EXPERIMENTAL ANALYSIS OF MOBILE AIR CONDITIONING SYSTEM USING R513A AS ALTERNATIVE REFRIERGANTS TO R134A

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Abstract

Therefrigeration sector has great impact on world climate changes. The phase-out of hydrofluorocarbons (HFCs) often needs to make choices between high GWP alternatives and more planet-friendly alternatives. In this regard, performance of refrigeration systems in countries characterized by long, hot, summers face problems in identifying suitable alternatives due to the impact of such temperatures. This study stands for technical paper to conduct an experimental comparison for the use of R513A as a substitute for R134a. Energy analysis was performed to evaluate performance of both refrigerants. The research was implemented by a device that simulate an automobile (A/C) system unit capacity 3kW. effect of several variables were studied: ambient temperature was varied from 30°C to 50°C step by 4°C, and internal load was (700, 1000 and 1300) W, speed of compressor was 1450 r.p.m and 2900 r.p.m. The results showed that average value of COP of R513A was convergent with slightly higher than R134a by about 1%-2%, and average cooling capacity, mass flow rate were higher by about 1.6%-3%, 15%-17% respectively. The compression ratio and temperature of discharge were less than R134a by about 4.6% - 6% and 7.5%-8% respectively. So R513A could be considered as a good alternative for R134a without any modification on A/C system.

Keywords: COP, Cooling capacity, automotive air conditioning, alternative refrigerant, R513A.

I. Introduction

The cooling system in the vehicles is designed to keep cold, comfortable, when the temperature outside is high. However, in Iraq and due to the hot climatic conditions and long summer season it operating for a long time. The Temperature inside cabin may be reach 70°C (Aljubury et al., 2015). Most automobile air conditioning systems operating by R134a. The necessity is growing for finding alternative refrigerants in order to replace (HFC) R134a because it has a great
influence on energy consumption, environmental pollution and human health is a result of its concentration increase in the atmosphere (Carpenter et al., and Rigby et al., 2014). Attempts are being complete all around the world to find the substitutes for high GWP refrigerants so as to save green and healthier future. Different mixtures of HFOs and R134a which are compatible with most of the materials typically used with the HFCs, have been proposed to overcome the energetic and flammability limitations the most promising HFO/HFC mixtures alternatives to R134a (Mota-Bablioni et al., 2015). R513A is a blend consist of (R134a/R1234yf 44% /56% in mass percentage) with an ODP of zero and a GWP of 631 suitable to replace R134a.

Table 1. Shows the difference properties between R134a and R513A. (Mota-Babalion et al., 2017) compared experimentally between the refrigerants R513A and R134a. The Experiments were conducted for different evaporation and condensation temperatures. The discharge temperature of R513A was 3% less than that of R134a and the pressure ratio was 2%-6% lower than R134a with slight increase of COP about 5%. (Mota-Babaloni et al., 2018) studied the feasibility of using fluid R513A as an alternative to refrigerant R134a according to exergy analysis. The experiment was carried out on a compression refrigeration cycle system under the influence of evaporation temperature which was ranged from -15°C to 5°C and condensation temperature 30°C and 35°C. The refrigerant behavior was acceptable according to the second law of thermodynamic as compared with R134a with little increasing of exergy efficiency about 0.4% (Mota-Babaloniet al., 2019) suggested the use of a heat exchanger to improve energy performance for R513A, the result showed increased of COP by 4% for R134a and 8% for R513A. The temperature of the discharge was greater than 26°C for both of them, cooling capacities enhancement by 5.6% & 3% for R513A & R134a respectively. They recommended that at high pressured internal heat exchanger (IHX) be appended to the system. (Yang M et al., 2019) conducted an experimental study of three different amounts of refrigerant R513A (70, 80 &90) gr. Under alike terms that of the ambient temperature were conveyed in a domestic fridge that was laden with the refrigerant R513A. The ideal amount of R513A was less than 5.9% than R134a which originally charged, the time of stop was 21% greater than R134a and Power consumption less than R134a about by 3% which indicated an improved performance concerning the cooling capacity. (Makhnatch et al., 2019) conducted an experimental studied two refrigerants, R450a and R513A that have a low GWP as a substitute to R134a in high temperature regions. The experiments were conducted under condensing temperatures of (40, 50 &60) °C while the evaporative temperature ranging varied from (-12.5) to (17.5) °C with a step increase of 2.5°C. The study showed that R513A and R450a have COP on average about (1.8% & 5.3%) less than R134a and mass flow rate R513A has 18.9% higher than R134a and R450a about 14.3% less than R134a. In addition the cooling capacity of R513A was higher by about 2.5%. Aljubury and Mohammed, 2019 used micro channel condenser
instead of the conventional type in automotive (A/C) system rig. The ambient temperatures range from 40 to 65°C, COP was increased by 20%.

The goals of this paper are technically to study performance analysis of R513A as an alternative to R134a in automotive air conditioning system, without any modification under the influence of several variables and made comparison between them.

Table 1. The properties of R134a and R513A

|                        | R134a                         | R513A                         |
|------------------------|-------------------------------|-------------------------------|
| Composition            | pure                          | 44%R134a/56%R1234yf          |
| Safety classification  | A1                            | A1                            |
| 100-year GWP           | 1430                          | 631                           |
| Normal boiling point (C)| -26.07                        | -28.3                         |
| Critical temperature (C)| 101.06                        | 97.7                          |
| Critical pressure (MPa)| 4.06                          | 3.7                           |
| Critical density (kg m$^{-3}$)| 511                          | 490.9                         |
| Latent heat (kj.kg$^{-1}$)| 198                           | 171                           |
| Liquid density (kg m$^{-3}$)| 1294                         | 1226                          |
| Vapor density (kg m$^{-3}$)| 14.4                          | 16.1                          |
| Glide during 0.1-2 MPa (C)| 0                            | 0.015                         |

II. Experimental test RIG and Instruments

Fig.1 Shows a schematic diagram of test rig. Which simulated automotive (A/C) as seen in Photographic picture shown Fig.2 nominal capacity of 3kW. The rig consisted of a reciprocating compressor driven by a two speed (1450 & 2900) r.p. electric motor together with a compartment by the dimensions (300x200x150) mm, the internal load that represented the passengers is simulated by a 1500W heater 1500W that can be changed by a regulator knob, there is also a thermostat so as to control the temperature inside the cabin. The rig was originally charged with R134a, in addition to this it also possess high and low pressure. The whole rig was surrounded by room with 2×1.5×2 m dimensions in order to simulate the conditions of environment called test room shows in Fig.3. The test room is so large insulated that the enclose heat inside. Heater resistance capacity 6 kW was used to simulate the environment temperature controlled by digital thermostat PID putting inside the room. As well as thermocouple type K used to measure a temperature in the different location. Using a turbine flow meter to measure the volumetric flow rate of the...
refrigerant. Two pressure gauges of Bourdon type used to evaluate the low and the high pressure for the system. Twelve type (k) thermocouples were distributed in numerous places at both the high and low pressure sides of the system; at the inlet and outlet of the compressor, condenser, expansion valve, evaporator, indoor and outdoor display, a flow rate measuring device was placed at the outlet of the condenser Table 2. Shows the uncertainties of measuring devise; and Table 3. illustrate average uncertainties of calculated parameters by using moffat's (1988).

Table 2. Uncertainty of measuring devices

| Measured variables               | Type                   | Average error |
|----------------------------------|------------------------|---------------|
| Temperature                      | Thermocouple-K         | ± 1°C         |
| Pressure                         | Bourdon gauge          | ± 0.25 bar    |
| Refrigerant mass flow rate       | Turbine flow meter     | ± 0.001 kg/s  |
| Electric power                   | Watt meter             | ± 0.2 W       |

Table 3. Uncertainty of determined parameters

| parameters                      | Relative Errors |
|---------------------------------|-----------------|
| Cooling capacity                | ± 0.152         |
| Work of compressor              | ± 0.048         |
| Coefficient of performance      | ± 0.13          |
Figure 1. Schematic diagram of test rig

Figure 2. Test Rig

Accordingly, (1) High-pressure gauge (2) Low-pressure gauge (3) Compressor speed selector (4) Indoor and outdoor temperature displayers (5) Power lamp (6) Ampere meter (7) Main switch (8) Heater control knob (9) Heater switch (10) Evaporator speed selector (11) Evaporator (12) Thermostat (13) Electrical heater (14) Passenger compartment (15) Dehydrating filter (16) Condenser fan (17) Environment compartment (18) Condenser (19) Compressor (20) Electrical motor.

Figure 3. Test room
III. Experimental Procedure

Before the experiment began, the test device was placed inside a room so that it simulated the ambient conditions of the climate called the environmental room. The temperature inside can be controlled by digital temperature controller. Controls the operation and extinguishing of a thermal source consisting of a heater resistance with capacity 6kW supplied air by circulation fan. The temperature was varied inside the room from range from 30°C to 50°C in steps of 4°C. Each experiment it took usually period of time about 30 min until the temperature inside room distribution be uniform. The test have been done in winter season. After that selected the required internal thermal load: 700, 1000, or 1300W inside compartment which represented passenger's loads, then selecting the desired speed of compressor: 1450 r.p.m or 2900 r.p.m, the temperature of cabin was set up at 22°C. Then allow device operation until reach the steady state condition, after that the readings had taken. The data were temperatures, suction and discharge pressures, and refrigerant flow rate. The same producers was followed for both refrigerants.

4. Governing Equations

Performance parameters were calculated using data obtained from p-h diagram shown in fig.4 as follows:

![P-h diagram for vapor compression cycle](image)

**Figure 4.** P-h diagram for vapor compression cycle
1. By measuring the mass flow rate and the difference in enthalpy between inlet and outlet we can find the effect of the loads on the evaporator (Dincer and Kanoglu, 2010)

\[ R.E = h_1 - h_4 \]  
\[ Q_L = m(h_1 - h_4) \]  

2. From the difference in enthalpy between the inlet and outlet of the condenser can be found the amount of condenser Q rejected.

\[ Q_C = m(h_2 - h_3) \]  

3. The work of the compressor can be obtained from the difference in enthalpy between its inlet and outlet.

\[ W = m(h_2 - h_1) \]  

For expansion valve \( h_3 = h_4 \)

\[ \text{COP} = \frac{h_1 - h_4}{h_1 - h_2} \]  

By feeding the experimentally obtained data into the EES software program, we were able to obtain the thermodynamic properties of both refrigerants and calculate COP, compressor work, cooling capacity.

V. Results and Discussion

When selecting a refrigerant for a certain application it is important to know the relationship between the saturation temperature and pressure such as that shown in Fig.5 which demonstrates graphically the relation between saturation pressure and temperature it shows very close relationship with small increasing for R513A. So it can be used in applications that use R134a such as in A/C and refrigerators without need to perform any modulation to the pipes thickness or size. It is an advantage that the low pressure side has a low boiling point with a high vapor pressure and above atmospheric pressure; in order to prevent air and moisture entering the system and mixing with the refrigerant.

Fig.6 refers to the vapor displacement of any compressor depending upon the specific volume of the refrigerant vapor and total refrigeration capacity, a low specific vapor volume is always desirable. The refrigerant R513A has low specific vapor as compared with R134a which suggests higher mass flow rate because it has high vapor density.

Fig.7 represent the latent heat as a function of saturation temperature. Refrigeration depends mainly on the latent heat at evaporator temperature specific enthalpy of evaporation R513A has latent heat less than R134a which effect to the refrigeration effect. Latent heat depends upon molecules weight therefore those of low molecular weight has high latent heat.
Fig.8 and Fig.9 depict the effect of the internal load and the speed of the compressor upon the relation between the ambient temperature and the work of the compressor. The refrigerant mass flow rate and compression ratio affected the work of compressor. When the ambient temperature and the internal load inside the cabin increases compression ratio will increase. That mean more work need to compress the refrigerant vapor, and when the compressor speed increase the work will increase due to higher mass flow rate that passes through the compressor, work of compressor of R513A was slightly higher than R134a by about 1.5%-2%, achieving lower specific work due to lower compression ratio. Although it has little specific work, it is affected by the large amount of refrigerant that passing through the compressor.

Fig.10 illustrates the influence of the ambient temperature and the compressor speed on the compression ratio. When the outside temperature increases the compression ratio will increase due to the elevation of the high pressure for given evaporator temperature. And when speed of compressor increase that mean the difference between the discharge and suction pressures would be bigger. So the result of compression ratio increase. R513A has an average compression ratio lower by about 4.6 % -6% than R134a, which means the compressor has more ability to work with R513A and will reduces the leakage between the two sides especially at a high compression ratio.

Fig.11 shows the effect of the speed of the compressor on the relationship between the ambient temperature and cooling capacity. When increasing the outdoor temperature, the cooling capacity was decreased due to increase saturation temperature and decreased sub cooling in the condenser which led to higher quality of vapor entering the evaporator. This means lower difference of enthalpy across the evaporator. As the compressor speed increase, the cooling capacity increased due to increased mass flow rate. R513A has average cooling capacity higher by about 1.6% - 3% than R134a because of higher mass flow rate.

Fig.12 and Fig.13 show the relation between the discharge and suction pressures depending upon operating conditions and system applications. When the compressor speed increase the difference between the discharge and suction pressure would be higher. Therefore, the cycle diagram on the p-h chart will diverge. With increasing the ambient temperature, both suction and discharge pressures increases due to increased saturation temperature. Operating pressures of R513A are always slightly higher than R134a, because of it has higher pressure for the same saturation temperature. Therefore the thickness of pipes does not need redesigned.
Fig.14 and Fig.15 depict the effect of ambient temperature, different compressor speed and internal load on COP at. When ambient temperature increases, COP will decrease because of the cooling capacity decrease and work of compressor increase. While increasing the speed of compressor that will increase the cooling capacity as well as increasing work of compressor and power needed. Therefore, the coefficient of performance decreases. COP is considered an important parameter when evaluating the performance of any refrigeration cycle. R513A has coefficient of performance slightly higher than R134a about 1%-2%.

Fig.16 shows the effect of the ambient temperature and speed of compressor upon the discharge temperature of the compressor. When the ambient temperature increases the discharge temperature increase because of increasing condensation temperature and compression ratio. And when compressor speed increase the discharge temperature increase due to increase compression ratio. R513A has discharge temperature of compressor about 7.5%-8% less than R134a because of that depends upon thermodynamic properties especially specific heat of vapor. This is considered an important point, because the compressor lubricant oil degradation at high temperatures especially when there is a leak or compressor work at high temperatures.

Fig.17 and Fig.18 depict the effect of ambient temperatures and speed of compressor on the super heat and subcooling temperatures. When the compressor speed increasing the super heat increases because of the compressor pumps vapor faster than the liquid that is admitted to the evaporator. It evaporated at short time that led refrigerant to be more superheated, and when the ambient temperature increases the temperature of refrigerant vapor will increase due to the high temperature. Subcooling decreases when the ambient temperature increases, because of the lower difference between the condensing temperature (saturation) and ambient temperature. The super heat of R513A was greater than R134a because of the higher vapor density and high thermal conductivity for vapor. While subcooling of R513A was less than R134a because of lower density and thermal conductivity for liquid.

Fig.19 shows the effect of ambient temperature on the refrigerant effect and compressor speed. When the ambient temperature increases, the refrigeration effect decreases because of the difference in enthalpy between the inlet and outlet of the evaporator decreases, and when the compressor's speed increase the effect of the refrigeration decreases because the saturation temperature inside the evaporator decrease. R513A has average refrigeration effect less than R134a by about 6.4% -9% due to the latent heat of evaporation which smaller than R134a.
VI. Conclusion

This experimental study produced numerous results and can be summarized as follows:

1. R513A has average a lower compression ratio less than R134a about 4.6%-6% which suggests more ability for the compressor which led to a longer life for the compressor.

2. The temperature discharge of the compressor average was less than R134a about 7.5%-8% that means that there is not lubricant degradation oil during a leak or when system operate under high temperature range especially at hot climate like Iraq, isentropic efficiency of R513A was higher than R134a.

3. There were a convergent between operation pressures for suction and discharge, with small increasing of R513A discharge, suction pressures. Therefore it could be used on the same refrigeration devises and applications that work with R134a without increase gage or size piping of cycle.

4. Mass flow rate of R513A was more than R134a about 15%-17% this suggest more cooling capacity.

5. COP was comparable between refrigerants and slight increasing about 1%-2% for R513A

6. R513A can be used as an alternative refrigerant without need to make any modifications on the system and the same kind of POE lubricant.

Figure 5. pressure saturation vs. temperature saturation of refrigerants
Figure 6. Specific volume of vapor vs. saturation temperatures

Figure 7. Saturation temperature vs. latent heat of evaporation
Figure 8. Ambient temperature vs. work of compressor at 1450 r.p.m at different thermal loads.

Figure 9. Ambient temperature vs. work of compressor at 2900 r.p.m at different thermal loads.
Figure 10. Ambient temperature vs. pressure ratio

Figure 11. Ambient temperature vs. discharge pressure
Figure 12. Ambient temperature vs. suction pressure

Figure 13. Ambient temperature vs. work of compressor at 1450 r.p.m with different thermal loads
Figure 14. Ambient temperature vs. COP at compressor speed 2900 r.p.m with different loads

Figure 15. Ambient temperature vs. temperatures discharge of compressor
Figure 16. Ambient temperature vs. superheating

Figure 17. Ambient temperature vs. liquid subcooling
Figure 18. Ambient temperature vs. cooling effect

Figure 19. Ambient temperature vs. cooling capacity
NOMENCLATURE

AC = Air conditioning
CFC = Chlorofluorocarbon
COP = Coefficient of performance
GWP = Global warming potential
h = Enthalpy (kJ/kg)
HC = Hydrocarbon
HCFC = Hydro chlorofluorocarbons
HFC = Hydro fluorocarbons
HFO = hydrofloroolefin
ODP = Ozone depletion potential
PID = Programing inter digital
RE = Refrigeration effect (kJ/kg)
Tamb = Temperature ambient °C
VCR = Vapor compression refrigeration
W = Compressor work (kJ/kg)
ṁ = Refrigerant mass flow rate (kg/s)

References

I. Aljubury, I., Farhan, A. and Mussa, M. (2015) “Experimental Study of Interior Temperature Distribution Inside Parked Automobile Cabin”, Journal of Engineering, 21(3), pp. 1-10.

II. Aljubury, I. and Mohammed, M. (2019) “Heat Transfer Analysis of Conventional Round Tube and Microchannel Condensers in Automotive Air Conditioning System”, Journal of Engineering, 25(2), pp. 38-56. doi: 10.31026/j.eng.2019.02.03.

III. Carpenter, L.J., Reimann, S., Burkholder, J.B., Clerbaux, C., Hall, B.D., Hossaini, R., Laube, J.C., Yvon-Lewis, S.A., Engel, A., Montzka, S.A.
and Blake, D.R., 2014. Update on ozone-depleting substances (ODSs) and other gases of interest to the Montreal protocol. *Scientific assessment of ozone depletion: 2014*, pp.1-1.

IV. Dinçer, I. and Kanoğlu, M., 2010. Refrigeration Systems and Applications, Second Edition. secondedi ed. Refrigeration Systems and Applications, Second Edition. United Kingdom.

V. Karber, K.M., Abdelaziz, O. and Vineyard, E.A., 2012, July. Experimental performance of R-1234yf and R-1234ze as drop-in replacements for R-134a in domestic refrigerators. In *Proceedings of the International Refrigeration and Air Conditioning Conference* (pp. 16-19). Purdue University.

VI. Makhnatch, P., Mota-Babiloni, A., López-Belchí, A. and Khodabandeh, R., 2019. R450A and R513A as lower GWP mixtures for high ambient temperature countries: Experimental comparison with R134a. *Energy*, 166, pp.223-235.

VII. Mota-Babiloni, A., Navarro-Esbrí, J., Barragán-Cervera, Á., Molès, F. and Peris, B., 2015. Analysis based on EU Regulation No 517/2014 of new HFC/HFO mixtures as alternatives of high GWP refrigerants in refrigeration and HVAC systems. *International journal of refrigeration*, 52, pp.21-31.

VIII. Mota-Babiloni, A., Makhnatch, P., Khodabandeh, R. and Navarro-Esbrí, J., 2017. Experimental assessment of R134a and its lower GWP alternative R513A. *International Journal of Refrigeration*, 74, pp.682-688.

IX. Mota-Babiloni, A., Navarro-Esbrí, J., Pascual-Miralles, V., Barragán-Cervera, Á. and Maiorino, A., 2019. Experimental influence of an internal heat exchanger (IHX) using R513A and R134a in a vapor compression system. *Applied Thermal Engineering*, 147, pp.482-491.

X. Mota-Babiloni, A., Belman-Flores, J.M., Makhnatch, P., Navarro-Esbrí, J. and Barroso-Maldonado, J.M., 2018. Experimental exergy analysis of R513A to replace R134a in a small capacity refrigeration system. *Energy*, 162, pp.99-110.

XI. Rigby, M., Prinn, R.G., O’Doherty, S., Miller, B.R., Ivy, D., Mühle, J., Harth, C.M., Salameh, P.K., Arnold, T., Weiss, R.F. and Krummel, P.B., 2014. Recent and future trends in synthetic greenhouse gas radiative forcing. *Geophysical Research Letters*, 41(7), pp.2623-2630.

XII. Mota-Babiloni, A., Makhnatch, P., Khodabandeh, R. and Navarro-Esbrí, J., 2017. Experimental assessment of R134a and its lower GWP alternative R513A. *International Journal of Refrigeration*, 74, pp.682-688.
XIII. Mota-Babiloni, A., Navarro-Esbri, J., Barragán-Cervera, Á., Molés, F. and Peris, B., 2015. Analysis based on EU Regulation No 517/2014 of new HFC/HFO mixtures as alternatives of high GWP refrigerants in refrigeration and HVAC systems. International journal of refrigeration, 52, pp.21-31.

XIV. Mota-Babiloni, A., Navarro-Esbri, J., Pascual-Miralles, V., Barragán-Cervera, Á. and Maiorino, A., 2019. Experimental influence of an internal heat exchanger (IHX) using R513A and R134a in a vapor compression system. Applied Thermal Engineering, 147, pp.482-491.

XV. Reasor, P., Aute, V. and Radermacher, R., 2010. Refrigerant R1234yf performance comparison investigation.

XVI. Schultz, K.S. Kujak, and J. Majurin., 2015. Assessment of next generation refrigerant r513a to replace r134a for chiller products. Proc. 24th Int. Congr. Refrig.

XVII. Velders, G.J., Fahey, D.W., Daniel, J.S., Andersen, S.O. and McFarland, M., 2015. Future atmospheric abundances and climate forcings from scenarios of global and regional hydrofluorocarbon (HFC) emissions. Atmospheric Environment, 123, pp.200-209.

XVIII. Yang, M., Zhang, H., Meng, Z. and Qin, Y., 2019. Experimental study on R1234yf/R134a mixture (R513A) as R134a replacement in a domestic refrigerator. Applied Thermal Engineering, 146, pp.540-547.