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ENERGY PERFORMANCE ASSESSMENT OF R134A/LPG BLEND AS REPLACEMENT OF R134A IN VAPOR COMPRESSION REFRIGERATION SYSTEM

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Abstract: In this study, a theoretical performance analysis of vapor compression refrigeration system using R134a and R134a/LPG blends (in the ratio of 18:82,28:72, 38:62,48:52 by weight) as a refrigerant was carried out using the coefficient of performance (COP), as the main performance parameter. The theoretical analysis of refrigerants has been executed by varying the evaporator temperatures from -30°C to 0°C and at three different condensation temperatures, particularly 40°C, 50°C and 60°C.The theoretical results obtained with R134a are taken as the baseline for comparison. The optimum composition of R134a/LPG blend was selected based on the theoretical performance results, such that system yield highest coefficient of performance over wide range operating conditions.In addition to this, other performance parameters like the mass flow rate, the compressor power consumption, the compressor discharge temperature and total equivalent global warming impact (TEGWI) of R134a/LPG blend (optimum composition) are compared with R134a.The results revealed that vapor pressure characteristics of R134/LPG (28:72) blend was found to be closer to R134a over a broad range of evaporator temperatures so without any modifications R134a compressor can be used for R134/LPG (28:72) blend. The COP of R134/LPG (28:72) blend was also found to be highest and it is higher than that of R134a by about 15.6–27.8%. The compressor power requirement and TEGWI of R134/LPG (28:72) blend were found to be lower than that of R134a by about 24.5–49.6% and 36.34% lower respectively at all operating temperatures. The compressor discharge temperature of R134a/LPG (28:72) blend is also found lower than that of R134a over a wide range of evaporator temperature due to which compressor better life may be expected. The outcomes confirmed that R134a/LPG (28:72) blend is an energy efficient and environment-friendly substitute for R134a in VCRS.

Keywords: R134a, R134a/LPG, Domestic refrigerator, volumetric cooling capacity, Total equivalent global warming impact.

I. INTRODUCTION

Inside developing nation, most of the vapor compression refrigeration system (VCRS), keep running on halogenated refrigerants because of their excellent thermodynamic as well as thermo-physical properties in addition to the low price. A single useful way to reduce this sort of greenhouse gas emissions can be using energy efficient and environmentally friendly refrigerants [1]. In accordance with the Kyoto Protocol to the United Nations Framework Convention on Climate Change, the particular emission of hydrofluorocarbon (HFC) refrigerants is required to possibly be lessened. Moreover protecting against the particular loss of R134a by refrigeration devices seriously isn't very easily attainable. Consequently, to obtain environmentally safe practices, R134a are going to be prohibited quickly. Also, there are a few additional difficulties connected with R134a for example, high global warming potential (GWP) of 1430 [2] and its immiscible nature along with

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conventional mineral oils [3]. For this reason, Polyolester oil (POE) is usually preferred for R134a systems. The high hygroscopic character of Polyolester oil requires stringent service practices to prevent the moisture absorption, therefore the necessity of long-term alternative refrigerants which meet the objectives of international protocols is obvious [4]. A large number of experimental and theoretical studies are found in literature pertaining to hydrocarbons (HC), HFC and their mixtures as alternatives to R134a by researchers from various parts of the world. Among the various studies of hydrocarbons mixture, LPG Liquefied petroleum gas (LPG) a blend of three hydrocarbons (30 % propane, 55 % n-butane, 15 % iso-butane) has given good results in comparison with R134a. Moreover, various researchers reported LPG as a drop-in substitute for R134a In the related work Fatouh and Kafafy [5] experimentally investigated the performance of domestic refrigerator working with Liquefied petroleum gas with different charges and capillary lengths in order to replace R134a. The results revealed that power consumption; pull-down time and pressure ratio of LPG refrigerator were lower than those of R134a by about 4.3%, 7.6%, and 5.5% respectively. Also, COP of LPG refrigerator was 7.6% higher than that of R134a. The reported results confirm that LPG is an attractive drop-in substitute for R134a in domestic refrigerators. Similarly, P.Srinivas et al. [6] experimentally accessed the performance of Liquefied petroleum gas (LPG) as a refrigerant in order to replace R134a in a domestic refrigerator. The Result revealed that LPG has the higher refrigeration effect and coefficient of performance (COP) whereas it required lesser work, as compared to R134a.In a similar work, Taiwo Babarinde et al. [7] experimentally investigated the performance of Vapour compression refrigeration system using LPG as replacement of R134a. The result revealed that COP of the system was enhanced by 9.5 % where power consumption was reduced by 12% when compared with R134a.

But LPG as an HC refrigerant has flammability difficulties which usually limit their usage in present refrigeration devices. Even so decrease in flammability may be accomplished simply by blending HC refrigerants along with HFC refrigerants [8] because it is quite possible to blend HC refrigerants along with HFC refrigerants [9]. Moreover, miscibility of HC/HFC blends with conventional mineral oil is also found good [10]. And the global warming potential (GWP) associated with HC/HFC blends is less than one-third of HFC if it is used alone [11]. Several works can be found in the literature presenting theoretical and experimental studies to determine the feasibility of direct substitution (or with slight modifications) using HFC/HC blend in the existing facilities In the related work HFC134a/HC600a (80:20 by weight) has been experimentally investigated [12] in a heat pump, and a 5% improvement in COP has been reported. In addition S.Joseph Sekhar et al.[13] have been experimentally investigated HFC134a/HC290/HC600a in the household refrigerator without changing the mineral oil, 4- 11% reduction in energy consumption and 3 to 8% improvement in the COP has been reported. .Similarly Shekhar and Lal [3] experimental investigated HFC134a/HC blend (91:9 by weight) in two low temperature systems and two medium temperature systems and the results reported that that 10–30% and 5–15% reduction in energy consumption in medium and low-temperature system, respectively also oil miscibility of the HFC134a/HC blend with mineral oil was also found to be good. In other work, Ravi Kumar and Mohan Lal [14] tested R134a/R600a/R290 refrigerant mixture in an automobile air-conditioning system with mineral oil as lubricant and test results showed that R134a/R600a/R290 performed better than the existing system. In the similar work, T.S Ravikumar and Mohan Lal [15] also tested R134a/R600a/R290 blend with mineral oil in automobile air conditioning system and tested results shows that it perform on par with the existing system and this mixture could perform better than R134a along with PAG as lubricating oil. In other work, R134a/R600a/R406a mixtures have been experimentally investigated by Akintunde [16] in the domestic refrigerator as replacement of R12, and a 10.5 % improvement in COP has been reported with R134a/R600a (50:50) under similar operating conditions.

Both R134a (HFC) and LPG (HC) refrigerants have high GWP, immiscibility with conventional mineral oils and flammability issues respectively, these issues can be overcome by mixing the R134a and LPG with an appropriate mass fraction. Moreover, the literature referred to this paper focused on the use HC/HFC blends as refrigerants in the existing facilities. Hence the present investigations of R134a/LPG an HC/HFC blend, as a replacement for refrigerant R134a in a vapor compression system was carried out. In this work, the optimum composition of R134a/LPG blend was selected based on the theoretical performance results, such that system yield highest coefficient of performance over wide range operating conditions.In addition to this, other performance parameters like the mass flow rate, the compressor power consumption, the compressor discharge temperature and total equivalent global warming impact (TEGWI) of R134a/LPG blend (optimum composition) are compared with R134a.The theoretical analysis of R134a/LPG has been executed varying the evaporator temperature and the condenser temperature. The results obtained with R134a are taken as the baseline for comparison.

II. THERMOPHYSICAL AND THERMODYNAMICS PROPERTIES OF R134A/LPG AND R134A

Purposed alternative refrigerants i.e. R134a/LPG (18:82), R134a/LPG (28:72), R134a/LPG (38:62) , R134a/LPG (48:52) composed of R134a and LPG in mass fractions of 28:72, 28:72, 38:62, 48:52 respectively .Moreover, LPG used in the purpose R134a/LPG mixture composed of (30 % propane, 55 % n-butane, 15 % iso-butane). The variation of vapor pressure characteristics of R134a/LPG blends and R134a with evaporator temperature for condenser temperature of 40^0C is shown in Figure 1. It is observed from the figure that the vapor pressure characteristics of R134a/LPG (28:72) blend closely matches with R134a for the same operating temperatures as shown in Figure1. Thus the proposed alternative R134a/LPG (28:72) blend can be used as drop-in replacement for R134a in a vapor compression refrigeration system. The thermo-physical properties such as liquid density, liquid viscosity, latent heat and thermal conductivity of R134a/LPG (28:72) over a wide range of evaporator temperatures have been studied and compared with R134a using REFPROP 9.1 software as shown in figs. 2-4. Since the liquid density and viscosity of R134a/LPG (28:72) is found lower than that of R134a as shown in figs 2-3, due to which its charge requirement significantly reduced when R134a compressor is used. Moreover lower the liquid viscosity of refrigerant also results in lesser power consumption and high heat transfer coefficient [1]. Also, due to the lesser viscosity of R134a/LPG (28:72) blend higher capillary lengths are required for the same pressure drop as compared with R134a [17-18]. The latent heat and thermal conductivity of R134a/LPG (28:72) are found higher than R134a as shown in figs 4-5, due to which higher refrigeration capacity and higher heat transfer coefficients are expected in the evaporator and condenser with R134a/LPG (28:72) blend. R134a/LPG (28:72) blend is zeotropic in nature and its properties such as latent heat, molecular weight, critical temperature, were calculated with REFPROP 9.1 and tabulated in Table.1. R134a/LPG (28:72) blend is zoetrope in nature and its properties such as molecular weight, boiling point, critical temperature, ozone depletion potential (ODP) and Global warming potential (GWP) in comparison with R134a were calculated with REFPROP 9.1 and tabulated in Table.1.The data tabulated in the demonstrated that R134a/LPG blends have zero ODP, low GWP in comparison with R134a.

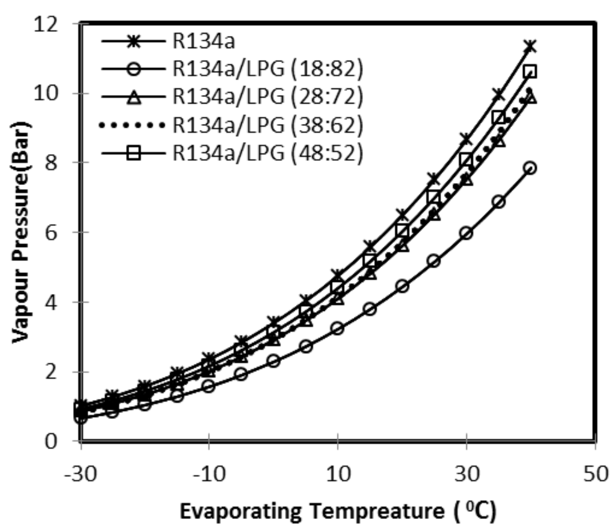

Figure 1: Vapor pressure variation with temperature

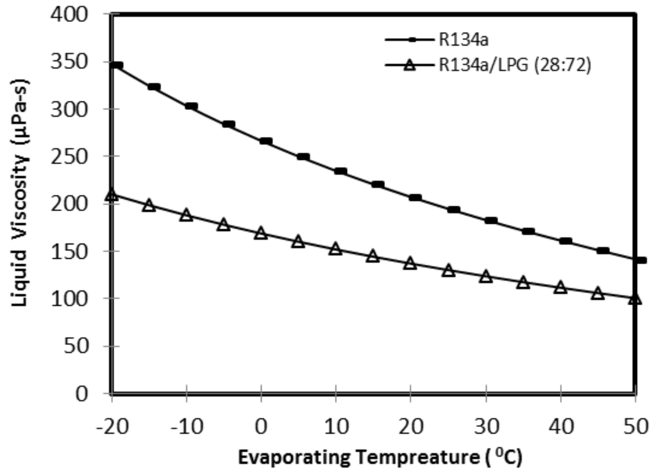

Figure 2: Liquid viscosity variation with temperature.

III. TOTAL EQUIVALENT GLOBAL INCREASED TEMPERATURE IMPACT (TEGWI).

The environmental impact as a result of the refrigerant throughout in the course of its whole life could be represented in terms of TEGWI [19]. TEGWI can be estimated as follows

$$TEGWI = \{(GWP \times m \times n \times L) + (GWP(1 - \alpha) \times m) + 365 \times N (n \times E \times \beta)\} \quad (1)$$

In this GWP is the 100-year global warming potential of refrigerant. Where m (in Kg) is the mass of refrigerant, L (in Kg/year) is the leakage rate of the system, N (in years) is the life of the system, and is the system running time per day, E (in kWh/year) is the action consumption per the calendar year, α is the recycling factor involving refrigerants its value is assumed as 75%, and β is the CO_2 emission factor with regard to electricity generation and its value is assumed being 0.89 kg of CO_2 /kWh in this work [9].

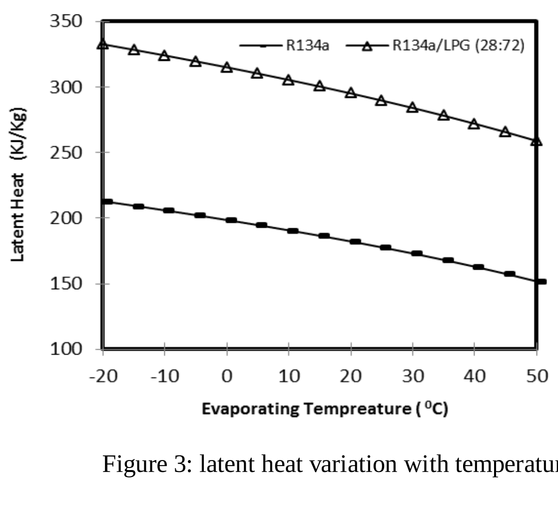

Figure 3: latent heat variation with temperature

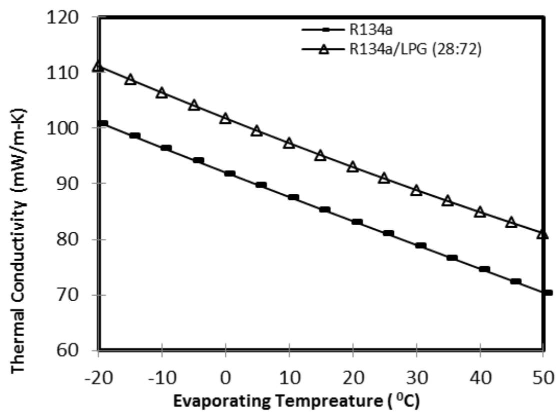

Figure 4: Thermophysical properties variation with evaporating temperature.

IV. VAPOUR COMPRESSION REFRIGERATION SYSTEM (VCRS)

The vapor compression refrigeration cycle on p-h diagram illustrated in Fig 6.The particular refrigeration system consists of four major elements: condenser, evaporator, compressor as well as an expansion device seeing that shown in Fig. 5. Inside the evaporator, the liquid refrigerant vaporizes simply by absorbing latent heat from the material being cooled down, and the caused low-pressure vapor refrigerant then passes the evaporator to the compressor. The compressor would be the heart of your refrigeration system. It pumps as well as circulates refrigerant through the system and supplies the required force to maintain system operation. The compressor raises the refrigerant pressure and hence the temperature, help heat rejection at a higher temperature inside the condenser. The condenser is often a device used pertaining to removing heat from the refrigeration system into a medium that includes a lower temperature than the refrigerant in the condenser. The high-pressure liquid refrigerant from the condenser passes through the evaporator through an expansion device or maybe a restrictor that reduces the pressure of the refrigerant to minimal pressure existing inside the evaporator. The expansion unit regulates or handles the flow involving liquid refrigerant towards the evaporator

Taking into consideration the cycle on p-h diagram throughout Fig. 7, the following assumptions are made:

1. An evaporation at constant pressure and temperature (Pe) & (T_e) respectively in the evaporator from point 4 to 1. The heat absorbed by the refrigerant in the particular evaporator or refrigerating effect (Q, KJ/kg) will be given as:

   $$Q = h_1 - h_4 \qquad (2)$$

   In which, h_1 = refrigerant specific enthalpy at the evaporator outlet (KJ/kg); and h_4 = refrigerant specific enthalpy at the evaporator inlet (KJ/kg).
2. An isentropic compression process in the compressor, from point 1 to 2. The compressor work input (W_C, KJ/kg) is

   $$W_c = h_2 - h_1 \qquad (3)$$

Where h_2 = refrigerant specific enthalpy at the compressor outlet (KJ/kg).

Table 1. Properties of R134a and R134a/LPG blend

Refrigerant	Boiling point (°C)	Molecular Weight (Kg/Kmol)	Critical temperature (°C)	ODP	GWP
R134a	-26.3	102.3	102.03	0	1430
R134a/LPG **(28:72)**	-33.15	56.65	121.11	0	367

3. Some sort of desuperheating of constant pressure (P_c) from compressor discharge temperatures (T_2) at position 2 to condenser temperatures (T_c) at position 2′, followed by a condensation at both constant temperatures (T_c) and constant pressure (P_c) from point 2′ to 3. The rejected heat in the particular condenser (Qc, KJ/kg) will be.

$$Q_c = h_2 - h_3 \quad (4)$$

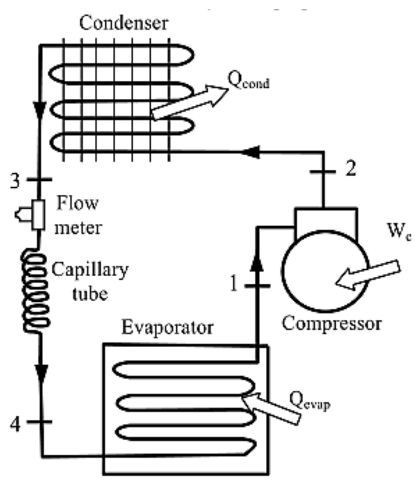

Fig 5. Components of refrigeration cycle

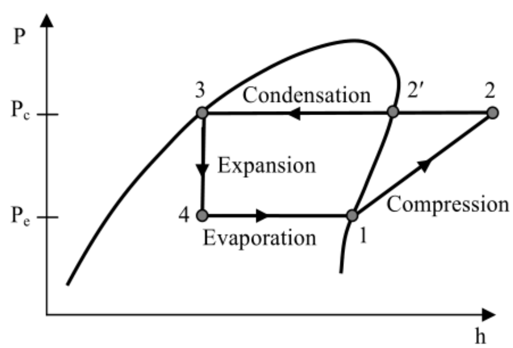

Fig 6. Vapor compression refrigeration cycle on p-h diagram

4. An expansion at constant enthalpy (isenthalpic) in the throttling valve from point 3 to point 4.

$$h_3 = h_4 \quad (5)$$

Where h_3 = refrigerant specific enthalpy at the condenser outlet (KJ/kg).The coefficient of performance (COP) is obtained as the ratio of Eq. (2) To Eq. (3). As it is the refrigerating effect produced per unit of compressor work required; therefore, COP is obtained as:

$$\mathrm{COP} = \frac{\mathrm{RE}}{\mathrm{W_C}} \quad (6)$$

The assumptions required for the simulation of vapor compression refrigerator are following:

1. The operation is steady state.
2. There is no pressure loss in the pipes, i.e. pressure change occurs only in the capillary tube and compressor.
3. There are no gain and loss of heat.
4. The compressor ideal isentropic efficiency and ideal volumetric efficiency of 75 % [6]
5. The Compressor speed (N) is assumed as 2900 RPM.

6. Stroke volume of the compressor (V_{dis}) is 6.14 cm^3 /rev whereas Clearance volume is 0.04cm^3.
7. The refrigerator life is 15 years with 3% of leakage rate.

V. RESULTS AND DISCUSSIONS

The outcomes obtained in this theoretical assessment regarding domestic refrigerators employing R134a and R134a/LPG blend as refrigerant with three different condensing temperature ranges, i.e. 40, 50 along with 60 °C, over a wide range of evaporator temperatures, which covers nearly all thermostat settings in domestic refrigerator i.e. −30°C to 0°C, are discussed in this study. The assessment in the VCRS was made according to ISO8187.

5.1 Coefficient of performance (COP) variation

The COP variation of systems working with R134a and R134a/LPG blends (in the ratio of 18:82, 28:72, 38:62, 48:52 by weight) against the wide range of evaporating temperatures i.e.-30^{o}C to 10^{o}C is shown in Fig 9-10. It is revealed from the figure that the COP of system using R134a/LPG (28:72) blend as refrigerant is highest among all examined refrigerants and it is 22.87%, 18.66%, and 15.68 % higher than that of R134a system at condensing temperatures by 40°C, 50°C and 60°C, respectively because of its smaller compressor work requirement and higher refrigeration capacity as compared to R134a [20]. It is also observed from the figure that there is about 5.5% and 6.2% increase in COP of both R134a and R134a/LPG (28:72) systems respectively, with an increase in evaporator temperature from −30°C to 0°C.

5.2 Refrigerant mass flow rate variation

The variations of mass flow rate against the evaporating temperatures regarding two examined refrigerants i.e. R134a and R134a/LPG (28:72) blend are shown in Fig.8. It is observed from the figure that the mass flow rate of R134a/LPG (28:72) blend was less than that of R134a by means of about 48.45–50.50 % because of its lower liquid density. Hence lower compressor work requirement to be expected with R134a/LPG (28:72). Because mass flow rate variation in terms of condensing temperature was observed small in this particular work due to which the variation regarding the condensing temperature of the two assessed refrigerants is actually neglected**.** Moreover, theoretical results also revealed that refrigerant mass flow rate increased with increased in the evaporator temperature**.**

5.3 Total equivalent global warming impact variation

At condensing and evaporator temperatures of 50 °C and −15°C respectively, and for 15 years lifetime of the vapor compression refrigeration system, the total equivalent global warming impact of the R134a and R134a/LPG (28:72) blend as refrigerants are calculated. The total equivalent global warming impact of the both the refrigerants are compared and demonstrated in Fig.15. Because of the higher energy efficiency and a lower global warming potential of R134a/LPG (28:72) blend its total equivalent global warming impact (TEGWI) was found to be lower than that of R134a by about 36.34 %**.**

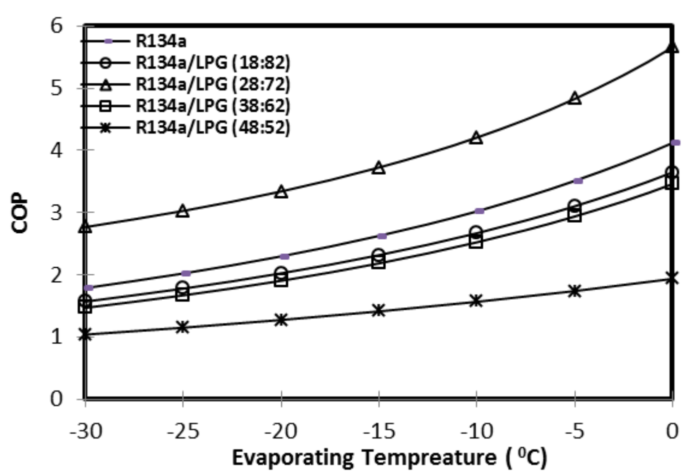

Figure 7: COP variation of systems using R134a/LPG blends and R134a with evaporator temperature, at 40^{o}C condenser temperature.

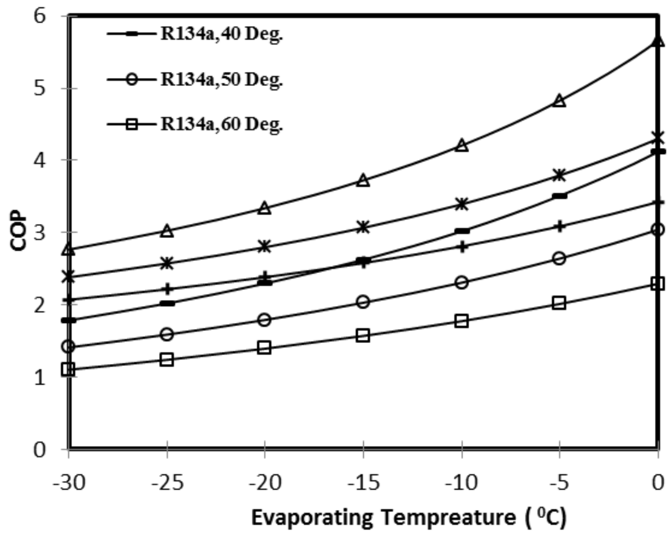

Figure 8: COP variation of R134a/LPG (28:72) and R134a systems with evaporator temperature, at 40^{o}C, 50^{o}C and 60^{o}C condenser temperatures.

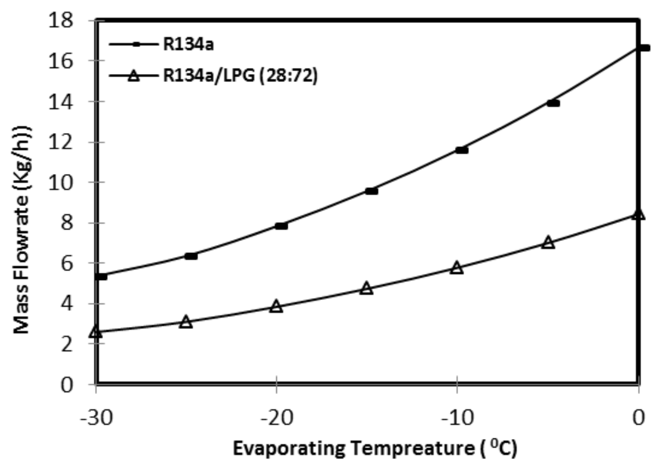

Figure 9: Mass flow rate variation with temperature

5.4 Compressor power consumption variation

Compressor power requirement of a vapor compression refrigeration system using R134a and R134a/LPG (28:72) blend as refrigerants are compared and shown in Fig.12.It is revealed from the figure that the compressor power requirement of vapor compression refrigeration system working with R134a/LPG (28:72) was found to be lower than that of R134a by 28.22−49.66 %, 26.43−45.70 % and 24.58−38.96% at condensing temperatures of 40°C, 50°C and 60°C, respectively for a range of evaporator temperatures between −30°C and 0°C.In addition to this, the figure depicts that the compressor power consumption of the Vapour compression refrigeration system increase with an increase in evaporator temperature because of increase in mass flow rate of refrigerants[20].

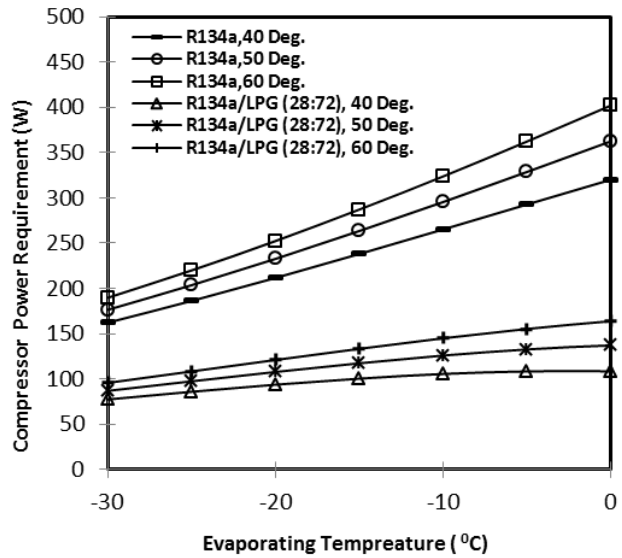

Figure 10: Compressor power variation with temperature

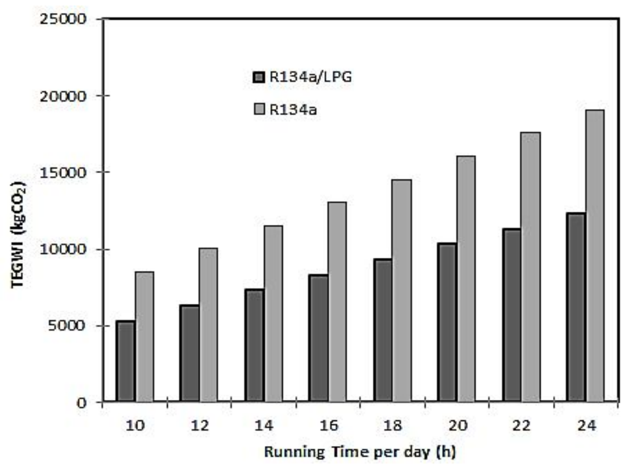

Figure 51: TEGWI variation with running time per day.

5.5 Compressor discharge temperature variation

One of the important aspects that should be taken into consideration is the discharge temperature of a refrigerant from the compressor because it affects the life the compressor. The properties of lubricants used in the compressor will affect due to higher compressor discharge temperature. The compressor discharge temperature comparison with both R134a and R134a/LPG (28:72) blend as refrigerants are depicted in Fig. 13. The compressor discharge temperature with R134a/LPG (28:72) blend was found to be lower than that of R134a by 5°C, at the evaporating temperature of -30°C and condensing temperatures of 40, 50 and 60 °C. It is also observed from the figure that the compressor discharge temperature with R134a/LPG (28:72) blend as refrigerant is less than that of R134a over a wide range of evaporator temperature ranges between -30 to 0^0C, due to which better life of the compressor may be expected if R134a/LPG blend replaces R134a . In addition to this it is also observed that, increase in the compressor discharge temperature with evaporator temperature from 0 to -30°C is approximately 15 °C and 14 °C for R134a and R134a/LPG (28:72) blend, respectively.

5.6 Mixture behaviors of R134a/LPG (28:72) blend.

The R134a/LPG (28:72) blend behaves like a Zoetrope mixture. As a refrigerant R134a has been encountered with a major issue that is its immiscible nature along with conventional mineral oil, thus, Polyolester oil (POE) is usually preferred for R134a systems but its high hygroscopic character requires stringent service practices to prevent the moisture absorption. This problem faced with R134a can be tackled by mixing it with LPG as that of hydrocarbon mixture (HC) because miscibility of HC/HFC blends with conventional mineral oil is good [10]. Thus, the proposed R134a/LPG blend can be used in compressors with conventional mineral oil as lubricants and as a result of which usage of synthetic oil can be eliminated**.**

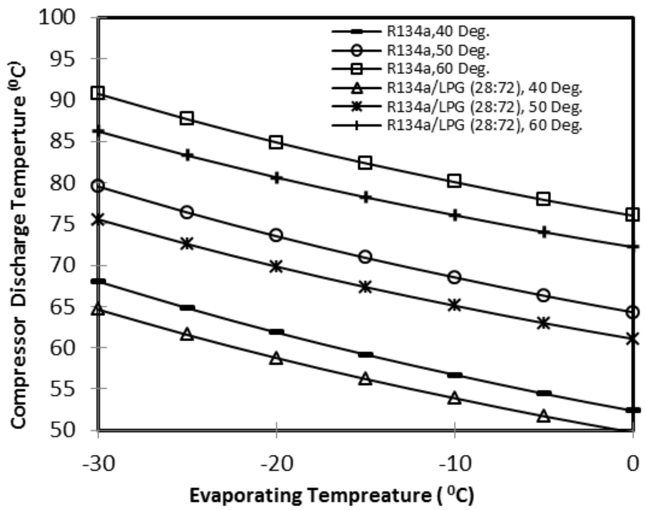

Figure 12: Compressor discharge temperature variation with temperature

VI. CONCLUSIONS

After the energy performance analysis of vapor compression refrigeration system using R134a and R134a/LPG blends (in the ratio of 18:82, 28:72, 38:62, 48:52 by weight) as refrigerant the following conclusions have been made.

- Vapor pressure characteristics of R134a/LPG (28:72) closely match with R134a thus same R134a compressor can be used.
- R134a/LPG blends have a low value of global warming potential as compared with R134a and for R134a/LPG (28:72) its value is about 367.
- The coefficient of performance of the vapor compression refrigeration system utilizing R134a/LPG (28:72) blend as refrigerant is 22.87%, 18.66%, and 15.68 % higher than that of R134a at condensing temperatures by 40, 50 and 60°C, respectively.
- The compressor work requirement of vapor compression refrigeration system working with R134a/LPG (28:72) blend was found to be lower than that of R134a by 28.2–49.7%, 26.43–45.7% and 24.5–38.9% at condensing temperatures of 40°C, 50°C and 60°C, respectively.
- The compressor discharge temperature of R134a/LPG (28:72) blend is lower than that of R134a over a wide range of evaporator temperature due to which better compressor life may be expected.
- The total equivalent global warming impact (TEGWI) of R134a/LPG (28:72) blend was observed to be lower than that of R134a by about 36.34 % due to its a lower global warming potential and the higher energy efficiency.
- As miscibility regarding HC/HFC blends with conventional mineral oil may be found good. Hence, the proposed R134a/LPG blend can be employed in compressors with conventional mineral oil as a lubricant a consequence of which usage of synthetic oil may be eliminated.

The above outcomes verified that 28:72 is an optimum composition for R134a/LPG blends and the R134a/LPG blend in the ratio of 28:72 by weight is energy efficient as well as an environment-friendly replacement for R134a in the existing facility without changing the R134a compressor.

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