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Thermodynamic analysis of the performance of a vapour compression refrigeration system, working with R290 and R600a mixtures

B.O. Bolaji[a,*] **and Z. Huan**[b]

a. *Department of Mechanical Engineering, Federal University of Agriculture, PMB 2240, Abeokuta, Nigeria.*

b. *Department of Mechanical Engineering, Faculty of Engineering and Built Environment, Tshwane University of Technology, Pretoria, South Africa.*

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Abstract. Environmentally benign natural refrigerants have recently attracted considerable attention due to refrigerant contributions to ozone depletion and global warming. In the group of natural refrigerants, hydrocarbons are most closely related to the HFCs. In addition to their zero ODP and very low GWP, they are compatible with common materials found in refrigerating systems. Therefore, in this study, a performance simulation of R290 and R600a mixtures (80/20, 70/30, 60/40 and 50/50 proportion by mass, designated by RM1, RM2, RM3 and RM4, respectively) in a vapour compression refrigeration system is conducted by the thermodynamic calculation of performance parameters using REFPROP software. The results show that the mixtures exhibit higher refrigerating effects than R134a. The average pressure ratio obtained for RM1, RM2, RM3 and RM4 was 19.5, 16.5, 14.0 and 11.8% lower than that of R134a, respectively. All the mixtures, except RM1, exhibited a low discharge pressure, which is more desirable in refrigeration systems. The COPs of the mixtures were close to that of R134a with the advantage of higher values. Generally, the overall performance of the selected mixtures was better than that of R134a. The performance of RM4 was the best in terms of low compressor work and discharge pressure, and high COP, at varying evaporating temperatures.

1. Introduction

Ozone depletion and global warming are major environmental concerns with serious implications for future development of refrigeration-based industries. The effects on the industry of actions taken to reduce ozone depletion and global warming are now apparent. Chlorofluorocarbon (CFC) and hydro-chlorofluorocarbon (HCFC) refrigerants are a large class of chemicals that have contributed greatly to these environmental problems. These refrigerants have many suitable properties, for example, non-flammability, low toxicity and material compatibility, which has led to their common widespread use by both consumers and industries around the world, especially as refrigerants in air conditioning and refrigerating systems [1-3]. However, they also have high ozone depleting and global warming potential.

The discovery of these two major environmental problems has resulted in a series of international treaties demanding a gradual phase out of halogenated fluids. The CFCs have been phased out in developed countries since 1996, and since 2010 in developing countries [4]. Initial alternatives to CFCs included some hydro-chlorofluorocarbons (HCFCs), but they will also be phased out internationally by year 2020 and

*. *Corresponding author.*
E-mail addresses: bolajibo@funaab.edu.ng (B.O. Bolaji); HuanZ@tut.ac.za (Z. Huan)

2030 in developed and developing nations, respectively, because their Ozone Depletion Potential (ODP) and Global Warming Potential (GWP) are at relatively high levels, though less than those of CFCs [5,6].

At the advent of the Montreal Protocol and in the process of phasing out the Ozone Depletion Substance (ODS), R134a was identified as an alternate refrigerant to R12, keeping in mind its zero ODP and its possibility for use in old and new equipment without much change in working and operating conditions. However, R134a is a HFC refrigerant and HFCs are one of the six chemicals in the "basket" that are specifically signalled out by the Kyoto Protocol on global warming as a greenhouse gas [7]. Use of HFCs has allowed the rapid phase out of CFCs in developed and developing countries [8-10]. The high GWP of R134a gas and its impact on climate change were not considered seriously during the selection of R134a as an alternate refrigerant to R12. Now, R134a, a zero ODP refrigerant but with high GWP, is used worldwide. The sale of R134a has significantly increased during the past two decades. The increased emissions of this refrigerant to the atmosphere are steadily increasing the concentration of greenhouse gases, resulting in adverse climate problems [11,12].

Some studies have been carried out on the performance of R134a in vapour compression refrigeration systems. Linton et al. [13] compared the performance of R134a with that of R12 in residential heat pumps. Their results showed that approximately the same heating output was achieved with R134a, but the Coefficient Of Performance (COP) of the system was approximately 15% less with R134a than with R12.

Another series of tests conducted at the ARI (Air-conditioning and Refrigeration Institute), as reported by Bolaji et al. [14], showed that R134a exhibits a 6-11% increase in COP for moderate and warm heat pump rating conditions, while R134a has a nearly identical COP to that of R12 for a cold rating condition. Akintunde [15] also investigated the moisture solubility in R12 and R134a at various temperatures in order to evaluate the performance of R134a as a substitute for R12 in relation to moisture retention. The results obtained showed that the R134a absorbed more moisture than R12 at all temperatures. Therefore, R134a systems will be more prone to rusting and copper plating due to large moisture content in the refrigerant. For this reason, a more efficient and environmentally benign alternative refrigerant is needed in the refrigeration system.

Hydrocarbon fluids, such as R290 and R600a refrigerants, provide alternatives to a number of CFC, HCFC and HFC refrigerants. In addition to their zero ODP and very low GWP, they are compatible with common materials found in refrigerating systems and are soluble in conventional mineral oils. Since hydrocarbon refrigerants contain no chlorine or fluorine atoms, they cannot undergo reaction with water and, hence, do not form the corresponding strong acids that can lead to premature system failure. R290 and R600a refrigerants have been proposed and actually used in small refrigeration systems [16-18]. Their thermodynamic and transport properties are very similar to R134a currently used in refrigeration and air-conditioning systems.

The most important concern regarding the adoption of hydrocarbons as a refrigerant is their flammability. It should be remembered that millions of tons 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 being followed by the refrigeration industry. Various applications have been developed in handling 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 safety precautions [19,20]. Therefore, in this study, the performance simulation of R290 and R600a mixtures in a vapour compression refrigeration system is conducted by thermodynamic calculation and analysis of performance parameters. Also, the results obtained were compared to the baseline refrigerant (R134a).

2. Materials and methods

2.1. Cycle analysis

A vapour compression refrigeration system is the most widely used. It uses circulating liquid refrigerant as a medium, which absorbs and removes heat from the refrigerating chamber and subsequently rejects that heat at the condenser. The ideal vapour compression cycle is shown on the Pressure-Enthalpy diagram of Figure 1. This cycle is comprised of two isobaric

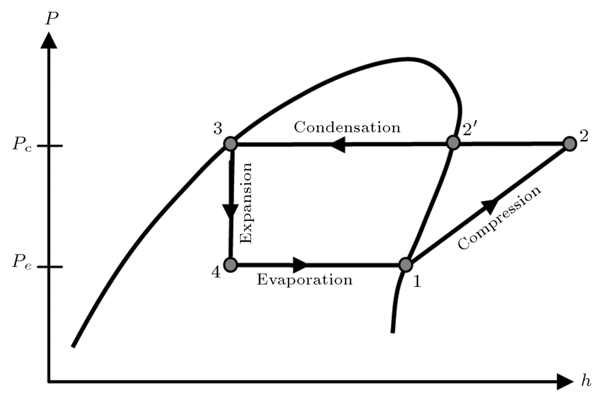

Figure 1. The ideal vapour compression refrigeration cycle.

heat exchange processes, a constant enthalpy expansion process and an isentropic compression process.

In the "ideal" vapour compression cycle, the refrigerant exit states from the condenser and the evaporator are on respective saturation lines. Beginning with state point 1, the saturated vapour refrigerant leaving the evaporator is compressed isentropically to the desired condenser saturation pressure by inputting mechanical work (process 1-2). The discharged vapour refrigerant is de-superheated and condensed by expelling heat through the wall of the condenser to another external fluid, which is usually air or water (process 2-3). The saturated liquid refrigerant leaved the condenser and enters the expansion device where some of the refrigerant is flashed at constant enthalpy (process 3-4), causing the remaining liquid portion to be at the desired temperature and pressure needed for a particular evaporator operating condition. The price paid for this process is that the entire refrigerant is not available for the vaporization process in the evaporator (process 4-1). The heat transferred to the refrigerant in the evaporator is called the refrigerating effect. For the purpose of rating the system's performance, for either the heating or cooling application, the efficiency term is the COP. It is, as all efficiency terms are, the desired output (e.g., the refrigerating effect) divided by the work input which, in this case, is the work input to the compressor.

Different refrigerants provide the desired product with more or less effectiveness. The desired product for a cooling application is the heat entering the evaporator. Considering the refrigeration system in the p-h diagram in Figure 1, the following assumptions are made:

(i) Evaporation is 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:

$$Q_{\text{evap}} = (h_1 - h_4), \tag{1}$$

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

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

$$W_{\text{comp}} = (h_2 - h_1), \tag{2}$$

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

(iii) A de-superheating is 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_{cond}, kJ/kg) is:

$$Q_{\text{cond}} = (h_2 - h_3), \tag{3}$$

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

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

$$h_3 = h_4. \tag{4}$$

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) for the refrigeration system and Eq. (3) to Eq. (2) for the heat pump system:

$$\text{COP}_{\text{ref}} = \frac{Q_{\text{evap}}}{W_{\text{comp}}}, \tag{5}$$

and:

$$\text{COP}_{\text{hp}} = \frac{Q_{\text{cond}}}{W_{\text{comp}}}. \tag{6}$$

2.2. Thermodynamic properties of refrigerants

The most fundamental issues of a working fluid's thermal properties 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, that is, an equation-of-state, utilizing specific heat. There exist a myriad of equations-of-state, which have been classified into families. Today, the most widely used refrigerant database is REFPROP [21,22]. 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 [11]. This software was used to compute the properties of the investigated refrigerants.

2.3. Thermodynamic analysis

Variation of the liquid saturated pressure with the temperature of the refrigerant (mixture) is an important criterion to consider when deciding on any changes in the compressor to accommodate a replacement refrigerant. The compressor size needs to be decreased for fluids of higher vapour pressure than

R134a, while it needs to be increased for fluids of lower vapour pressure in order for the system to provide the same cooling load. According to Kumar and Rajagopal [11], the expected capacities are proportional to the vapour pressure. Therefore, the saturated pressure and saturated temperature (boiling point) alone would be a good indicator in determining the relative size of the compressor displacement volume. Compressor performance and cycle efficiency vary considerably with the operating conditions (condensing and evaporating temperatures) of the system.

In refrigerant selection, according to Didion [21], capacity and efficiency are the two most important criteria to consider. From the viewpoint of economic operation and low energy consumption, it is desirable that a refrigerant has physical and thermal characteristics that will result in a minimum power requirement per unit of refrigerating capacity, that is, a high coefficient of performance (COP). In this study, the performance of the following hydrocarbon mixtures was investigated and compared with that of baseline refrigerants (R134a): RM1 (80% R290 + 20% R600a), RM2 (70% R290 + 30% R600a), RM3 (60% R290 + 40% R600a) and RM4 (50% R290 + 50% R600a). These mixtures are further referred to in this paper as RM1, RM2, RM3 and RM4, respectively.

3. Results and discussion

Figure 2 shows the variations of saturated vapour pressure and temperature for R134a, RM1, RM2, RM3 and RM4. As shown in this figure, the saturated vapour pressure curves for various proportions of R290 and R600a mixtures are very close to the vapour pressure curve of the R134a refrigerant; moreover, the mixture of RM2 is closer than the other mixtures. This indicates that these mixtures can exhibit similar properties and could be used as a substitute for R134a.

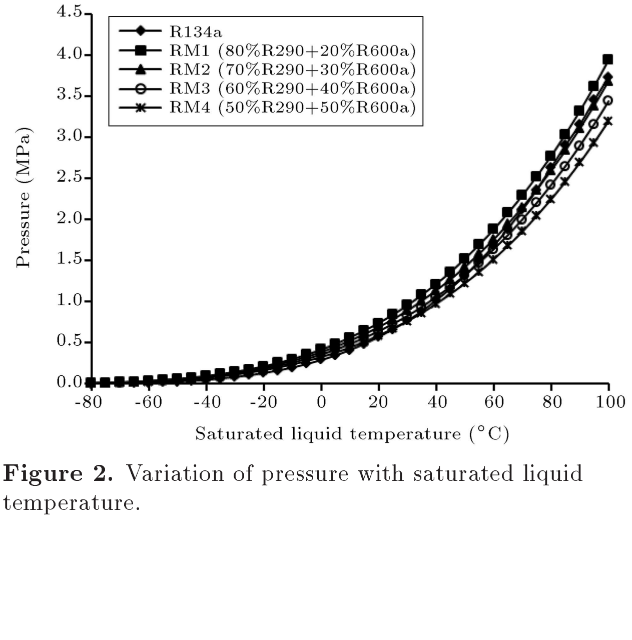

Figure 2. Variation of pressure with saturated liquid temperature.

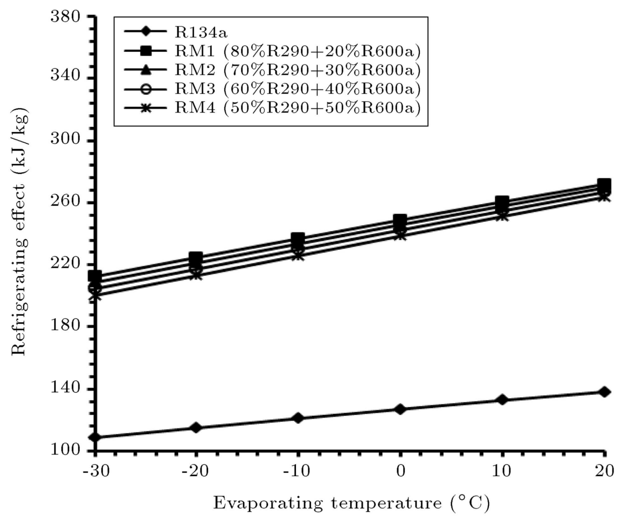

Figure 3. Variation of refrigerating effect with evaporating temperature at condensing temperature of 50°C.

However, Figure 2 also shows that the higher the percentage of R600a in the mixture, the higher the deviation of the pressure curve of the mixture from that of the baseline refrigerant.

Figure 3 shows the variation of the refrigerating effect with evaporating temperature at a condensing temperature of 50°C. As shown in the figure, the higher the percentage of R600a in the mixture, the lower the refrigerating effect of the mixture. The refrigerating effect increases with an increase in evaporating temperature, which is due to the increase in the latent heat value of the refrigerant. A very high latent heat value is desirable, since the mass flow rate per unit of capacity is less. When the latent value is high, the 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, more compact pieces of equipment. It is clearly shown in Figure 3 that the hydrocarbon mixtures exhibited higher refrigerating effects than R134a. Therefore, a very low mass of refrigerant would be required for the same capacity, and a smaller compressor size would also be required, due to their high latent heat values.

Figure 4 shows the variation of specific volume of vapour refrigerant mixtures of R290 and R600a with saturated liquid temperature. Specific volume increases as saturated temperature reduces. The higher specific volume of the hydrocarbon mixtures indicates the need for a higher compressor size for the same mass flow rate, but their higher latent heat values have compensated for this. However, the higher the percentage of R600a in the mixture, the higher the specific volume, which forms the basis of limiting the percentage of R600a considered.

The variation of compressor work input with

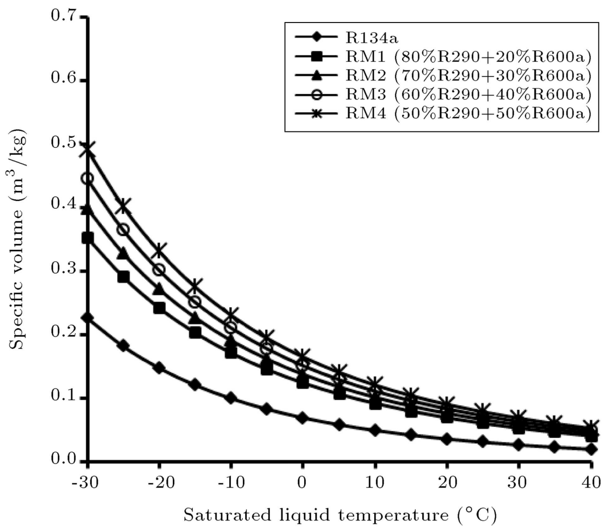

Figure 4. Variation of specific volume of refrigerant vapour with saturated liquid temperature.

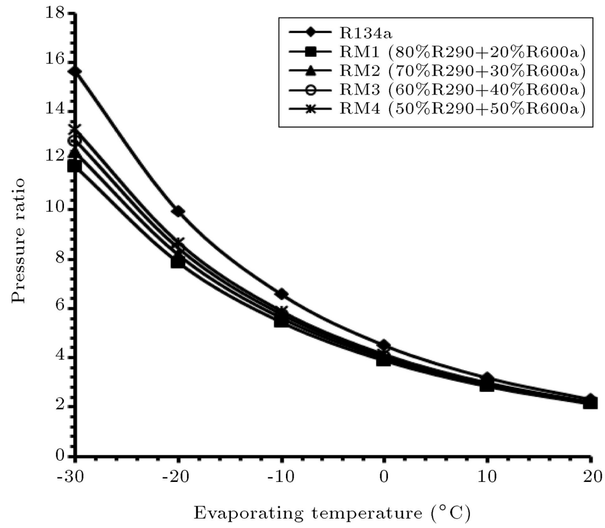

Figure 6. Variation of pressure ratio with evaporating temperature at condensing temperature of 50°C.

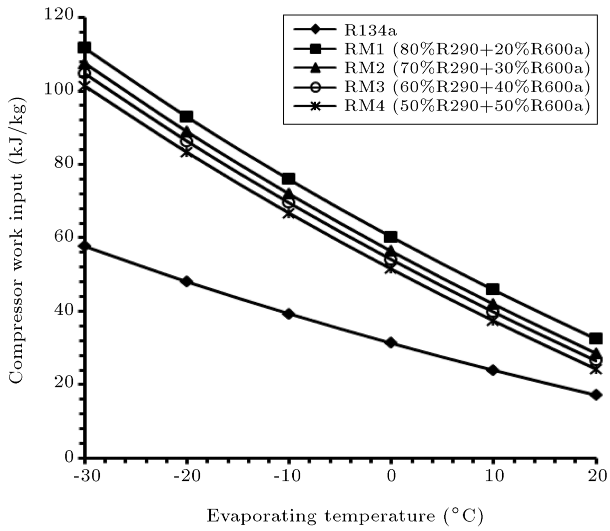

Figure 5. Variation of compressor work input with evaporating temperature at condensing temperature of 50°C.

evaporating temperature at a condensing temperature of 50°C is shown in Figure 5. The figure shows that the work of compression decreases as the temperature of the evaporator increases. This is due to the fact that when the temperature of the evaporator increases, the suction temperature also increases. At high suction temperatures, the vaporizing pressure is high and, therefore, the density of the suction vapour entering the compressor is high. Hence, the mass of refrigerant circulated through the compressor per unit time increases with the increase in suction temperature for a given piston displacement. The increase in the mass of refrigerant circulated decreases the work of compression. The works of compression using mixtures of R290 and R600a are higher than that of R134a, but they also exhibited much higher refrigerating effects than those of R134a (Figure 3). The lowest compressor work was obtained using RM4 (mass proportion of 50/50).

Figure 6 shows the variation of pressure ratio as a function of evaporating temperature for 50°C condensing temperature. The compressor pressure ratio reduces as evaporating temperature increases. The compressor pressure ratio is one of the criteria used for choosing a suitable alternative to any refrigerant in a conventional refrigeration system. High pressure ratio is detrimental to the performance of the system. Therefore, refrigerants with lower pressure ratio are more suitable and better than those with high pressure ratios. As shown in Figure 6, mixtures of R290 and R600a exhibit close behaviour to R134a with RM4; mass proportion of 50/50 being the closest. Other mixtures (RM1, RM2 and RM3) have the advantage of having a lower pressure ratio than both R134a and RM4. The lowest pressure ratio was obtained using RM1. The average pressure ratios obtained for RM1, RM2, RM3 and RM4 were 19.5, 16.5, 14.0 and 11.8% lower than that of R134a, respectively.

The evaporating temperature versus condenser duty for 50°C condensing temperature is shown in Figure 7. As clearly shown in the figure, the condenser duty slightly reduces as the evaporating temperature increases. The increase in the work of compression increases the heat added to the hot refrigerant during the compression process, which also increases the quantity of heat to be removed by the condenser. Various mass proportion mixtures of R290 and R600a considered in this study produced a better condenser duty than the R134a refrigerant (Figure 7). The highest condenser duty was obtained using RM1.

Figure 8 shows the discharge pressure at 50°C condensing temperature for various mass proportion

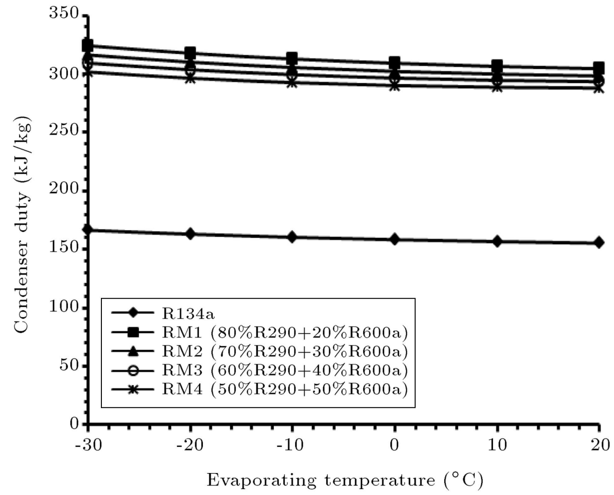

Figure 7. Variation of condenser duty with evaporating temperature at condensing temperature of 50°C.

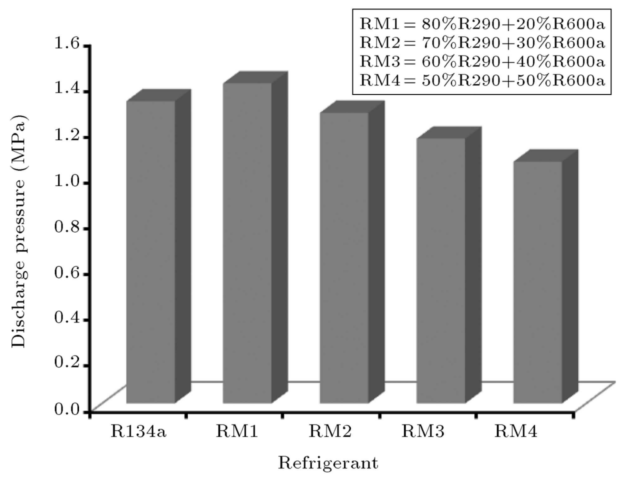

Figure 8. Discharge pressure at 50°C condensing temperature.

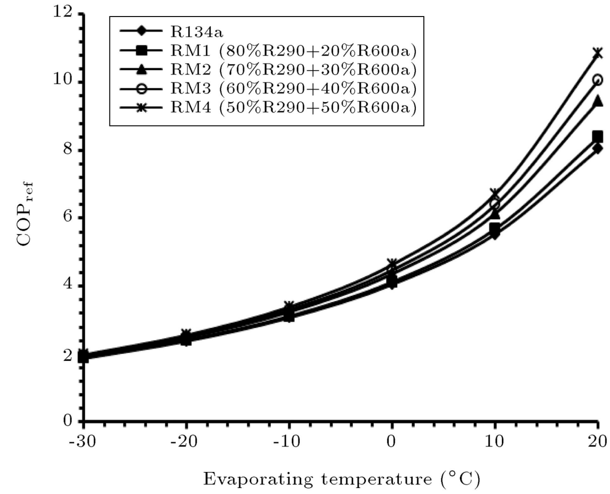

Figure 9. Variation f Coefficient Of Performance of refrigeration (COP_{ref} with evaporating temperature at condensing temperature of 50°C.

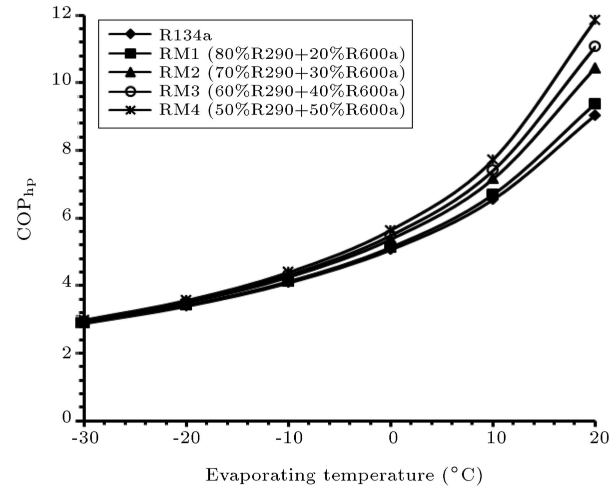

Figure 10. Variation of Coefficient Of Performance of heat pump (COP_{hp} with evaporating temperature at condensing temperature of 50°C.

mixtures of propane and iso-butane, and the baseline refrigerant (R134a). Except for RM1, all the hydrocarbon mixtures exhibited lower discharge pressure than R134a. 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 alternatives, and better than those with high discharge pressure. The lowest discharge pressure was obtained using RM4, which consists of a R290 and R600a mixture in a mass proportion of 50/50.

Comparisons of the COP_{ref} and COP_{hp} for R290 and R600a mixtures with R134a, as a function of various evaporating temperatures and a condensing temperature of 50°C, are shown in Figures 9 and 10. Similar trends were observed in the two figures, for all the refrigerants considered, both COP_{ref} and COP_{hp} increased with an increase in evaporating temperature. The COPs of the hydrocarbon mixtures are higher than those of R134a. Again, the highest COPs were obtained using RM4.

The performance parameters (refrigerating effect, compressor work input and COP_{ref}) were also analysed at various condensing temperatures and fixed evaporating temperature. Figure 11 shows the variation of refrigerating effects with condensing temperature at a fixed evaporating temperature of -20°C. As shown in the figure, similar to the result obtained at varying evaporating temperature, the higher the percentage of R600a in the mixture, the lower the refrigerating effect of the mixture. But, contrary to the increase in refrigerating effect with increase in evaporating temperature, the refrigerating effect reduces as condensing temperature increases. Again, Figure 11 clearly shows that the hydrocarbon mixtures exhibited higher refrig-

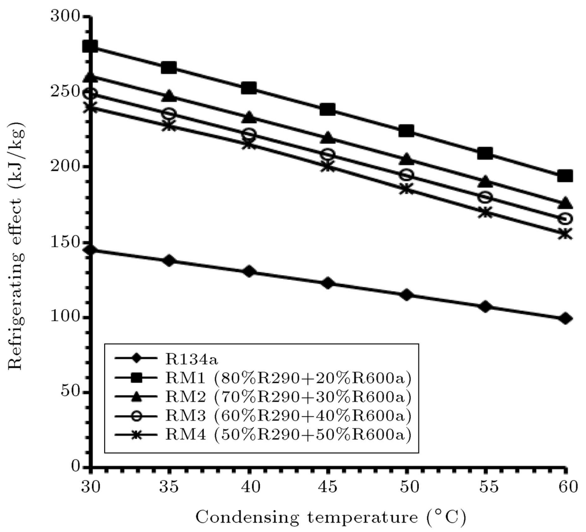

Figure 11. Variation of refrigerating effect with condensing temperature at evaporating temperature of -20°C.

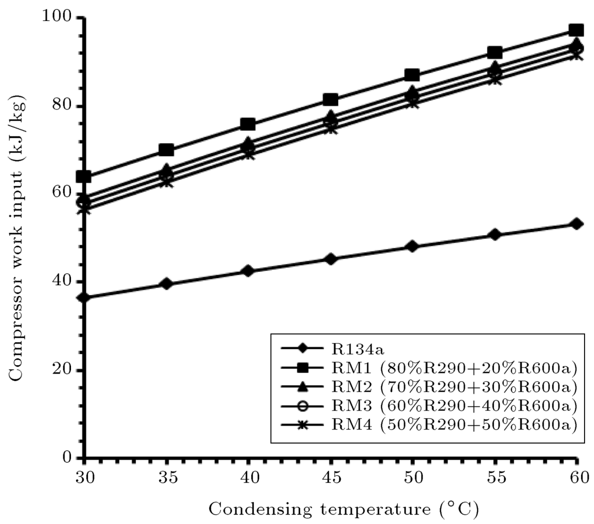

Figure 12. Variation of compressor work input with condensing temperature at evaporating temperature of -20°C.

erating effect than R134a. The highest refrigerating effect was obtained using RM1.

The variation of compressor work input with condensing temperature at a fixed evaporating temperature of -20°C is shown in Figure 12. The figure shows that the compressor work input increases as the condensing temperature increases, whereas, in Figure 5, the work of compression decreases as the evaporating temperature increases. The works of compression using mixtures of R290 and R600a are higher than that of R134a, but their refrigerating effects are almost double those of R134a (Figure 11). The lowest compressor work was obtained using RM4 (mass proportion of 50/50).

The Coefficient Of Performance (COP) of a refrigeration cycle reflects the cycle performance. Figure 13 shows the effect of condensing temperature on the COP at a fixed evaporating temperature of -20°C for R134a and the alternative refrigerants. As shown in the figure, COP reduces with an increase in condensing temperature, and an increase in the percentage of R600a in the mixture also reduces the COP of the system. The average COP obtained for RM1, RM2, RM3 and RM4 refrigerants was 8.6, 5.7, 2.5 and 0.3% higher than that of R134a, respectively.

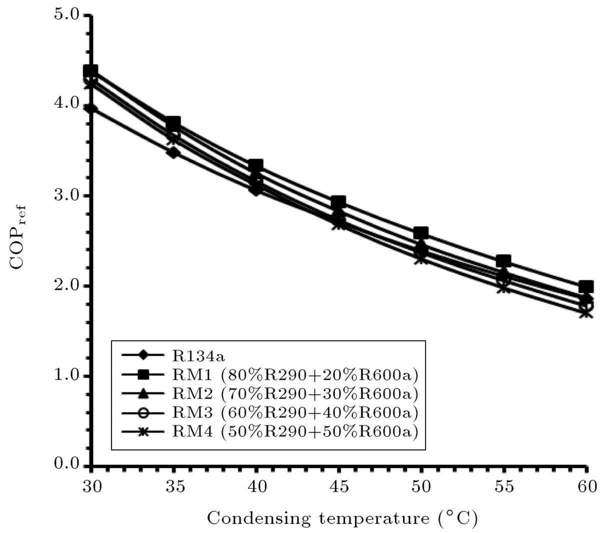

Figure 13. Variation of Coefficient Of Performance of refrigeration (COP_{ref} with condensing temperature at evaporating temperature of -20°C.

4. Conclusion

In this study, the performance simulation of R290 and R600a mixtures in a vapour compression refrigeration system is conducted by thermodynamic calculation and analysis of performance parameters, and the results obtained were compared with the baseline refrigerant (R134a). Propane (R290) and iso-butane (R600a) are the most prominent hydrocarbon refrigerants presently being used in refrigeration and air-conditioning systems. In order to obtain the performance and the best mass proportion mixture of these refrigerants to match the performance of R134a, mixtures of various proportions by mass of R290 and R600a (80/20, 70/30, 60/40 and 50/50) were selected and their thermodynamic properties were computed using REFPROP software. Performance parameters, such as refrigerating effect, compressor work input, pressure ratio and Coefficient Of Performance (COP), were computed and analysed at various evaporating and condensing temperatures.

The results obtained showed that the vapour pressures of R290 and R600a mixtures are very close to that of R134a, and mixtures of 70/30 (RM2) and 60/40 (RM3) were the best match in term of vapour pressure.

The various mixtures of R290 and R600a exhibited higher refrigerating effect than R134a. Therefore, very low mass of refrigerant charge will be required for the same capacity. They also exhibited a higher compressor work input than R134a, but their higher refrigerating effects have compensated for this.

The compressor pressure ratios of the mixtures are very close to that of R134a, and a mixture with mass proportion of 50/50 had the closest pressure ratio. All the mixtures, except that of 80/20 mass proportion, exhibited a low discharge pressure, which is more desirable in refrigeration systems. The lowest discharge pressure was obtained using the mixture with 50/50 mass proportion. Also, all the refrigerant mixtures produced a better condenser duty than R134a. For both the mixtures and R134a, the COP increased with an increase in evaporating temperature. The COPs of the mixtures were close to that of R134a, with the advantage of higher values. The highest COP was obtained using a mixture of 50/50 mass proportion. The performance parameters (refrigerating effect, compressor work input and COP_{ref}) were also analysed at various condensing temperatures and fixed evaporating temperature. The results showed that the system performed better at lower condensing temperatures. The higher the condensing temperature, the higher the compressor work input and the lower the refrigerating effect and the COP. Generally, the overall performance of the selected mixtures was better than that of R134a. However, as the percentage of R600a in the mixture increases, the saturation pressure deviates significantly from that of R134a. The specific volume and pressure ratio increase, which will affect their substitution in the existing system. The performance of the mixture with 50/50 mass proportion was better than all other refrigerants considered, in terms of low compressor work and discharge pressure and high COP, at varying evaporating temperatures.

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Biographies

Bukola O. Bolaji holds a PhD degree in Mechanical Engineering (Building Services Option), and is Associate Professor of Energy and Thermo-fluid. Currently, he is the Head of the Department of Mechatronics Engineering in the Federal University of Agriculture, Abeokuta, Nigeria.

He is a registered engineer and member of the Nigerian Society of Engineers, Environment Behaviour Association of Nigeria and member of the Southern African Association for Energy Efficiency.

He has supervised a number of students in the areas of solar thermal systems, thermo-fluids and environmental engineering at both undergraduate and postgraduate levels, and has consulted in a variety of industries in Nigeria on refrigeration, air-conditioning and solar power systems for over fifteen years.

Zhongjie Huan holds a PhD degree in Thermal Engineering from Tianjin University, China, and is Professor of Mechanical Engineering at Tshwane University of Technology (TUT), Pretoria, South Africa. He has supervised a number 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.