of performance of 14.58 at evaporator temperature of -5oC, followed by R134a/R12 (50/50 %wt), R134a and then R12 respectively at same evaporator temperature of -5oC and the R12 has the least coefficient of performance of 4.4 at eveporator temperature of -25oC. This also means that irrespective of the evaporator temperature variation, both weight percentages of the refrigerant blends (R134a/R12) has higher coefficient of performance compare to the pure refrigerants (R134 and R12).

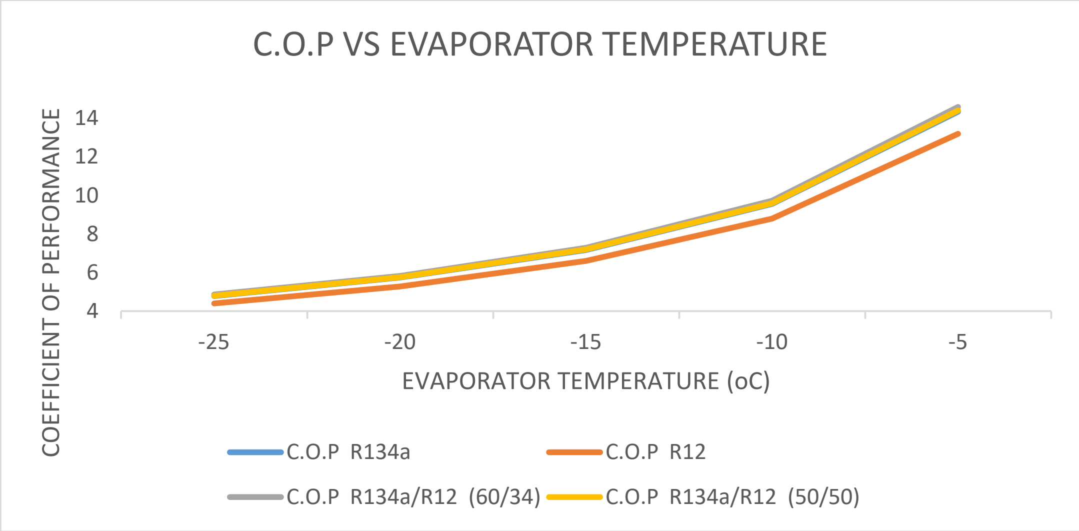

Figure 4.3: Graph of Coefficient of Performance (C.O.P) of refrigerants vs Evaporator Temperature.

4.6 Effect of condenser temperature on the Coefficient of Performance

It is seen from figure 4.4 that condenser temperature is inversely proportional to the coefficient of performance of the refrigeration system. That is increase in condenser temperature results to reduction in the coefficient of performance of the refrigeration system. Table 4.7 shows that R134a/R12 (66/34 %wt) has the highest coefficient of performance of 6.8 at condenser temperature of 40oC, followed by R134a/R12 (50/50 %wt), R134a and then R12 respectively at same condenser temperature of 40oC and the R12 has the least coefficient of performance of 2.76 at condenser temperature of 90oC. This means that irrespective of the condenser temperature variation, both weight percentages of the refrigerant blends (R134a/R12) has higher coefficient of performance compare to the pure refrigerants (R134 and R12).

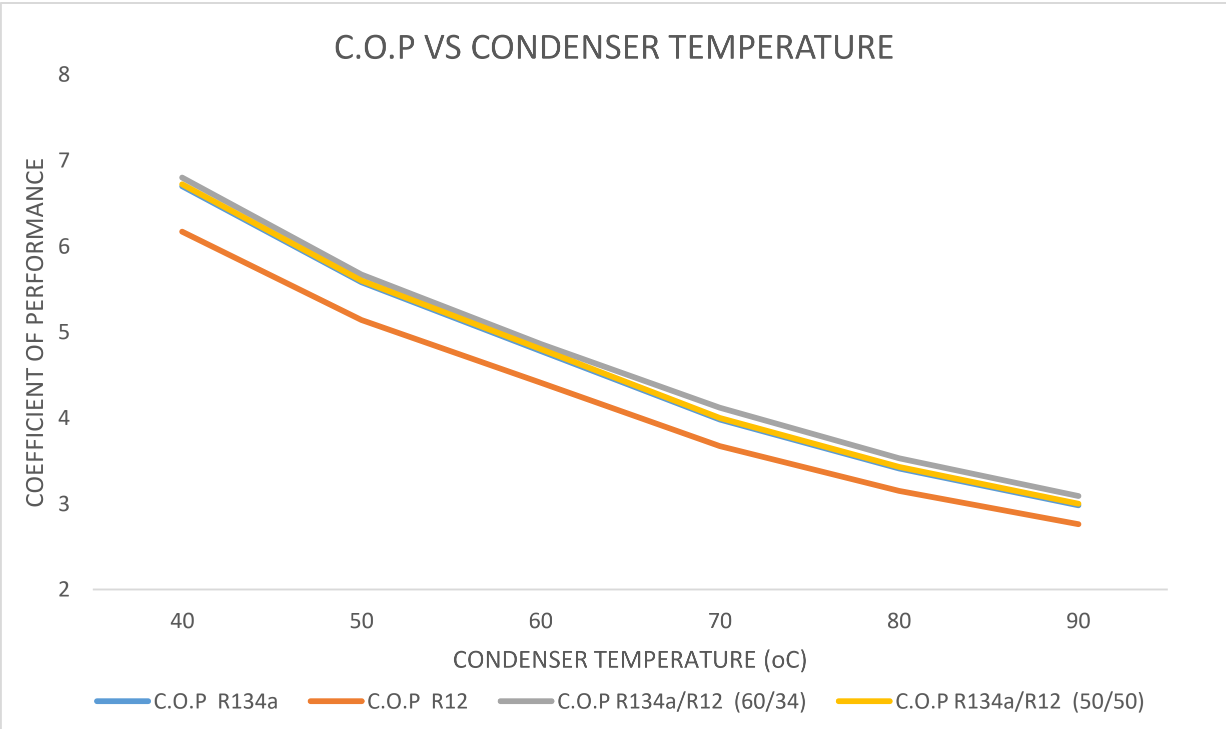

Figure 4.4: Graph of Coefficient of Performance (C.O.P) of refrigerants vs Condenser Temperature.

CONCLUSION

The main aim of this study is to fabricate the refrigerant blending rig and investigate the performance of various refrigerant blends. The apparatus was successfully commissioned using refrigerant R134a. The operating range of the refrigeration unit is within -20 to 100^{o}C, sealing under vacuum pressures of 26.6 kPa at room temperature and a maximum pressure limit of 1.9 MPa. Results from the trial runs carried out with refrigerant R134a indicate that the equipment could produce accurate and reliable refrigeration effect, therefore, deeming it to be suitable for refrigeration studies. The commercial refrigerant blends studied in the project were R134a/R12 (66/34 by wt %) and R134a/R12 (50/50 by wt %). In comparing the performance of commercial refrigerant blends, refrigerant blend R134a/R12 (66/34 by wt %) performed better than refrigerant R134a/R12 (50/50 by wt %). in the experimental analyses. The COP values for R134a, R12, R134a/R12 (66/34 by wt %) and R134a/R12 (50/50 by wt %). in the experimental studies were 4.78, 4.41, 4.86 and 4.8 respectively. However, the performance of refrigerant R134a/R12 (66/34 by wt %) and R134a/R12 (50/50 by wt %) is comparable to that of refrigerant R134a, used as the reference refrigerant in this study. For this reason, refrigerant blend R134a/R12 (66/34 by wt %) is a potential candidate to be applied as a replacement of refrigerant R134a. Considering the blends synthesized for this study, refrigerant blend R134a/R12 (66/34 by wt %) displayed better performance than R134a/R125 (50/50 by wt %). The former had a COP value of 4.86, refrigeration effect of 0.34 kW and compressor power of 0.05 kW whereas the latter had COP value of 4.80, refrigeration effect of 0.24 kW and compressor power of 0.07 kW.

This outcome is favorable since R134a, with a lower GWP, constitutes a larger percentage in the blend. Thus, the R134a/R12 (66/34 by wt %) blend has a minimal environmental impact compared to the R134a/R125 (50/50 by wt %) blend. Moreover, the synthesized R134a/R12 blends displayed better performance than the pure refrigerants (R134a, COP value of 4.78 and R12, COP value of 4.41) in the comparative analysis of the experimental results. Incorporating the environmental factors R134a/R12 (66/34 by wt %) with GWP value of just above 1200 had an overall more desirable performance than R507 with a GWP value of 1300.

It was also observed that compressor speed, ambient temperature, evaporator temperature and condenser temperature affect the coefficient of performance of the refrigeration system. Increase of compressor speed, ambient temperature and condenser temperature reduces the Coefficient of performance of the refrigeration system and vice versa. While increase in the evaporator temperature increases the Coefficient of performance of the refrigeration system.

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