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Comparison of R134a and R516A's Performance at Different Air Velocities in Two Evaporator Ejector Cooling System

Umit Iskan Mahmut Cuneyt Kahraman Mehmet Direk
Yalova University, Department of Energy Systems Engineering, Yalova, Turkey

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***Correspondence to**: Mahmut Cuneyt KAHRAMAN,*
Yalova University, Energy Systems Engineering, 77200, Yalova, TURKEY
E-Mail: mcuneyt.kahraman@yalova.edu.tr

ABSTRACT

This study aims experimentally to investigate the performance parameters of R134a and the alternative R516A refrigerant in two evaporator ejector cooling system (DEES) at different air velocities of evaporator#1. Firstly, the tests were carried out with R134a refrigerant under steady-state conditions at different air velocities and then repeated with low GWP R516A refrigerant. As the tests were carried out with R134a, higher cooling capacity was achieved at different air velocity values. When the air velocity value was 1.1, 1.7, 2.2, and 2.7 m s-1, the COP value obtained from the tests with R134a was 1%, 2%, 5%, and 4% higher than R516A, respectively. Additionally, test results illustrate that the higher air velocity contributed to increasing performance parameters, however air velocity higher than 2.2 m s^{-1} had a slight effect. The study concluded that R516A performance values are slightly lower than R134a performance and can be alternatively used as a refrigerant in vapor compression refrigeration (VCR) systems.

Keywords:
COP; Cooling capacity; R516A; R134a; Ejector

INTRODUCTION

The efficiency of heating and cooling devices has become one of the priority issues due to the concerns of natural gas supply. VCR systems, which are widely used in heating and cooling applications, must be operated with more environment-friendly refrigerants and in high energy efficiency. Determining the alternative refrigerants is based on several important criteria such as having similar thermodynamic properties, environmental factors (zero ODP, low GWP, etc.), and safety factors, etc [1,2]. The investigation of an alternative refrigerant to R134a, which is the most used refrigerator in VCR systems, is still on the agenda due to the high GWP value of R134a (above the legislation limits). Therefore, some refrigerants, such as R1234yf, R152a, and R1234ze(E), which have low GWP values, are investigated as alternatives due to their similar thermophysical properties to R134a [3,4]. On the other hand, energy loss arises in the refrigeration cycle due to the irreversibility of the throttling process with conventional methods in VCR, thus the COP value decreases [5]. The ejector utilization is one of the prominent approaches to prevent throttling losses. The system is alternatively operated by adding an ejector and a liquid-vapor separator to conventional VCR [6]. The COP value increases with ejector utilization in VCR [7,8]. However, the earnings in cooling capacity drop since the liquid-vapor separator cannot fully fulfill the separation [9]. In cooling systems with ejector expansion, a second evaporator is operated instead of a liquid-vapor separator. This provides another evaporation temperature and efficiency rising [10]. Lawrence and Elbel [5] concluded from their investigation that the COP value of the system operating with R134a and R1234yf with ejectors and two evaporators is higher by 8% and 12%, respectively, compared to the classical two-evaporator cooling system. Geng et al. [11] found that the COP value of DEES was between 16%-30% greater than VCR. Tahir Erdinc et al. [12], evaluated the performance of a heat pump utilizing an ejector. As a result of the study, it was found that the COP increased by 22.6% compared baseline heat pump system. Moreover, Alkhulaifi et al. [13] performed a study to investigate the DEES in terms of exergy and economy. The authors operated the second evaporator to condition the water used for cooling the battery. Consequently, thanks to ejector utilization, a 28% reduction in exergy destruction was achieved compared to the classical VCR. In another study, Gao et al. [14] theoretically compared DEES to VCR. In their study, they indicated that the COP value increased up to 26.5% - 44.8% and the exergy efficiency up to 32.4% - 41.7%. İşkan and Direk [15] studied the impact of expansion valves in a refrigeration system with an ejector and two evaporators sourced by water and air. They determined that the COP value of the single thermal expansion valve (TXV) system was 38% greater on average than the double TXV system. Liu et al. [16] tested a DEES with CO2 refrigerant by using two ejectors in their experimental study. It was indicated that the COP value found from the DEES was 15%-27% greater than the VCR. Environmental impact is a critical parameter for all refrigerant selections. In addition, many regulations have been implemented to reduce the negative effects of refrigerants used in VCR systems [17]. Alternatives to HFC refrigerants, azeotropic, near azeotropic and zeotropic refrigerants can be used. Azeotropic refrigerants are formed by the combination of two or more refrigerants and behave like a pure refrigerant at the same temperature and pressure [18]. Due to its performance values close to R134a, R516A as an azeotropic refrigerant is one of the most prominent refrigerants which are investigated recently [19,20]. In a study by İşkan and Direk [19], six different refrigerants were tested in DEES regarding their condenser temperature and entrainment ratio (ER) values. Finally, they suggested R516A (GWP value;131) owing to its better performance values. A refrigerant having an ODP value of '0' and a low GWP value (GWP < 150) are vital for environmental impact [21]. Therefore, the systems must be operated with lower GWP refrigerants while providing high efficiency values.

Table 1. Refrigerant specifications [22].

	R134a	R516A
Composition	-	R152a R1234yf R134a
Mass percentage (%)	-	14 /77.5 /8.5
critical pressure (kPa)	4059.3	3615.2
Boiling point (K)	247.1	243.8
Vapour density (kg m^{-3})	32.35	34.58
Critical temperature (K)	374.2	369.8
Vapour conductivity (W m^{-1} K^{-1})	$81.13.10^{-3}$	$70.09.10^{-3}$
Cp liquid (kJ kg^{-1} K^{-1})	1.425	1.456
Liquid density (kg m^{-3})	1206.7	1066.8
Cp vapor (kJ kg^{-1} K^{-1})	1.032	1.089
Liquid conductivity (Wm^{-1} K^{-1})	$13.83.10^{-3}$	$14.38.10^{-3}$
Latent heat value (kJ kg^{-1})	216.9	202.83
GWP	1300	131
ASHRAE safety class	A1	A2L

Table 1 shows that R156A has good environmental impact properties and has similar physical properties to R134a. However, only a few experimental studies were carried out by using R516A in refrigeration systems. Therefore, detailed experimental research of R156A in refrigeration systems is required to investigate its performance parameters. In this study, the performance of R516A was investigated as an alternative to R134a for several evaporator#1 air velocities in the experimental ejector-assisted refrigeration system. The tests were carried out in DEES at steady regime conditions.

EXPERIMENTAL FACILITY

DEES prototype experimental facility consists of a compressor, a condenser, an ejector, evaporators sourced by air and water, and auxiliary equipment (Fig. 1). The air of the condenser is regulated to the proper values by a fan placed in the duct. The air temperature is regulated by using electric heaters that are controlled with a proportional, integral, derivative (PID) control circuit. The power of the electrical heater in the duct increased or decreased as the fan speeds changed. Thus, the condensing temperature in the condenser could be kept constant during the experiments. Before the tests, the system was put into the vacuum, and after ensuring that there was no leakage, 1000 g of refrigerant was charged to the system. The refrigerant paths in the test facility are shown in Fig. 2.

In this section, the system operation is detailed with the numbers shown in Fig. 2,

- 1-2; the refrigerant leaves the compressor as high-pressure superheated vapor

- 2-3; the condenser transformed the refrigerant to a high-pressure liquid. Then, the refrigerant comes to the ejector in two different paths,

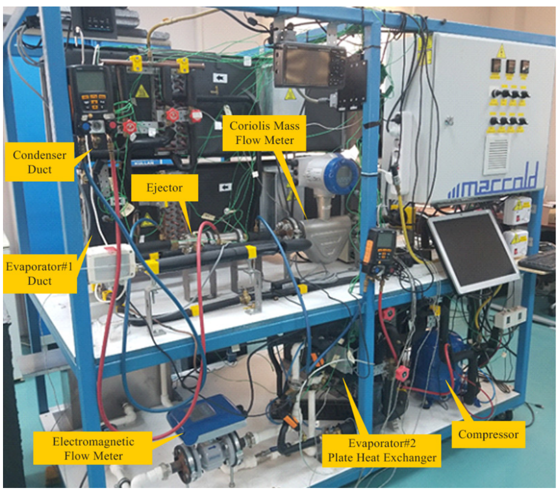

Figure 1. Test facility and the test components [19].

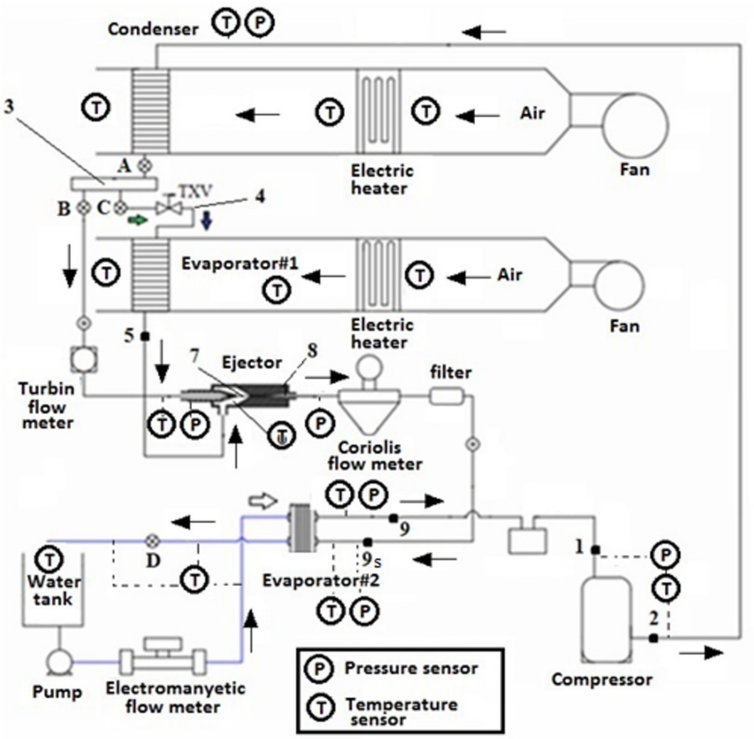

Figure 2. Refrigerant paths at the test facility [19].

- 3-4; second fluid expands at the TXV and transforms into a liquid-vapor mixture,

- 4-5; seconds flow comes to evaporator#1 at low pressure and evaporates in evaporator#1. The refrigerant transforms into low-pressure superheated vapor at the outlet of evaporator#1,

- 5-6; the second fluid comes to the ejector at point 6;

- 7-8; first and second flow mix in the mixing chamber,

- 8-9s; mixed refrigerant leaves the ejector and enters evaporator#2;

Table 2. Technical specifications of the test components.

Test Component	*Specification*
Compressor	*Reciprocating compressor* *Volume: 30.23 cm³ rev⁻¹*
Condenser	*Total area: 9.9 m²* *357mm x 330mm x 132mm*
Evaporator#1	*300mm x 305mm x 110mm* *Total area: 6.6 m²*
Evaporator#2	*Plate heat exchanger, 12 plates* *Heat transfer area: 0.5 m²*

- 9s-9; the refrigerant leaving evaporator#2 arrives at the inlet of the compressor,

- 9-1; finally, the cycle is completed.

The heat from the water cycle evaporates the refrigerant in Evaporator#2. The water cycle, which is shown in Fig. 2, consists of a heat exchanger, a water tank, and a pump. The electric heater of the water tank regulates the water temperature. In addition, the pump brings the mass flow rate of the water to the proper values that an electromagnetic flowmeter is read by.

The ejector is a stagnant system component that makes DEES distinct from VCR and goals to drop the compressor power by making energy conversions and thus raising the COP value.

The ejector operated in the tests was a constant-pressure model ejector.

The ejector energy conversions are as follows.

- The refrigerant directly coming from the condenser enters the ejector from the ejector's first inlet with high pressure. It is throttled at point 7, reducing its pressure and increasing its velocity and kinetic energy.

- The low-pressure refrigerant evaporating in Evaporator#1 is applied to a suction force at the ejector's second

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Table 3. Technical capacities of the devices [23].

Measurement Type	Device	Accuracy
Frequency Inverter	*ABB-ACS 355*	*±0.2 %*
Pressure	*Electronic Manifold*	*± 0.5 %*
Air Velocity	*Anemometer*	*± 2.0 %*
Mass Flow	*Turbine flowmeter*	*± 0.1 %*
Water Flow	*Electromagnetic*	*± 0.3 %*
Mass Flow	*Coriolis flowmeter*	*± 0.05 %*
Temperature	*K type Thermocouple*	*± 1.5 ºC*
Power Measurement	*Brymen BM-157 Clamp Meter*	*± 0.5 %*

inlet owing to the vacuum impact and these two fluids are mixed in the mixing section.

- The velocity of mixed flow decreases, and the pressure rises in the diffuser. High pressure refrigerant leaving the ejector enters the compressor, thus irreversibility and earnings from compressor work are obtained.

The test facility was equipped with three mass flowmeters to measure water and refrigerant flow rate. An electromagnetic flowmeter measured the water mass flow rate of evaporator#2. Electronic manifold and thermocouples measured temperature and pressure values before each of piece equipment respectively. The data transfer system transferred all measured data to the computer. The uncertainty analysis equation, which is shown in Table 4, was solved by applying the accuracy values of measurement devices, as stated in Table 3. As a result, uncertainty values of evaporator#1 cooling capacity, total cooling capacity, compressor power and COP value were calculated %0.8, %0.95, %0.25 ve % 4.3 respectively.

CALCULATION OF COMPONENT PARAMETERS

Component parameters were calculated by applying the experimental data to the equations shown in Table 4. Moreover, some values indicated in Table 5 are kept constant during the tests.

RESULTS AND DISCUSSION

Firstly, the tests were carried out with R134a refrigerant under steady-state conditions at different air velocities and then repeated with low GWP R516A refrigerant. The air velocity values of evaporator#1 were preferred between 1-3 m s^{-1} but kept constant during the test.

Fig. 3 indicates the change of compressor inlet pressure depending on the evaporator#1 air velocity. The figure indicates the rising of the compressor inlet pressures in both refrigerants as the evaporator#1 air velocity increased. For instance, as the air velocity value increased from 1.1 m s^{-1} to 2.2 m s^{-1}, compressor inlet pressures increased by 6% and 5% for R134a and R516A, respectively. However, the air velocity higher than 2.2 m s^{-1} did not affect the compressor inlet pressure for both refrigerants. Furthermore, higher comp-

Table 4. Equations used to calculate component parameters.

Component Parameters	Equations
Compressor power	$\dot{W}_{comp} = (\mathrm{h}_2 - \mathrm{h}_1).\dot{m}_{total}$
Compressor Outlet Enthalpy	$h_2 = \frac{h_{2s} - h_1}{\eta_{isentropic}} + h_1$
Evaporator#1 Cooling Capacity	$\dot{Q}_{evap\#1} = (\mathrm{h}_5 - \mathrm{h}_4).\dot{m}_5$
Total Cooling Capacity	$\dot{Q}_{evap,total} = \dot{Q}_{evap\#1} + \dot{Q}_{evap\#2}$
Total mass flow rate	$\dot{m}_{total} = \dot{m}_7 + \dot{m}_5$
COP	$COP_{cooling} = \frac{\dot{Q}_{evap\#1} + \dot{Q}_{evap\#2}}{\dot{W}_{comp}}$
ER	$ER = \frac{\dot{m}_5}{\dot{m}_{total}}$
Superheating Degree of Evaporator#1	$SuperheatingDegree = T_{evap\#1,out} - T_{evap\#1}$
Superheating Degree of Evaporator#2	$SuperheatingDegree = T_{evap\#2,out} - T_{evap\#2}$
Uncertainty Analysis	$U_y = \sqrt{\sum_{i=1}^{n} (\frac{\partial_y}{\partial_{xi}} \mathrm{u}_{\mathrm{x_i}})^2}$

Table 5. Constant values in the tests.

Parameters	Values
Water mass flow rate (kg s⁻¹)	0.282
Condensing temperature (°C)	35
Ambient temperature (°C)	25
ER	0.70 – 0.75
Water temperature (°C)	35

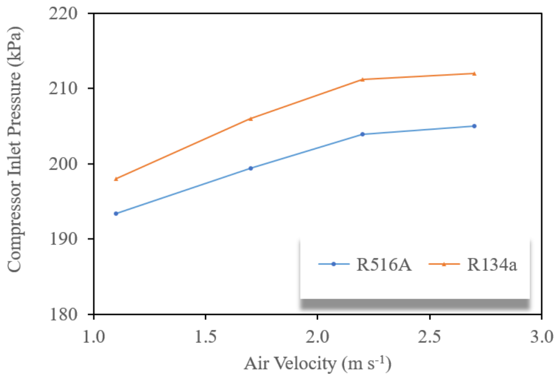

Figure 3. Change of compressor inlet pressures depending on air velocity.

ressor inlet pressures were obtained in the R134a case for all air velocities owing to higher critical pressure.

Fig. 4 shows the change of evaporator#1 inlet pressures depending on air velocities. The figure indicates that evaporator#1 inlet pressures increased in both refrigerants as the evaporator#1 air velocity increased. However, the higher air velocity of more than 2.2 m s^{-1} did not have a high effect on the evaporator#1 inlet pressures for both fluids. Moreover, higher evaporator#1 inlet pressures were obtained in the R134a case for all air velocities owing to higher critical pressure. The difference in evaporator#1 inlet pressures of both cases was up to 5%.

Fig. 5 indicates the change of evaporator#1 cooling capacity depending on air velocities. The figure illustrates the

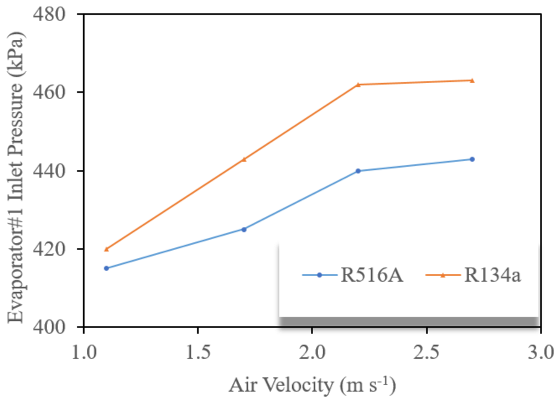

Figure 4. Change of evaporator#1 inlet pressures depending on air velocity.

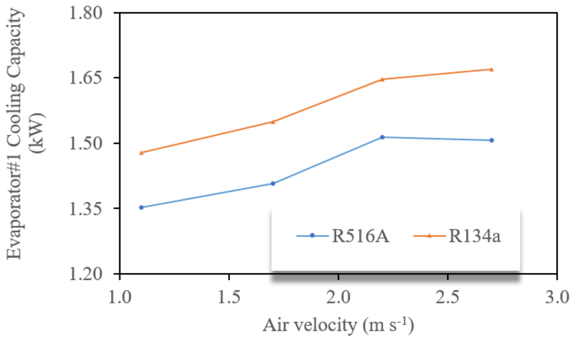

Figure 5. Change of evaporator#1 cooling capacity depending on air velocity.

slightly rising of the evaporator#1 cooling capacity in both refrigerants as the evaporator#1 air velocity increased. The main reason for that was the higher heat transfer as the air velocity increased. However, as in the previous cases, the higher air velocity higher than 2.2 m s^{-1} did not have a strong effect on the evaporator#1 cooling capacity for both fluids. Moreover, higher evaporator#1 cooling capacity was calculated in the R134a case for all air velocities. For instance, the difference in evaporator#1 cooling capacity of both cases was approximately 5%.

Fig. 6 illustrates the change of mass flow rate depending on air velocities. Since the ER value was constant in the range of 0.7-0.75 during the tests, the evaporation capacity of evaporator#1 became the most important parameter affecting the mass flow rates. Higher air velocity values increased the mass flow rate difference between refrigerants. While the air velocity was 1.7 m s^{-1}, the mass flow rate of R134a was 1% higher than R516A, on the other hand, while the air velocity is 2.2 m s^{-1}, the difference was 4%. The reason for this was the difference between the vapor density values of the refrigerants evaporating in evaporator#1. R516A vapor density is higher for the same temperatures. As Fig. 4 illustrates, the vapor pressure of R134a is higher owing to lower evaporator#1 inlet pressures of R516A.

Fig. 7 illustrates the change of total cooling capacity depending on air velocities. Higher total cooling capacities

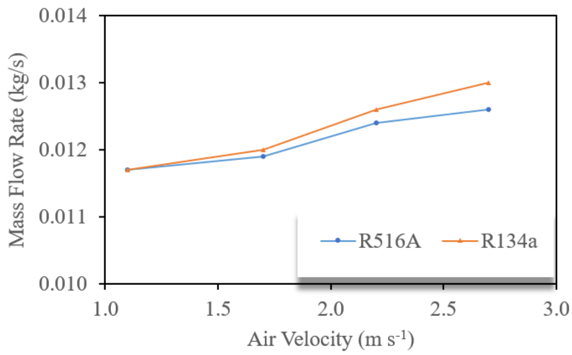

Figure 6. Change of mass flow rate depending on air velocity.

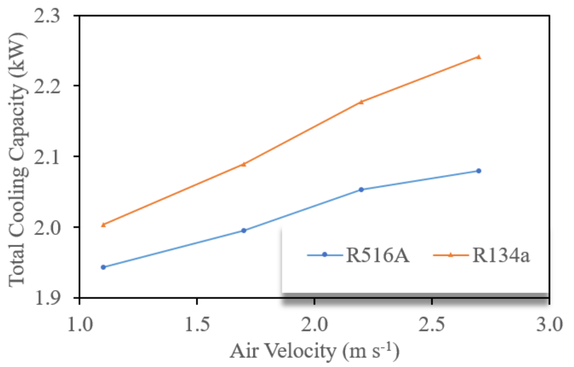

Figure 7. The change of total cooling capacity depending on air velocity.

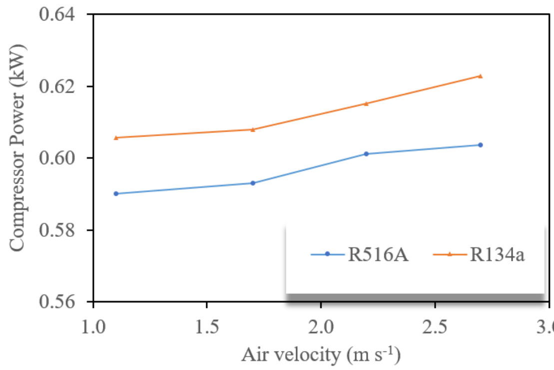

Figure 9. The change of compressor power depending on air velocity.

were calculated in the R134 case for all air velocities. For instance, the total cooling capacity of R134a at 2.2 m s^{-1} velocity is 170 W more than R516A. The reasons for this difference were not only mass flow rates and also higher R134a latent heat value. Moreover, Fig. 7 illustrates the rising of the refrigerant heat transfer as the air velocity of evaporator#1 increased, thus the total cooling capacities increased.

Fig. 8 indicates the change of COP depending on air velocities. This figure proves the rising of COP as the evaporator#1 air velocity increased. However, while the air velocity was over 2.2 m s^{-1}, the rising rate slightly dropped. The main reason for this trend was the rising of compressor power with increasing total cooling capacity. Additionally, slightly higher COP was calculated in the R134 case for all air velocities. For instance, the maximum difference in COP of both cases was approximately 5%. These results match the results of Heredia-Aricapa et al. [1].

Fig. 9 indicates the change of compressor power depending on air velocities. Higher compressor power was determined in the R134 case for all air velocities. For example, while the air velocity was 2.2 m s^{-1}, R134a compressor power was 25 W higher than R516A. The main reasons for differences were firstly mass flow rate owing to air velocity and secondly, the density values corresponding to the compressor inlet pressures. Higher air velocity required higher compressor power due to the rising mass flow rate.

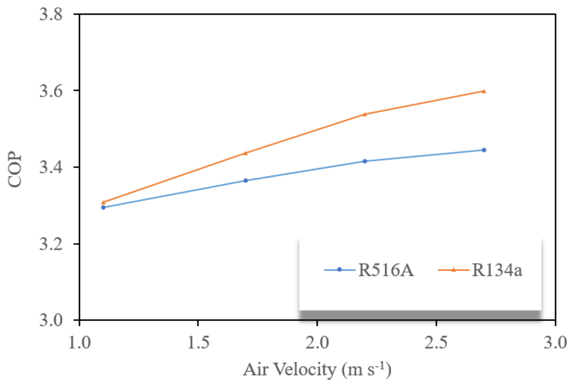

Figure 8. The change of COP depending on air velocity.

CONCLUSION

The present study investigated the air velocity-dependent performance parameters of DEES test facility operating with widely utilized R134a and low GWP R516A refrigerants. This study firstly aims to analyse the performance parameters of these two refrigerants. Another goal of this study is to investigate the impact of air velocity on performance parameters such as total cooling capacity, COP, compressor power, etc. It was found that the rising of the air velocity improved performance parameters. However, when the air velocity was over 2.2 m s^{-1}, only a slight effect could be witnessed. Moreover, higher performance parameters such as total cooling capacity and COP were found when the system was operated with R134a. For instance, the COP values obtained from R134a were 1%, 2%, 5%, and 4% higher than R516A in several air velocities ranging between 1.1-2.7 m s^{-1}. It is concluded that R134a performance parameters were only slightly higher than R516A. Therefore, low GWP R516A is recommended as an alternative to R134a due to low GWP and low flammability. The limitation of the present study is that only one alternative refrigerant was tested with a limited air velocity range. In future studies, new tests should be carried out with other low GWP refrigerants with a wider range of air velocity.

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NOMENCLATURE

COP	Coefficient of performance
DEES	Dual evaporator ejector system
ER	Entrainment ratio
GWP	Global warming potential
HFC	Hydrogen-Fluorine-Carbon composition refrigerant
ODP	Ozone depletion potential
TXV	Thermal expansion valve
VCR	Vapor compression refrigeration

SYMBOLS

h	Enthalpy [kJ kg^{-1}]
i	Inlet
is	Isentropic
$\dot{m}$	Mass flow rate [kg s^{-1}]
o	Outlet
P	Pressure [kPa]
Q	Heat transfer [kW]
T	Temperature [K]
tot	Total
W	Work [kW]

SUBSCRIPTS

comp	Compressor
evap	Evaporator
ref	Refrigerant

CONFLICT OF INTEREST

Authors approve that to the best of their knowledge, there is not any conflict of interest or common interest with an institution/organization or a person that may affect the review process of the paper.

AUTHOR CONTRIBUTION

Umit Iskan: Investigation, Carrying out the Tests, Formal Analysis
Mahmut Cuneyt Kahraman: Conceptualization, Methodology, Formal Analysis, Visualization, Writing – Original Draft.
Mehmet Direk: Conceptualization, Supervision, Writing – Review & Editing

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