ENERGY PERFORMANCE OF ECO-FRIENDLY RE170 AND R510A REFRIGERANTS AS ALTERNATIVES TO R134a IN VAPOUR COMPRESSION REFRIGERATION SYSTEM

B.O. Bolaji and Z. Huan
Department of Mechanical Engineering, Faculty of Engineering and Built Environment, Tshwane University of Technology, Pretoria, South Africa

ABSTRACT

Ozone depletion and the atmospheric greenhouse effect due to refrigerant emissions have led to drastic changes in the refrigeration and air-conditioning technology. For this reason, environmentally benign, 'natural' refrigerants have attracted a considerable attention. In the group of natural refrigerants, hydrocarbons are the most closely related to the HFCs with similar thermodynamic and transport properties, which make them suitable as substitute refrigerants in the existing HFC systems without any major changes in the design. In this paper, the energy performance of two eco-friendly hydrocarbon refrigerants (RE170 and R510A) with zero ozone depletion potential and negligible global warming potential in vapour compression refrigeration system is investigated theoretically under different operating conditions. The results obtained showed that RE170 and R510A have similar saturation vapour pressure characteristics and thermophysical properties with R134a. The Energy performance of both R510A and RE170 was better than that of R134a. The average COPs of R510A and RE170 are higher than that of R134a by 20.7 and 13.1%, respectively. The lowest discharge temperature and energy consumption, and highest COP of the system were obtained using R510A. Generally, R510A and RE170 performed better than R134a; their energy consumptions are 20.4 and 14.1% less than that of R134a, respectively. Therefore, they can be used as drop-in substitutes in the existing R134a refrigeration systems.

1. INTRODUCTION

Chlorofluorocarbons (CFCs) and hydro-chlorofluorocarbons (HCFCs) have been the traditional refrigerants applied to domestic refrigeration applications, until 1990s, but due to the environmental concerns about their depletion of the earth's protective stratospheric ozone layer, CFCs have been phased out in developed countries since 1996, and from January 1st 2010, production and their use were prohibited completely all over the world [1, 2]. HCFCs will also be phased out internationally by 2020 and 2030 in developed and developing countries respectively [3-5].

In agreement with the Montreal protocol many refrigerants containing CFCs and HCFCs were increasingly replaced with hydro-fluorocarbons (HFCs) which have zero ozone depletion potential (ODP) but their global warming potential (GWP) is relatively high [6]. The HFC refrigerants are considered as one of the six target greenhouse gases under Kyoto protocol. Kyoto protocol was approved by many nations, called for the reduction in emissions of greenhouse gases including HFC refrigerants. The presence of fluorine atoms in HFC refrigerants is responsible for the major environmental impact (GWP) [1].

In addition to zero ozone depletion potential (ODP), the working fluids in refrigeration systems must also have very low global warming potential (GWP) [2]. Due to this new requirement, once again, industry will be forced to change refrigerants. This time from the newly introduced chlorine-free alternative refrigerants to those do not absorb the infrared re-radiation from the earth's surface. Also, additional stringent criteria relating to system efficiency will be necessary so that the new refrigerants do not cause additional CO_2 generation at the power source. For this reason, environmentally benign, 'natural' refrigerants have attracted a considerable attention.

The natural refrigerants are the naturally occurring substances such as ammonia, hydrocarbons, carbon dioxide, water and air. In this group, the hydrocarbons are most closely related to the HFCs. Their thermodynamic and transport properties are very similar to most HFCs currently used in refrigeration and air-conditioning systems, which make them suitable as substitute refrigerants in the existing HCFC and HFC systems without any major changes in the design [7-9].

Hydrocarbon refrigerants have been in use since 1867, and, in conjunction with ammonia, were the most widely used refrigerants prior to the introduction of chlorinated fluorocarbon refrigerants in the 1930s [10]. Interest in the use of hydrocarbons as refrigerants in domestic and commercial refrigerators has increased in the last decade [11]. Hydrocarbons have neither an ozone depletion effect nor a global warming effect [12]. They are non-toxic, cheap, plentiful and compatible with both mineral and synthetic oils [13]. It has been reported that the amount of charge associated with hydrocarbons was roughly half that of CFC refrigerants in refrigerators. Also, the tests conducted indicated that hydrocarbons are quite safe in domestic refrigerators due to the very small amounts involved [11]. The use of hydrocarbon refrigerants has the direct environmental advantage of a greatly reduced GWP when compared to HFC refrigerants.

The most important concern regarding the adoption of hydrocarbons as a refrigerant is their flammability. It should be remembered that millions of tonnes of hydrocarbons are used safely every year throughout the world for cooking, heating, powering vehicles and as aerosol propellants. In these industries, procedures and standards have been developed and adopted to ensure the safe use of the product. The same approach is also been followed by the refrigeration industry. Various applications have been developed in handling the flammability and safety problems such as using enhanced compact heat exchangers, optimizing system designs, reducing the charge of systems and establishing rules and regulations for the safety precautions [14, 15].

Many investigations have been conducted in the research into substitutes for R12 and R134a. Lee and Su [16] conducted an experimental study on the use of iso-butane in a domestic refrigerator. The results showed that the coefficients of performance were comparable with those obtained when R12 and R22 were used as refrigerants. Akash and Said [17] studied the performance of LPG obtained from the local market (30% propane, 55% n-butane and 15% iso-butane by mass) as an alternative refrigerant to R12 in domestic refrigerators. The experiments were done with LPG at various mass charges of 50, 80 and 100 g. The results showed that a mass charge of 80 g gave the best performance.

Wongwises and Chimres [18] presented an experimental study on the application of a mixture of propane, butane, and isobutene to replace R134a in a domestic refrigerator. The results showed that a mixture of 60% propane and 40% of butane was the most appropriate alternative refrigerant. Wongwises et al. [19] presented an experimental study on the application of hydrocarbon mixtures to replace R134a in automotive air conditioners. The hydrocarbons investigated are propane (R290), butane (R600), and isobutene (R600a). The measured data are obtained from an automotive air-conditioning test facility utilizing R134a as the refrigerant.

Chen and Yu [20] presented a new refrigeration cycle, introduced as an alternative refrigeration cycle applied in residential air-conditioners, using the binary nonazeotropic refrigerant mixture R32/R134a. As a result of the comparison between the conventional cycle configuration and the new one, the coefficient of performance (COP) increases by 8% to 9% compared to the conventional cycle configuration, and the volumetric refrigerating capacity is increased by approximately 9.5%.

The literature review revealed that many investigations have been performed on domestic refrigerators working with either R290 and R600a or their mixtures as drop in replacements for R134a. However, the possibility of using RE170 (dimethyl ether or DME) and R510A to replace R134a in domestic refrigerators needs to be investigated under different operating conditions. RE170 is a hydrocarbon refrigerant with zero ODP and very low GWP (Table 1). Also, R510A is an azeotropic refrigerant mixture composed of RE170 and R600a (88 and 12% in weight, respectively). Therefore, in the present study, energy performance characteristics of RE170 and R510A were evaluated theoretically using a standard vapour compression refrigeration system and compared with performance of baseline refrigerant (R134a) in the system.

Table 1: Environmental and thermophysical properties of investigated refrigerants [1, 21]

Properties	Refrigerants		
	R134a	RE170	R510A
Molar mass (kg/kmol)	102.03	46.07	47.24
Normal boiling point, NBP (^oC)	-26.07	-24.78	-23.21
Critical Temperature (^oC)	101.06	127.23	125.67
Ozone Depleting Potential (ODP)	0	0	0
Global warming potential (GWP)	1300	3	3

2. MATERIALS AND METHODS

2.1 COST AND AVAILABILITY OF HYDROCARBON REFRIGERANTS

Hydrocarbons gasses are naturally occurring substances that are obtained from refineries after distillation. Some of the gasses and their mixtures are available from international manufacturers mostly located in Europe [22]. The cost of hydrocarbon refrigerants is almost 33.3% less than the cost of halocarbon refrigerants (CFC, HCFC and HFC) as shown in Table 2. Also the amount of hydrocarbon refrigerants charged to a certain sized of refrigerating machine is almost 50% less than that of halocarbon refrigerants for the same refrigerating machine (Table 3).

Table 2: Cost comparison between hydrocarbon and halocarbon refrigerants (CFC, HCFC and HFC) [1]

Refrigerants	Cost range (US$/kg)	Cost reduction (%)
Halocarbon	0.9 – 5.0	–
Hydrocarbon	0.6 – 3.0	33.3%

Table 3: Comparison of quantity of refrigerant charged between hydrocarbon and halocarbon refrigerants

Refrigerating machine	Quantity charged (g)		Reference
	Halocarbon	Hydrocarbon	
240 litre freezer	160	80	[17]
283 litre refrigerator	100	60	[11]
400 litre refrigerator	140	70	[23]

2.2 VAPOUR COMPRESSION REFRIGERATION SYSTEM

Figure 1 shows vapour compression refrigeration cycle on p-h diagram. The refrigeration system is made up of four major components: condenser, evaporator, compressor and expansion device. In the evaporator, the liquid refrigerant vaporizes by absorbing latent heat from the material being cooled, and the resulting low pressure vapour refrigerant then passes from the evaporator to the compressor. The compressor is the heart of refrigeration system. It pumps and circulates refrigerant through the system, and supplies the necessary force to keep the system operating. It raises the refrigerant pressure and hence the temperature, to enable heat rejection at a higher temperature in the condenser.

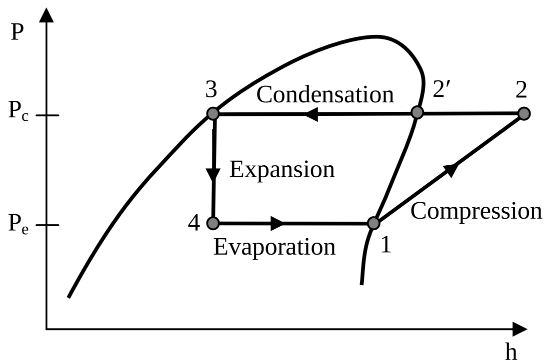

Figure 1: Vapour compression refrigeration cycle on p-h diagram

The condenser is a device used for removing heat from the refrigeration system to a medium which has lower temperature than the refrigerant in the condenser. The high pressure liquid refrigerant from the condenser passes into the evaporator through an expansion device or a restrictor that reduces the pressure of the refrigerant to low pressure existing in the evaporator. The expansion device regulates or controls the flow of liquid refrigerant to the evaporator.

Considering the cycle on p-h diagram in Figure 1, the following assumptions are made:

(i) An evaporation at constant pressure (P_e) and constant temperature (T_e) in the evaporator from point 4 to point 1. The heat absorbed by the refrigerant in the evaporator or refrigerating effect (Q_{evap}, kJ/kg) is given as:

$$Q_{evap} = (h_1 - h_4) \qquad (1)$$

where, h_1 = specific enthalpy of refrigerant at the outlet of evaporator (kJ/kg); and h_4 = specific enthalpy of refrigerant at the inlet of evaporator (kJ/kg).

(ii) An isentropic compression process in the compressor, from point 1 to point 2. The compressor work input (W_c, kJ/kg) is

$$W_c = (h_2 - h_1) \qquad (2)$$

where, h_2 = specific enthalpy of refrigerant at the outlet of compressor (kJ/kg).

(iii) A de-superheating at constant pressure (P_c) from compressor discharge temperature (T_2) at point 2 to condenser temperature (T_c) at point 2′, followed by a condensation at both constant temperature (T_c) and constant pressure (P_c) from point 2′ to point 3. The heat rejected in the condenser (Q_c, kJ/kg) is

$$Q_c = (h_2 - h_3) \qquad (3)$$

where, h_3 = specific enthalpy of refrigerant at the outlet of condenser (kJ/kg).

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

$$h_3 = h_4 \qquad (4)$$

The coefficient of performance (COP) is the refrigerating effect produced per unit of work required; therefore, COP is obtained as the ratio of Eq. (1) to Eq. (2):

$$COP_{ref} = \frac{Q_{evap}}{W_{comp}} \qquad (5)$$

2.3 DETERMINATION OF THERMODYNAMIC PROPERTIES OF REFRIGERANTS

The most fundamental of a working fluid's thermal properties that are needed for the prediction of a refrigerant system's performance are the pressure-volume-temperature (PvT) in an equilibrium state. Other properties, such as enthalpy and entropy as well as the Helmholtz and Gibbs functions, may be derived from a PvT correlation utilizing specific heat. There exists a myriad of equations-of-state, which have been classified into families. These equations have been used to develop the most widely used refrigerant database software known as REFPROP [21, 24]. It was developed and is maintained by the National Institute of Standards and Technology and is currently in its ninth edition. It uses several equations-of-state to correlate 33 single component refrigerants and 29 predefined mixtures, along with the ability to construct virtually any desired mixture of up to five components [25]. This software was used in this work to compute the properties of refrigerants.

3. RESULTS AND DISCUSSION

Figure 2 shows the comparison of the saturation vapour pressure curves for RE170, R510A and R134a. As shown in this figure, the saturation vapour pressure curves for the alternative refrigerants are very close to the vapour pressure curve of R134a refrigerant. This indicates that these refrigerants can exhibit similar properties and could be used as substitute for R134a.

The refrigerating effects of R134a, RE170 and R510A at varying evaporating temperature for condensing temperatures of 30, 40 and 50^{o}C are shown in Figures 3 to 5, respectively. As shown in these figures, refrigerating effect increases as the evaporating temperature increases for all the investigating refrigerants. This is due to the increase in latent heat energy of the refrigerant. Very high latent heat energy is desirable since the mass flow rate per unit of capacity is less. When the latent value is high, the energy efficiency and capacity of the compressor are greatly increased. This decreases the power consumption and also reduces the compressor displacement requirements that permit the use of smaller and more compact equipment.

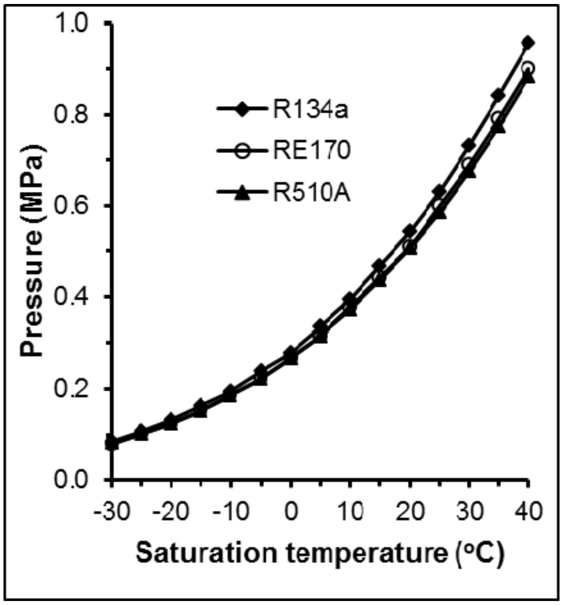

Figure 2: Saturation vapour pressure curves

The refrigerating effects of the alternative refrigerants (RE170 and R510A), as clearly shown in Figures 3 to 5, are higher than that of R134a. The average refrigerating effect for varying condensing temperature is shown in Figure 6. Refrigerating effect for the three refrigerants reduces as the condensing temperature increases. This is due to increase in the enthalpy of refrigerant at inlet to the evaporator as a result of the increase in the condensing temperature, which reduces the refrigerating effect. The average refrigerating effect obtained using RE170 and R510A were 38.7 and 32.6% higher than that of R134a, respectively.

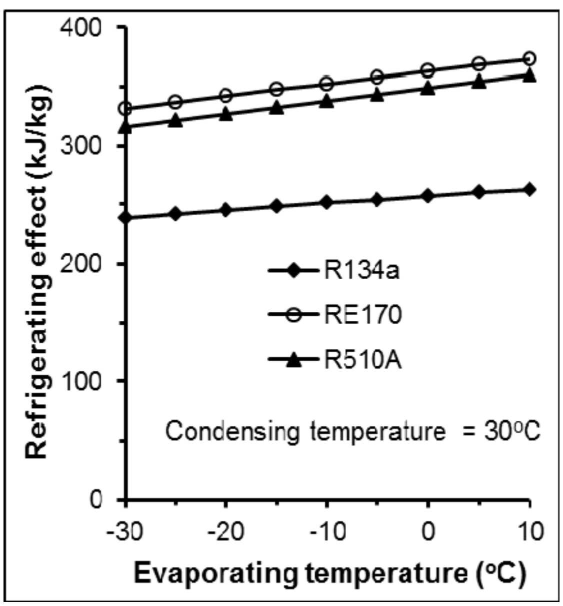

Figure 3: Refrigerating effect versus evaporating temperature at condensing temperature of 30°C.

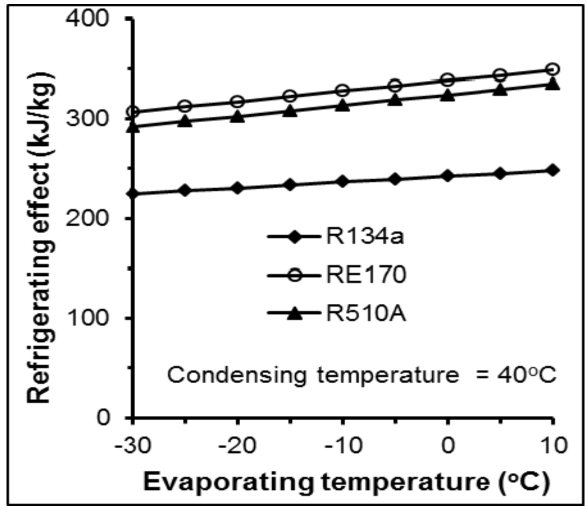

Figure 4: Refrigerating effect versus evaporating temperature at condensing temperature of 40°C.

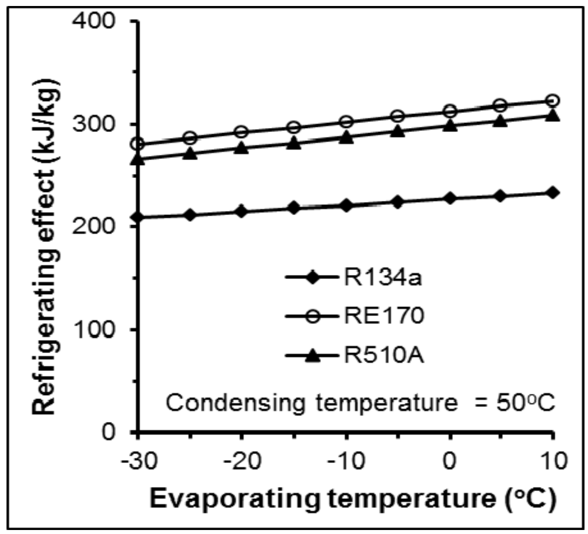

Figure 5: Refrigerating effect versus evaporating temperature at condensing temperature of 50°C.

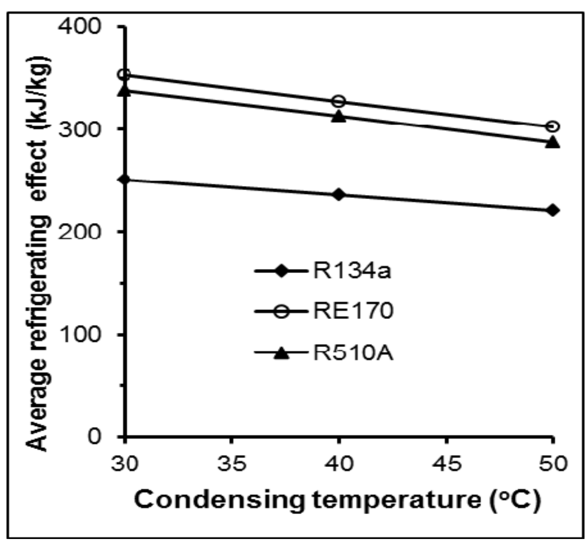

Figure 6: Variation of refrigerating effect with condensing temperature.

The compressor energy input for R134a and its two hydrocarbon alternative refrigerants at varying evaporating temperature for condensing temperature of 30, 40 and 50°C are shown in Figures 7 to 9, respectively. These figures clearly revealed that compressor energy input increases with increase in evaporating temperature. Similar trend and variations of compressor energy input were obtained for all the investigated refrigerants. The lowest compressor energy input was obtained using R134a, followed by R510A refrigerant. Figure 10 shows the variation of average compressor energy input as a function of condensing temperature. As shown in the figure, the compressor energy input of R510A is very close to that of R134a with average value of 7.4% higher, while that of RE170 is 19.3% higher than that of R134a.

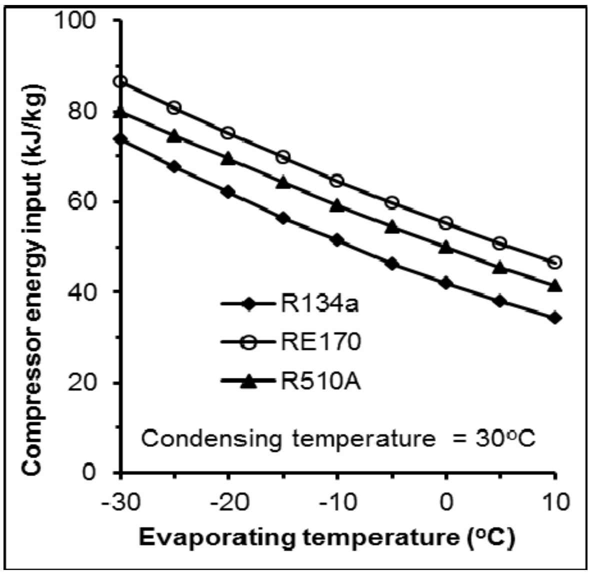

Figure 7: Compressor energy input versus evaporating temperature at condensing temperature of 30°C

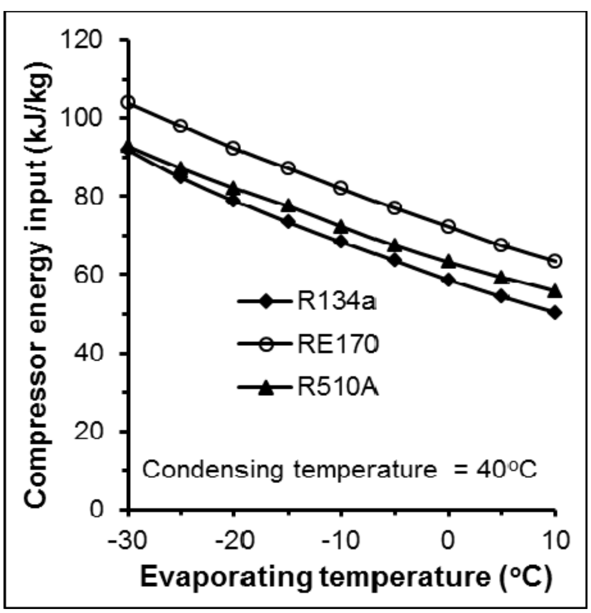

Figure 8: Compressor energy input versus evaporating temperature at condensing temperature of 40°C

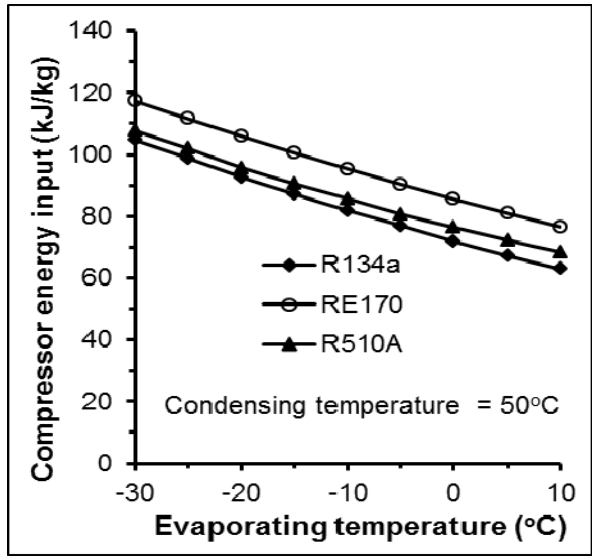

Figure 9: Compressor energy input versus evaporating temperature at condensing temperature of 50°C

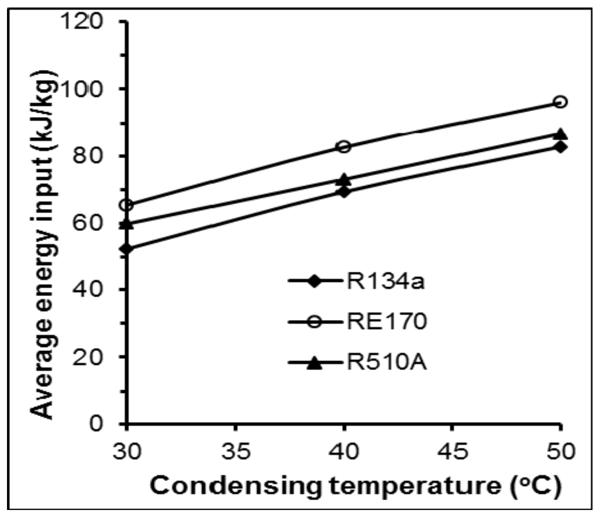

Figure 10: Variation of average compressor energy input with condensing temperature.

The coefficient of performance (COP) of a refrigeration cycle reflects the cycle performance and energy efficiency. It is the major criterion for selecting a new refrigerant as a substitute. Figures 11 to 13 show the COPs for RE170, R510A and R134a at varying evaporating temperature for condensing temperature of 30, 40 and 50°C, respectively. COP increases with increase in evaporating temperature. The COPs of RE170, and R510A are higher than that of R134a. Also, variation of the COP with condensing temperature is presented in Figure 14. It is clearly shown in this figure that when condensing temperature increases the COP reduces for all the three refrigerants. COP is inversely proportional to the power input through the compressor, therefore, increase in compressor power due to increase in condensing temperature reduces the COP of the system. The highest COP was obtained using R510A in the system. The average COPs obtained for R510A and RE170 were 20.7% and 13.1% higher than that of R134a, respectively.

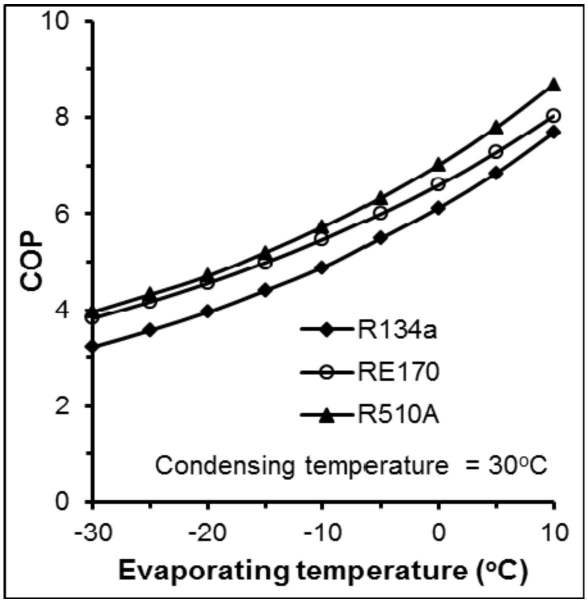

Figure 11: Variation of coefficient of performance (COP) with evaporating temperature for condensing temperature of 30°C

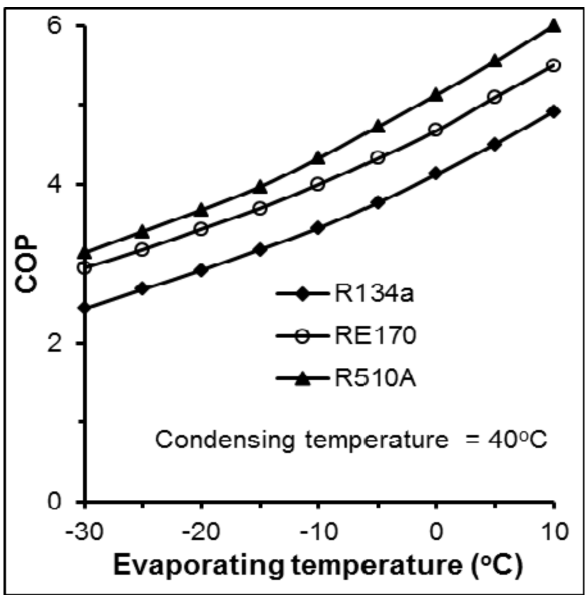

Figure 12: Variation of coefficient of performance (COP) with evaporating temperature for condensing temperature of 40°C

Figure 15 shows the curve of discharge pressure at varying condensing temperature for the three investigated refrigerant. The discharge pressure is an important parameter that affects the performance of a refrigerating system. It influences the stability of the lubricants and compressor components. Therefore, refrigerants with lower discharge pressure are more suitable alternative and better than those with high discharge pressure. R510A exhibited lower discharge pressure than other investigated refrigerants. The discharge pressure for R510A and RE170 are 13.3 and 11.6% lower than that of R134a, respectively.

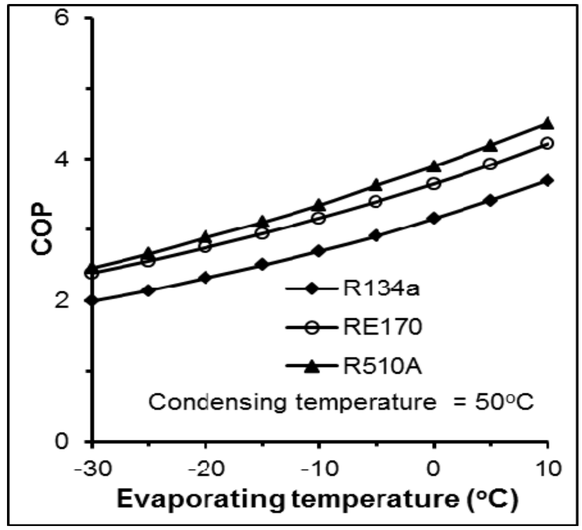

Figure 13: Coefficient of performance (COP) versus evaporating temperature at condensing temperature of 50°C

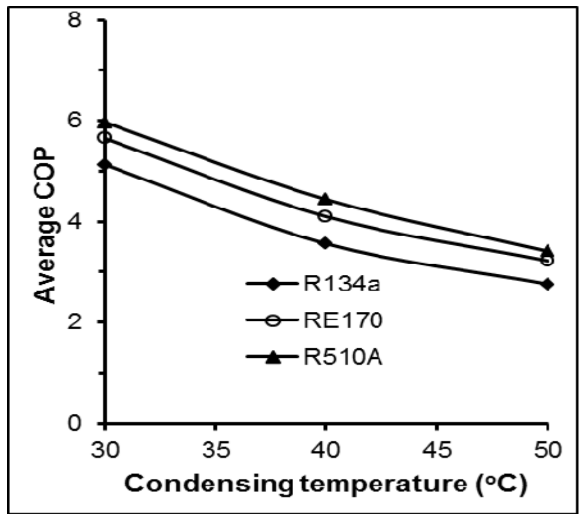

Figure 14: Variation of average coefficient of performance (COP) with condensing temperature.

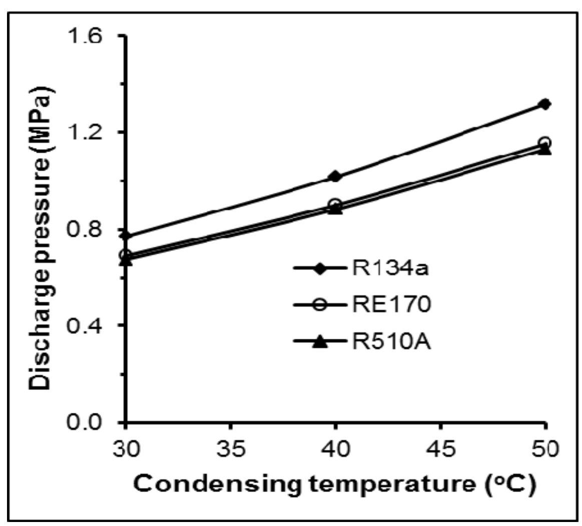

Figure 15: Variation of discharge pressure with condensing temperature.

The influence of evaporating temperature on the power consumption per ton of refrigeration (PPTR) at condensing temperature of 40°C for R134a and the two investigated alternative refrigerants is shown in Fig. 16. As shown in the figure, the PPTR reduces as the evaporator temperature increases for all the investigating refrigerants. Two alternative refrigerants exhibited lower energy consumption than R134a. In this result, R510A has emerged as the most energy efficient refrigerant among all the investigated refrigerants being the one that exhibited the lowest power consumption with the average PPTR of 20.4% less than that of R134a, while the value for RE170 is also 14.1% less than that of R134a.

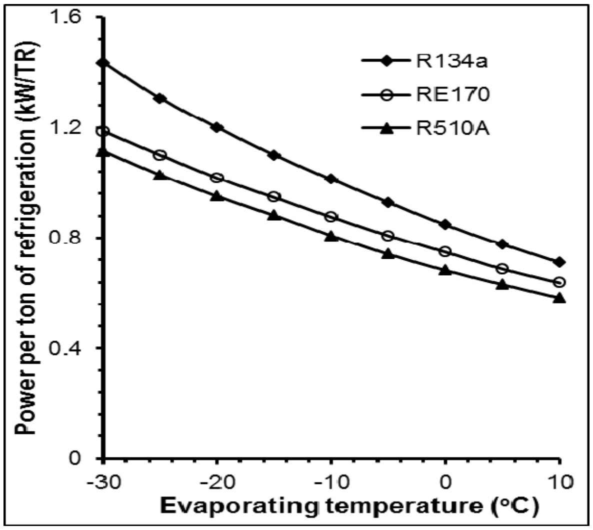

Figure 16: power per ton of refrigeration versus evaporating temperature at condensing temperature of 40°C.

4. CONCLUSION

Based on the investigation results, the following conclusions are drawn:

(i) The investigated alternative refrigerants (RE170 and R510A) have similar saturation vapour pressure characteristics and thermophysical properties with R134a, but also they are more environmentally-friendly refrigerants than R134a.

(ii) The Energy performance of both R510A and RE170 was better than that of R134a in all the operating conditions of a standard vapour compression refrigeration system.

(iii) The lowest discharge temperature and energy consumption, and highest coefficient of performance (COP) of the system were obtained using R510A.

(iv) The average COPs of R510A and RE170 are higher than that of R134a by 20.7% and 13.1% respectively.

(v) Generally, R510A and RE170 performed better than R134a; their energy consumptions are 20.4 and 14.1% less than that of R134a, respectively. Therefore, they can be used as drop-in substitutes in the existing R134a refrigeration systems. The best performance is obtained using R510A in the system.

REFERENCES

[1] UNEP. *Study on the Potential for Hydrocarbon Replacements in Existing Domestic and Small Commercial Refrigeration Appliances*. United Nations Environment Programme (UNEP), France. http://www.unepie.org Accessed on May 10, 2012.

[2] Bolaji, B.O., Akintunde, M.A. and Falade, T.O.: "Comparative analysis of performance of three ozone-friendly HFC refrigerants in a vapour compression refrigerator" *Journal of Sustainable Energy and Environment*, Vol. 2, No. 2, 2011, pp. 61-64.

[3] Devotta, S., Padalkar, A.S. and Sane, N.K.: "Performance assessment of HCFC-22 window air conditioner retrofitted with R407C" *Applied Thermal Engineering*, Vol. 25, 2005, pp. 2937-2949.

[4] McMullan, J.T.: "Refrigeration and the environment issues and strategies for the future" *International Journal of Refrigeration*, Vol. 25, 2002, pp. 89-99.

[5] Fatouh, M., Ibrahim, T.A. and Mostafa, A.: "Performance assessment of a direct expansion air conditioner working with R407C as an R22 alternative" *Applied Thermal Engineering*, Vol. 30, No. 2, 2010, pp. 127-133.

[6] Bolaji, B.O.: "Performance investigation of ozone-friendly R404A and R507 refrigerants as alternatives to R22 in a window air-conditioner" *Energy and Buildings*, Vol. 43, 2011, pp. 3139-3143.

[7] Castro, J.B., Urchueguia, J.F., Corberan, J.M. and Gonzalvez, J.: "Optimized design of a heat exchanger for an air-to-water reversible heat pump working with propane (R290) as refrigerant: modelling analysis and experimental observations" *Applied Thermal Engineering*, Vol. 25, 2005. pp. 2450-2462.

[8] Park, K.J. and Jung, D.S.: "Thermodynamic performance of HCFC22 alternative refrigerants for residential air-conditioning applications" *Energy and Buildings*, Vol. 39, 2007, pp. 675-680.

[9] Palm, B.: "Refrigeration systems with minimum charge of refrigerant" *Applied Thermal Engineering*, Vol. 27, 2007, pp. 1693-1701.

[10] Bhatti, M.S.: "A Historical Look at Chlorofluorocarbon Refrigerants" *ASHRAE Transactions Part 1*, 1999, pp. 1186-1206.

[11] Fatouh, M. and El-Kafafy, M.: "Assessment of Propane/ Commercial Butane Mixtures as Possible Alternatives to R134a in Domestic Refrigerators" *Energy Conversion and Management*, Vol. 47, 2006, pp. 2644-2658.

[12] ASHRAE. *Thermophysical Properties of Refrigerants*, Atlanta (GA), Chapter 20, ASHRAE Fundamental, Inc., 2001, pp. 1-67.

[13] Palm, B.: "Hydrocarbons as refrigerants in small heat pump and refrigeration systems - a review" *International Journal of Refrigeration*, Vol. 31, 2008, pp. 552-563.

[14] Thonon, B.: "A review of hydrocarbon two-phase heat transfer in compact heat exchangers and

enhanced geometries" *International Journal of Refrigeration*, Vol. 31, 2008, pp. 633-642.

[15] Fernando, P., Palm, B., Lundqvist, P. and Granryd, E.: "Propane heat pump with low refrigerant charge: design and laboratory tests" *International Journal of Refrigeration*, Vol. 27, 2004, pp. 761-773.

[16] Lee, D., Ahna, V., Kim, Y., Changa, Y. and Nam, L.: "Experimental investigation on the drop-in performance of R407C as a substitute for R22 in a screw chiller with shell-and-tube heat exchangers" *International Journal of Refrigeration*, Vol. 25, No. 5, 2002, pp. 575-585.

[17] Akash, B.A. and Said, S.A.: "Assessment of LPG as a possible alternative to R-12 in domestic refrigerator" *Energy Conversion Management*, Vol. 44, 2003, pp. 381-388.

[18] Wongwises, S. and Chimres, N.: "Experimental study of hydrocarbon mixtures to replace HFC-134a in a domestic refrigerator" Energy Conversion and Management, Vol. 46, 2005, pp. 85-100.

[19] Wongwises, S., Kamboon, A. and Orachon, B.: "Experimental investigation of hydrocarbon mixtures to replace HFC-134a in an automotive air conditioning system" *Energy Conversion and Management*, Vol. 47, 2006, pp. 1644-1659.

[20] Chen, J. and Yu, J.: "Performance of new refrigerant cycle using refrigerant mixture R32/R134a for residential air-conditioner applications" *Energy and Buildings*, Vol. 40, 2008, pp. 2022-2027.

[21] Lemmon, E.W., McLinden, M.O. and Huber, M.L.: *NIST reference fluids thermodynamic and transport properties REFPROP 7.0*, National Institute of Standards and Technology (NIST), Gaithersburg, MD, USA, 2002.

[22] UNIDO. Preparing for HCFC phase-out: Fundamentals of uses, alternatives, implications and funding for Article 5 countries. United Nations Industrial Development Organization (UNIDO), Vienna, 2009.

[23] Sattar, M.A., Saidur, K. and Masjuki, H.H.: "Performance investigation of domestic refrigerator using pure hydrocarbons and blends of hydrocarbons as refrigerants" *Proceedings of World Academy of Science, Engineering and Technology*, Vol. 23, 2007, pp. 223-228

[24] Didion, D.A.: "The influence of the thermophysical fluid properties of the new ozone-safe refrigerants on performance" *International Journal of Applied Thermodynamics*, Vol. 2, 1999, pp. 19-35.

[25] Kumar, K.S. and Rajagopal, K.: "Computational and experimental investigation of low ODP and low GWP R123 and R290 refrigerant mixture alternate to R12" *Energy Conversion and Management*, Vol. 48, 2007, pp. 3053-3062.

Principal Author: Dr Bukola O. Bolaji is currently a Postdoctoral Research Fellow at Tshwane University of Technology, Pretoria in the Department of Mechanical Engineering. He is a Senior Lecturer in the University of Agriculture Abeokuta, Nigeria, Registered Engineer with Council for Regulation of Engineering in Nigeria (COREN), corporate member of the Nigerian Society of Engineers (NSE) and member of Environment Behaviour Association of Nigeria (EBAN).

Co-author: Zhongjie Huan is a Full Professor of Mechanical Engineering. He holds a PhD degree in Thermal Engineering from Tianjin University, China. He has trained a lot of students at both undergraduate and postgraduate levels, and has consulted in a variety of industries in the areas of refrigeration, air-conditioning and cold storage. At present he is the Head of Department of Mechanical Engineering at Tshwane University of Technology (TUT), Pretoria.

Presenter: The paper is presented by Dr Bukola O. Bolaji.