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Experimentally Investigating of Friendly Alternative Refrigerant for R134a in Automotive Air Conditioning System

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ABSTRACT

Automobile air conditioning systems (AACS) generally use R134a as a refrigerant; however, this has a negative impact on the environment because of its high global warming potential (GWP) value. National constraints have thus been applied to phase this out, which require refrigerants with zero ozone depletion potential ODP and low GWP to be used instead. In this work, blends of Liquid Petroleum Gas (LPG) and R134a were evaluated as alternatives to pure R134a in AACS, in order to decrease the flammability of the mixture and the effects of R134a. The properties of refrigerants were theoretically analysed in the authors' previous work to determine compatibility with the proposed AACS, and the mixture that thus emerged as an alternative refrigerant, named Rmix, consisted of R600A/R290/R134a with mass ratios 43/35/22. Rmix and R134a were thus evaluated experimentally in an experimental test rig, with results that revealed that the cooling capacity and coefficient of performance (COP) values for Rmix were l7.5% and 10% lower than those of R134a, respectively, and that the evaporator loads for these refrigerants were 4KW to 5KW at a compressor speed of 1,000 RPM. The reduction was increased at 2,300 RPM, with evaporator loads reaching 10% and 20%, respectively. The required charge size of hydrocarbons (HCs) was reduced, with a reduction ratio of 22. Overall, Rmix was found to be usable in existing AACS without modification.

KEYWORDS

ACC System, R134a, alternative refrigerants, low GWP, HC.

INTRODUCTION

Heat can cause an automobile's inside temperature to increase, making the internal environment uncomfortable for drivers and passengers: temperatures inside a car may reach up to 70 °C if it is left in the sun with closed windows [1]. Automobiles designer companies thus provide most cars with AAC systems based on Vapour Compression Refrigeration (VCR) systems, which traditionally utilise chlorofluorocarbon (CFC) as a refrigerant. Molina and Rowland (1974) first reported on the hypothesis of ozone depletion, the thinning of the layer of O_3 around the earth's atmosphere that plays an important role in filtering out harmful UV rays [2].They showed that the chlorine element catalytically demolished the ozone layer in the stratosphere, and thus the termination of the production of CFCs was begun in 1996 and completed in 2010 [3]. The usual alternative, HFC, has no direct impact on the ozone layer, but does have a high GWP. AAC systems thus produce both direct and indirect emissions that increase the greenhouse effect. Thus, on January 31, 2006, the European Commission legislated constraints on the installation of R134a systems in all new automotive types beginning on January 1, 2011, with the intent of covering all vehicles from January 1, 2017. Any alternative refrigerants must also be environmentally friendly, with GWPs less than 150 [4]. Thus, research began to find an alternative refrigerant to replace R134a.

Wongwises et al. (2006) [5] experimentally investigated hydrocarbon mixtures as alternative refrigerants in AACs, using propane (R290), butane (R600) and isobutane (R600a) mixed at four different mass ratios (R290/R600/R600a of 20/60/20, 50/40/10, 70/25/5, and 100/0/0). These were tested in an AAC test rig with a capacity of 3.5 kW driven by a diesel engine. The results showed that the mixture with mass ratio 50/40/10 was characterised by better performance in comparison with R134a at an engine speed of 1,500 rpm and an evaporator temperature of 4 to 6 °C. The charge size was 550 g, equivalent to 800 g of R134a. Mohanraj et al. (2009) [6] experimented with R290/R600a with a mass ratio 45.2/54.8 as a substitute for R134a in a 200-litre domestic refrigerator. The mass charge of R134a was 110 g and the capillary tube length was 4 m. The refrigerator charge sizes from the HC mixture were 40g, 50g, 60g, and 70g, with capillary tube lengths of 4, 4.5, 5, 5.5, and 6 m. The ambient temperatures were 24, 28, 32, 38, and 43 ˚C.

The mixture of HC with a mass charge of 60g showed the lowest values of energy consumption as compared with R134a. In addition, the COP of the HC mixture was higher by 3.25 to 3.6%, and the discharge temperature was lower by 8.5 to 13.4 K than with R134a, potentially improving the life of the compressor. However, modification of the system was required, including increasing the length of the capillary tube by 25%. Rasti et al. (2012) [7] substituted R134a with R436a, a mixture of R290/R600a with a mass ratio of 56/44. Experiments were then done on a domestic refrigerator with capacity 238 litres without modification. The compressor was charged with 32, 40, 55, and 105g of R436a, and the results showed that R436a performed better than R134a with respect to various parameters. By using R436a, the time to reach the set point (ON time) was reduced by 13%, and the energy consumption per day was reduced by 5.3%. The refrigerant charge of R134a required was 105 g, while the best amount of refrigerant for R436a was determined at 55 g, a saving of 48% in refrigerant charge. The evaporator inlet temperature was reduced by 3.5 ˚C, and the Total Equivalent Warming Impact (TEWI) of R436a was thus 11.8% lower than that of R134a, suggesting that R436a offers a good replacement for R134a.

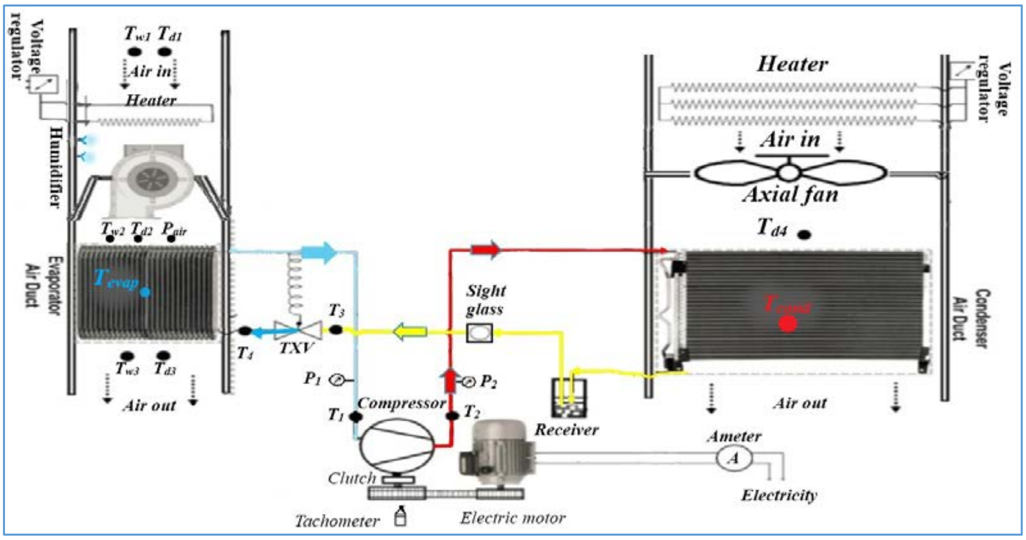
Esbría et al. (2013) [8] experimentally analysed a vapour compression system using R1234yf as a drop-in replacement for R134a. The energy performance of both refrigerants was compared, with R134a as a baseline. The results showed that the cooling capacity of the vapour compression system obtained with R1234yf was lower by about 9% in the studied range. In addition, the COP values of the system with R1234yf were lower by about 19%. The differences in energy performance were, however, reduced on modification of the system to use an internal heat exchanger (IHX). Dahlan et al. (2014) [9] experimentally tested a monitor AACS using R134a and AHCR as working fluids for the compressor. Four different speeds, 50, 70, 90, and 110 km/h, were used in the study. The tests were conducted using a roller dynamometer and a hatchback vehicle with a 1.3-litre petrol engine to stimulate an actual vehicle on a level road. The AAC unit was operated at a temperature set point of 21 °C and the blower speed was held at position 2. The ambient temperature was 30 °C and the humidity of surrounding conditions was 50 to 65%, with an internal heat load of 1,000 W. The results showed that temperature distribution and fuel conservation showed positive improvements and AHCR cooling capacity was higher at every vehicle speed at increments ranging from 0.86% to 4.14%, with a COP higher than HFC-R134a by up to 10.16%. It was thus deemed possible to use AHCR as an alternative refrigerant without the need to amend the current systems.

Kandhaswamy et al. (2017) [10] experimentally evaluated the performance of AAC systems by using HC mixtures as alternative refrigerants to R134a. A mixture of R290/R600a in a mass ratio 50/50 was retrofitted to the system and the performance was evaluated using different compressor speeds and cabin loads. The inlet air temperature to the condenser varied in the range 30 to 50 °C. The results showed that the alternative HC mixture had a faster cooling rate due to its high latent heat of vaporisation: the compressor power consumption of the HC mixture was observed to be higher by about 26% than that of R134a, and the higher refrigeration effect of the HC improved the COP of the system by about 5% over using R134a. Zhang et al. (2020) [11] reveal that there were three factors affected R290 distribution such as leak hole, wind speed and leak pressure. Also, the maximum concentration and duration time of R290 increased by increasing the hole and decreasing the wind speed and the mostly risk situation in the system was the leakage in the evaporator. Mohanraj and Abraham (2020) [12] reviewed the suitable refrigerant that to be used in the air conditioners of the automobile. They presented the chemical characteristics and thermodynamic properties of the refrigerants. The study discoved that the hydrocarbon refrigerants will use in the automobile air conditioning sector in wide range becaused of they have good thermodynamic properties.

Zhang et al. (2020) [13] tested and compared HFC-152a in SL system and the SL system presented heating in the performance in contrast with DX system. The heat was used in the car cabin by using heat exchanger and the coefficient of performance was also increased. Muslim et al. (2020) [14] Conducted an experimental work to study the effect of cooling condenser air to the performance of VCRC. The results proved that the COP was decreased due to increase in environment temperature and reducing the air temperature was possible. In the authors' previous work (2020) [15], a theoretical analysis was made of the cooling performance of an AAC system using R134a. The results were compared with blends formed by mixing hydrocarbons R290 and R600a, which are flammable, environmentally-friendly and ultra-low GWP refrigerants, with R134a (R600a/R290/R134a). The mixing of HCs with R134a helped to reduce the flammability of the blend and the bank of R134a. Refprop software was used for this thermodynamic analysis, and two mixtures emerged as alternatives to R134a, known as Mix1 and Mix2, consisting of R600A/R290/R134a with mass ratios 33/45/22 and 43/35/22, respectively. The theoretical results revealed that Mix2 provided the same level of refrigerant performance as R134a with lower GWP, and that it could be used directly without any modification to the system. In this work, therefore, the performance of a test rig representing an AACS was evaluated experimentally with R134a and Mix2, renamed Rmix, to examine the latter's applicability as an alternative refrigerant to R134a.

THE EXPERIMENTAL TEST RIG

The test rig was implemented to evaluate the cooling performance of R134a and the alternative refrigerant Rmix.



T_1, T_2: Refrigerant temperature inlet and outlet compressor.
T_3, T_4: Refrigerant temperature inlet and outlet TXV.
T_{w1}, T_{d1}: Wet and dry internal fresh air to evaporator duct.
T_{w2}, T_{d2}: Wet and dry internal fresh air to evaporator.
T_{w3}, T_{d3}: Wet and dry outlet fresh air from evaporator duct.
T_{d4}: Dry internal fresh air to condenser.
P_1, P_2: Refrigerant pressure inlet and outlet compressor.
P_{air}: Fresh air pressure before evaporator.

Figure 1. Schematic Diagram of Test Rig

The test rig was built locally by using typical parts of an automotive air-conditioning system that used R134a as a designed refrigerant. These parts are shown schematically in figure (1). The test rig includes compressor type Denso model 10PA15, aluminum finned heat exchanger representing the evaporator size (30*30*10) cm, aluminum finned heat exchanger representing the condenser with built-in receiver dryer size (40*65*2) cm, thermal expansion valve (TXV) block type made from aluminum, electrical blower fan was installed before the evaporator, and electrical axial flow fan was used for rejected the heat form condenser worked on 12 volts. Addition parts were added as shown in figure (1) to the test rig for assimilate the work condition of the AACS. Electrical motor (single phase, 3kW, 3000 RPM) was installed on iron chassis parallel with compressor instead of car engine, which was connected with compressor by rubber belt through their pulleys. Insulation duct was put before evaporator fan size (0.27 x 0.27 x 2) m used to modifying and controlling the moisture air properties before interning the evaporator. Insulation duct was put before condenser size (0.5 x 0.5 x 0.5) m with two elbows size (0.5m x 0.5m x 90°) used to modifying and controlling the moisture air properties before interning the condenser. Two 3.5kW electrical heaters five stages were installed inside the evaporator and condenser ducts used to control the temperature of internal fresh moisture air. Humidifying system was installed after electrical heater inside evaporator duct included a submersible high-pressure pump (8 bar x 3L/min), four micro nozzles with connection pipes and water tank with electrical heater 1000W to produce water steam. Two variable resistances 500W were used to change the speed of fans for evaporator and condenser.

Electrical board included multi switches and gauges for voltage, ampere, pressure, and temperature as a control board. The required measuring tools were fixed after completed the installing of all parts for test rig. These tools were used to measure the properties of refrigerants and fresh air, which required to calculate the performance of refrigerants and made comparison between them. Two digital pressure gauges diaphragm type were used to measure the low pressure before the compressor and high pressure after it during tests. Thermocouple model tm500 type K with Data-logger type EXTECH used to measure the temperatures in different positions. Digital ampere meter was used to measure the current. Digital voltage meter used to measure the voltage and digital hot wire was used to measure the speed of fresh moisture air. The charge size of refrigerants was measured by portable refrigerant electronic scale. Digital electronic tachometer was used to measure the revolution speed of compressor.

Preparing the Test Rig

Figure (2) shows the experimental setup of the test rig including the assembly of all the parts. The designed amount of oil must be added before installing the compressor, Polyalkylene glycol (PAG), which was used as the lubricating oil for HFC-R134a. The proposed amount of PAG was 150 ml [5]. HCs are soluble with mineral oils and synthetic oils [16]. Therefore, there was no need for changing the oil type during all tests. After fixed all connections, a preparing procedure has been done before staring the experimental work including the following steps: -

1- The leakage of system was tested by compressed clean fresh air inside the system with pressure up to 20 bars. External compressor was used in order to conduct the process and monitoring the pressure gauge for 10 minutes. If a decreasing in the gauge's data happened, this means leaks is available and must be fixed.
2- Evacuation the system by using vacuum pump with pressure reach (-2) bar to ensure extracting all moisture air or previous refrigerant from system.
3- Charging the required refrigerant size by using an electronic scale.
4- Repeating the steps 2-3 when the refrigerant is changing.

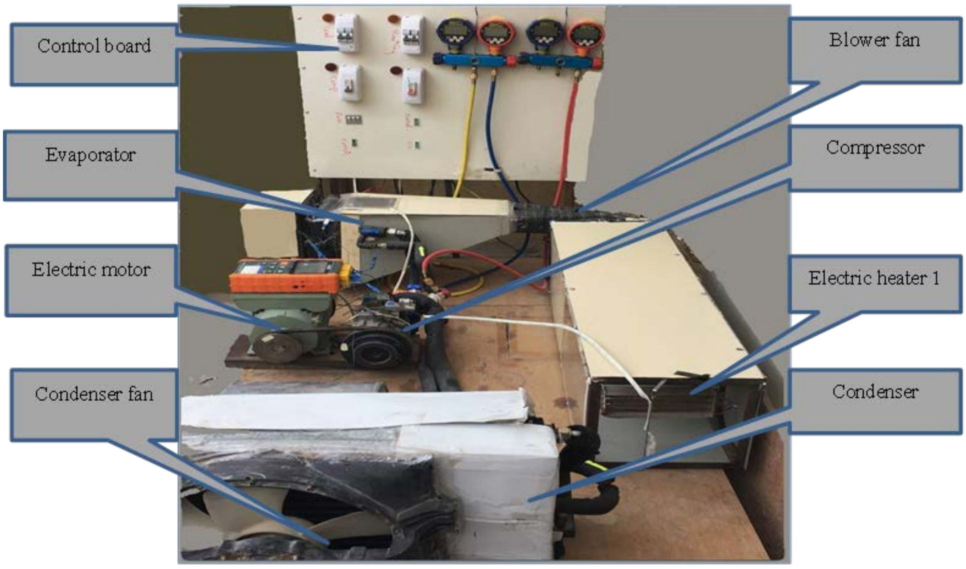


Figure 1. The test Rig

The Charge Size

The charge size was 530 g of R134a according to the serves manual for Toyota Camry 2005. It was charged gradually through the low-pressure port by using electronic scale and pressure charging was performed after a vacuuming process to the system. The charging operation was stopped, when the pressure inside the system was equal to the pressure of refrigerant cylinder. The test rig was turned ON before reaching the desired value. The pressure on the low side decreases gradually, and the charging operation starts again. During that, the sight glass which is installed on the output line of condenser was monitored to know the phase of refrigerant that exit from the condenser. In the first, bubbles were seen through the sight glass, and then the procedure was continued until all bubbles disappeared and the charge size reached 500g. The different in charge size returned to the modifying pipes by reducing their length according to the dimensions of the test rig. The charge sizes for HCs mixture was nearly 0.5 of R134a size [17]. According to the pressure of the components of the mixture, the lowest one must be charged first. The sequence then R600a, R134a and R290, but for safety, R600a and R290 are charged firstly, then R134a to insure charge HCs before turning the test rig. The total charge size of the mixture was 250 g and the charge size of components was according to the mass fraction of the mixture.

The Test Procedure

The condenser fan speed was adjusted to hold the condenser temperature on nearly 55°C. The scenario of the test was evaluating R134a and its alternative in the same working conditions at deferent evaporator loads, and for two compressor revolution speeds. The load of evaporator was increased at fixed compressor revolution speed; through increasing the evaporator fan speed gradually every five minutes until reach the maximum fan speed. The test rig was operated at lowest evaporator fan speed at least 20 mints until reached steady state at suggested conditions with all measuring devices were turned ON. This scenario was repeated for all refrigerants. Three pairs of thermocouples were installed at the beginning of evaporator duct, before and after evaporator coil to detect the dry and wet temperatures. The flow rate of air was detected by using anemometer. The cooling capacity could be calculated from the deferent in moisture air properties before and after evaporator coil [18]. The power which consumed by compressor was measured by using voltmeter and ammeter gauges. The COP was calculated for all refrigerants [15]. The accuracy, range, and resolution of measurement instrumentations that used in this study are shown in table (1).

Table 1: Resolution, range and accuracy of measurement instrumentations

Instrumentation	Measurement	Resolution	Range	Accuracy
Temperature recorder EXTECH tm500SD	temperature	0.1°C	-100 to1300°C	$\underline{+}$ 0.4%°C
Hot wire anemometer model HT-9829	air velocity	0.1m/s	0.1~25.0m/s	±5%+0.1m/s
Digital Pressure Gauge type DSZH	Pressure	1KPa	0KPa~6mpa	±0.5%
Digital ampere meter TAIFA DP3 – 72A	Current	0.2A	1-5000/5A CT	1% ±1digit
Digital voltage meter Type NK105-22D/C	Voltage	1	10-600V	1% ±1digit
Portable refrigerant scale RCS-N9030	Mass	5g	5 - 100Kg	0.05%+/1digit
Digital electronic tachometer (DT2234A)	Revolution speed	2.5	2.5 - 99999 RPM	±0.05%+1digit

Experimental Results

The experimental evaluations for refrigerants were done for deferent five loads at average condenser temperature 55°C and three compressor speeds. The results were illustrated in figures (3) to (14) for two refrigerants, five loads, and two revolution compressor speed 1000 and 2300 RPM and represented by *Sp1* and *Sp2* respectively. The suction pressure and temperature values of compressor for R134a and Rmix were increased with increasing the evaporator load. The refrigerant evaporates was interning the evaporator through absorbing the heat from evaporator shell, and this happens at constant temperature depending on its latent heat. The heat exchanging between evaporator shells and outside was increased through increasing the fan speed due to increase the load and the temperature of refrigerant was increase, since its latent heat, not enough to absorb all heat rate. As a result, the pressure will increase too. The expansion valve, which has thermal needle valve, was sensed the output evaporator temperature and increased the mass flow rate of refrigerant. The latent heat was increased. The zeotropic behavior for Rmix made the output temperature of evaporator more than R134a at same working conditions. The expansion valve was increased the mass flow rate and the pressure of evaporator later. Therefore, the Rmix had temperature and pressure in low pressure side more than R134a at same working conditions. The suction pressure and temperature values of compressor for Rmix at *Sp1* were nearly more 25% and 8% respectively as appeared in figures (3) and (4) respectively.

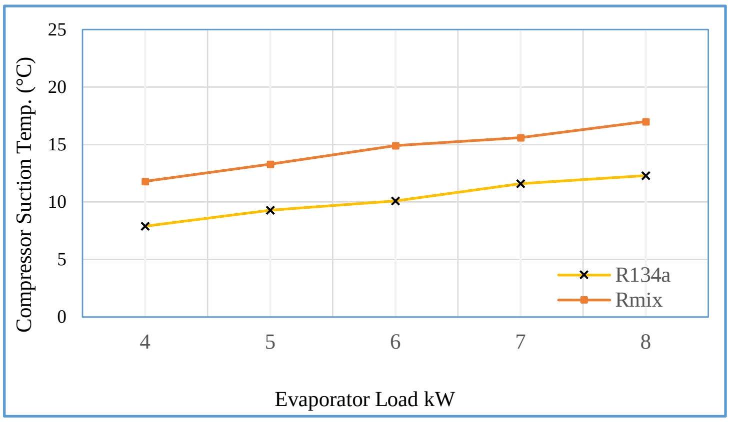


Figure 3. Compressor suction temperature with evaporator loads at compressor speed 1000 RPM

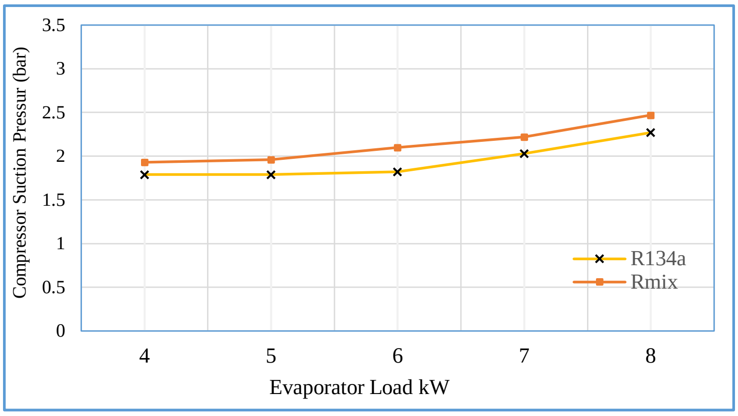


Figure 4. Compressor suction pressure with evaporator loads at compressor speed 1000 RPM

The increasing of rotational speed for compressor made the compressor pumped the formed vapour faster than the liquid. The pressure and temperature in the evaporator fall more than Rmix due to R134a had liquid density more than HCs. This made the deference in temperatures and pressures nearly duplicate at *Sp2* as shown in figures (5) and (6).

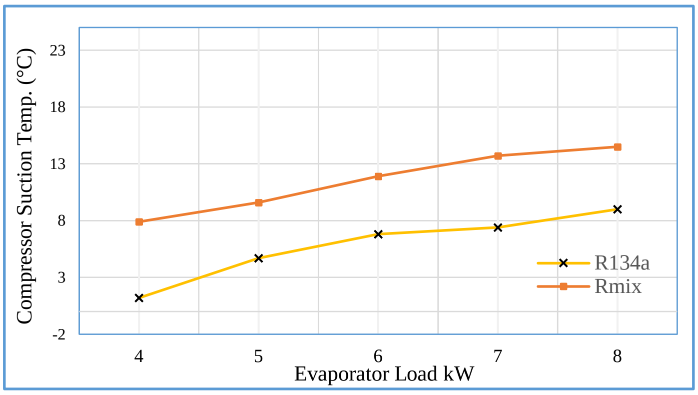


Figure 5. Compressor suction temperature with evaporator loads at compressor speed 2300 RPM

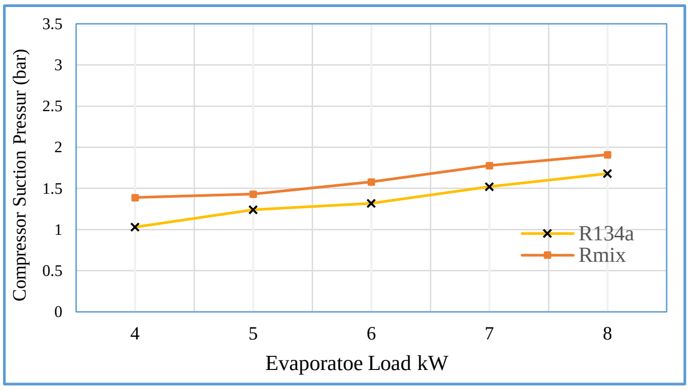


Figure 6. Compressor suction pressure with evaporator loads at compressor speed 2300 RPM

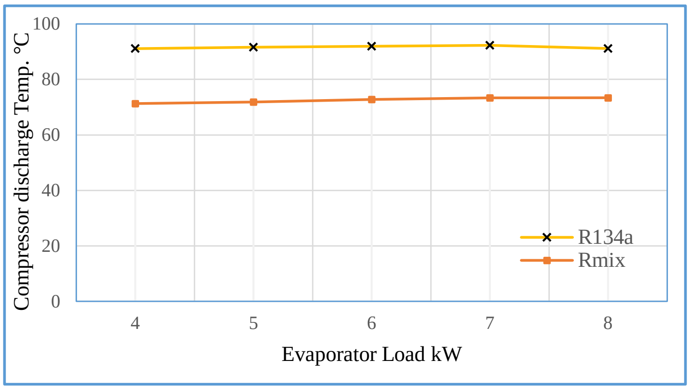


Figure 7. Compressor Discharge Temp. with Evaporator Load at Compressor Speed 1000 RPM

The discharge temperature values of compressor for Rmix were nearly less 20% as illustrated in figure (7) at *Sp1*. The high thermal conductivity for HCs was increased the heat loss through compressor's shell.The heat loss through compressor shell also effected on the compressor's discharge pressure and caused the pressure of Rmix less in average 5% than R134a at Sp1 as shown in figure (8). Nearly the same deferent for discharge temperature and pressure values for compressor at Sp2.

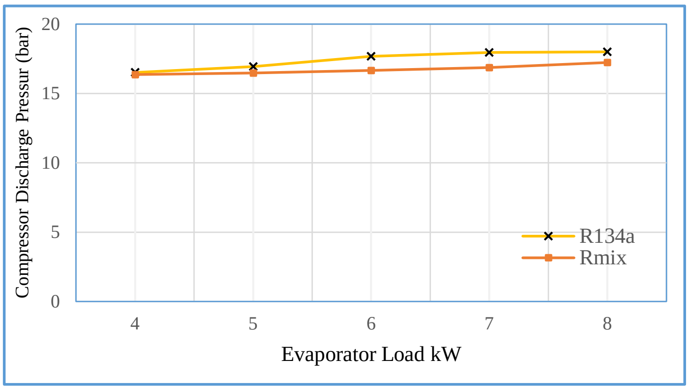


Figure 8. Compressor discharge pressure with evaporator load at compressor speed at 1000 RPM

The cooling capacity of the test rig was increased with increasing evaporator load at the same compressor speed Sp1 and Sp2 due to increasing the refrigerant mass flow rate through expansion valve as illustrated in figure (9). The HCs have refrigerant effect more than R134a and lower values of density. The lower isentropic compressor efficiency was due to high thermal conductivity. As results, the mass flow rate for R134a was more than Rmix at the same displacement and speed for compressor. Also, the cooling capacity of Rmix with evaporator load values form 4kW to 5kW at Sp1 was reduced by 7.5%. The reduction was increased to reach 11% at evaporator load 6 kW as illustrated in figure (9). The increasing of compressor speed led to increase the pressure and temperature where together helped to increase the leakage ratio through the clearance between piston and their cylinder. The HCs have vapor viscosity values less nearly 50% than R134a [15] which caused to increase the reduction in mass flow rate of Rmix. As result, the difference between the cooling capacity of Rmix and R134a was increased as shown in figure (10).

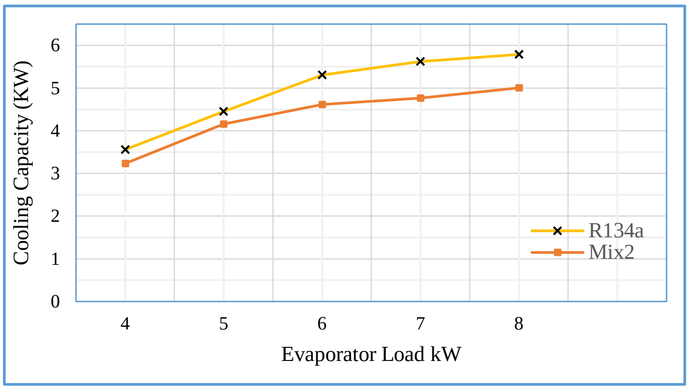


Figure 9. Cooling Capacity with Evaporator Load at Compressor Speed 1000 RPM

The power consumption for Rmix and R134a was appeared slightly the same values at SP1 with increasing ratio nearly 14% as shown in figure (11). However, the ratio of power consumption of SP2 for compressor was 25% due to increasing the amount of leakage through clearance and accumulated behind the piston, which add more power for sweeping it at suction stroke.

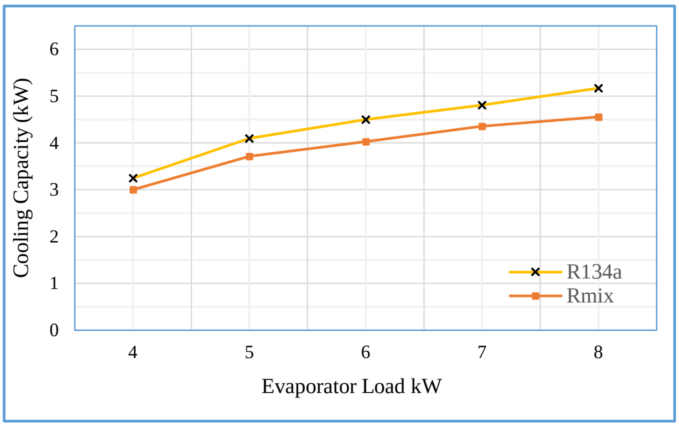


Figure 10. Cooling capacity with evaporator load at compressor speed 2300 RPM

The power consumption of compressor for Rmix at *SP2* had values nearly more 14% than R134a as shown in figure (12). The low viscosity of vapour for Rmix and the zeotropic behavior increased the mass flow rate of Rmix helped to increasing the power consumption of compressor. The COP values of test rig appeared increasing for two refrigerants with increasing the evaporator load at SP1 as shown in figure (13) due to the increasing of cooling capacity ratio more than the increasing consumption power of compressor ratio. The increasing of power consumption for compressor at SP2 continued until the evaporator load reached 6kW. After that, the COP values started decreasing until reach the maximum evaporator load due to increasing the losing.

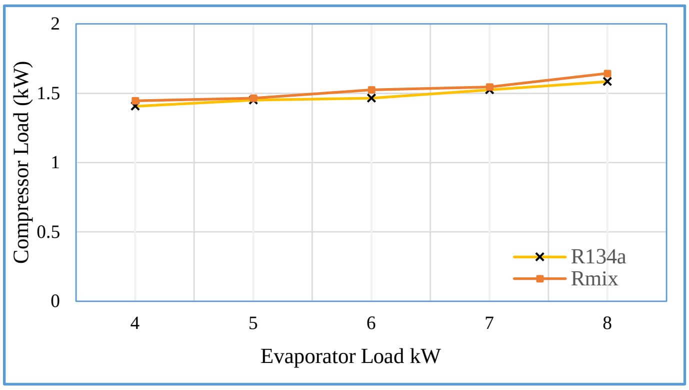


Figure 11. Compressor power consumption with evaporator load at compressor speed 1000 RPM

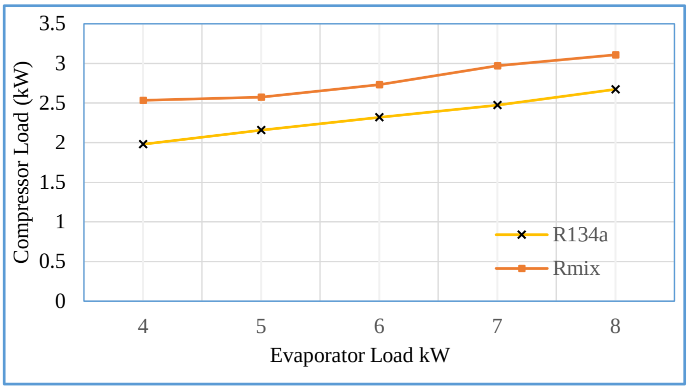


Figure 12. Compressor power consumption with evaporator load at compressor speed 2300 RPM

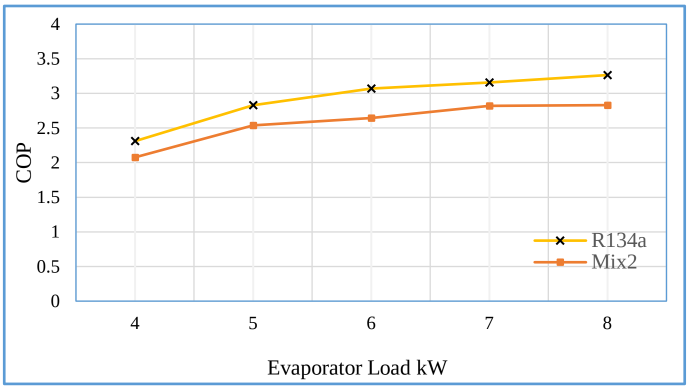


Figure 13. Coefficient of performance with evaporator load at compressor speed 1000 RPM

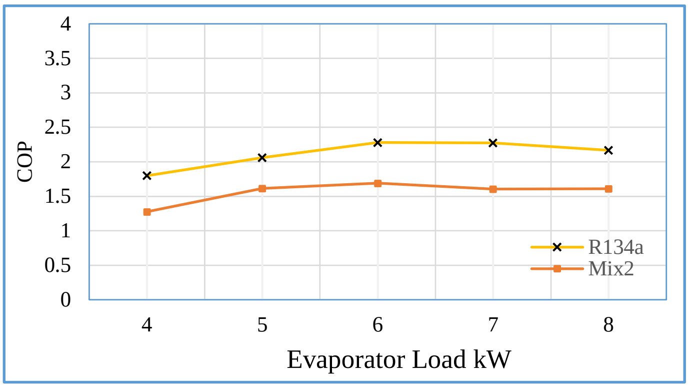


Figure 14. Coefficient of performance with evaporator load at compressor speed 2300 RPM

The reduction of cooling capacity and the high power consumption for Rmix were effected on their performance and caused the COP values less 10% and 20% than R134a at SP1 and SP2 respectively as showed in figure (13) for *SP1* and (14) for *SP2*.

CONCLUSIONS

The performance of automobile air conditioning system using R134a and Rmix was experimentally compared. The comparison was carried out at deferent evaporator loads. The following conclusions were drawn.

1. The opportunities to use HCs as an alternative refrigerant in available AAC systems that used R134a were through reducing the required charge size by redesigns it or mixing them with nonflammable refrigerant.
2. The mixing of R290 or R600a with R134a as alternative refrigerant caused the mass fraction of R134a more than R290 or R600a due to the high pressure for R290 and the low density of R600a and the GWP of mixtures exceed the allowable range.
3. Rmix (the mixing of R600a/R290/R134a) with mass fraction (43/35/22) reduced the required mass charge of HCs by 22%.
4. The GWP value for Rmix was 147.42 which less 89% than GWP of R134a.
5. The results of Rmix were acceptable at the temperature and pressure values for the condenser and evaporator at deferent loads and rotational compressor speeds.
6. The cooling capacity values of R134a were more 7.5% than Rmix at low compressor speed within the range from 3 to 4.5kW and for high speed in the same range were more 10% than Rmix.
8. The COP values of Rmix at a low and high speed less 10% and 20% than R134a respectively.

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