ESPOCH Congresses: The Ecuadorian Journal of S.T.E.A.M.
Volume 3, Issue no. 4, DOI 10.18502/espoch.v3i4.17166
Production and Hosting by Knowledge E

Conference Paper

Analysis of Automotive Air Conditioning System Performance with Alternative Refrigerants to R-134a

Análisis del desempeño del sistema de aire acondicionado automotriz con refrigerantes alternativos al R-134a

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X CONGRESO INTERNACIONAL DE CIENCIA TECNOLOGÍA EMPRENDIMIENTO E INNOVACIÓN SECTEI 2023

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Published: 25 September 2024

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Abstract

This work analyzed the performance of the automotive air conditioning system, operating with several alternative refrigerants to R-134a. For this, a thermodynamic model of the automotive vapor compression cycle was implemented to determine the COP of the system, the refrigeration capacity, and the consumption of the compressor. The model considers the compressor rotation speed, ambient temperature, cabin temperature, air speed in the condenser, and volumetric airflow in the evaporator as operating parameters. The refrigerants chosen for the analysis were R-152a, R-1234yf, R-290, R-410A, and R-32. The simulation was carried out with climatic data from the city of Riobamba, and temperatures inside the cabin were measured experimentally. The study shows that the refrigerant R-152a has a COP 10% better than R-134a and could be an alternative for direct replacement of refrigerant. On the other hand, R-1234yf presented a COP 9% lower than that of R-134a, which requires adjustments in the sizing of components to provide the same cooling capacity as R-134a. Finally, R-290, R-410A, and R-32 refrigerants have lower performance and higher compressor consumption.

Keywords: *automotive air conditioning, vapor compression cycle, COP, cooling capacity, refrigerants.*

Resumen

En este trabajo se analizó el desempeño del sistema de aire acondicionado automotriz operando con varios refrigerantes alternativos al R-134a. Para ello, se implementó un modelo termodinámico del ciclo de compresión de vapor automotriz para determinar el COP del sistema, la capacidad frigorífica y el consumo del compresor. El modelo considera como parámetros de funcionamiento la velocidad de giro del compresor, temperatura ambiente, temperatura del habitáculo, velocidad del aire en el condensador y flujo volumétrico del aire en el evaporador. Los refrigerantes escogidos para el análisis fueron el R-152a, R-1234yf, R-290, R-410A, y el R-32. La simulación se realizó con datos climáticos de la ciudad de Riobamba y temperaturas del interior del habitáculo medidas experimentalmente.

How to cite this article: DA Silva-Espinoza, ME Taipe-Chariguamán, DC Vásconez-Núñez, SM Bonilla-Novillo, and FM Tello-Oquendo (2024). Analysis of Automotive Air Conditioning System Performance with Alternative Refrigerants to R-134a. *ESPOCH Congresses: The Ecuadorian Journal of S.T.E.A.M.*, 3(4), 72–86. DOI 10.18502/espoch.v3i4.17166

El estudio muestra que el refrigerante R-152a presenta un COP 10% mejor que el R-134a, pudiendo ser una alternativa para un remplazo directo de refrigerante. Por otro lado, el R-1234yf presentó un COP 9% menor que el del R-134a, con lo cual se requieren ajustes en el dimensionamiento de componentes para proporcionar la misma capacidad de refrigeración que el R-134a. Finalmente, los refrigerantes R-290, R-410A y R-32 presentan un menor rendimiento y mayor consumo de compresor.

Palabras Clave: *aire acondicionado automotriz, ciclo de compresión de vapor, COP, capacidad frigorífica, refrigerantes.*

1. Introduction

The automotive air conditioning system evacuates the high temperatures that occur inside the cabin of a vehicle, giving a feeling of thermal comfort. This system uses a refrigerant responsible for absorbing and releasing heat. The main part of its operation is due to the compressor of the automotive air conditioning system, which is connected to the internal combustion engine by a belt. Consequently, when using this air conditioning system, the power of the car is reduced and fuel consumption increases, which turns out to be a major environmental problem. Currently, the refrigerant used in automotive air conditioning systems is R-134a, which has a global warming potential of 430 and is somewhat harmful to the environment. Since it remains in the atmosphere for 14 years compared to R-1234yf, which only remains in the atmosphere for 11 days [1, 2].

One of the ways to analyze the performance of the air conditioning system and the use of refrigerants is through thermodynamic models. The thermodynamic model for the automotive air conditioning system brings together equations that provide the heat transfer produced in heat exchangers and the work of the compressor. In the condenser modeling, the following heat transfer zones are identified: desuperheating, condensation, and subcooling. For the evaporator, the superheating and evaporation zones are identified. For the compressor, volumetric and isentropic efficiency curves provided by experimental studies are used. The modeling can be implemented in calculation programs that can have in their libraries the physical characteristics of the refrigerants, thermodynamic functions, and transport properties with high precision for various fluids and solid materials. Thus, allowing us to efficiently integrate all the equations.

Several studies have evaluated the performance of some refrigerants, seeking to replace R-134a. Mendoza J. [3] studied R-1234yf and R-152a refrigerants as alternatives to R-134a for refrigeration systems in terms of coefficient of performance (COP), refrigeration load, compressor consumption, and discharge temperature. The COP results using R-152a increase between 2.46% and 30.79%, and with the refrigerant R-1234yf, they decrease from 2.7% to 18.14% compared to the refrigerant R-134a. The refrigeration load of R-152a is similar to that of R-1234yf at 270 K at the condensation temperature. Additionally, the refrigeration load of R-152a is similar to that of R-134a, except at a condensation temperature of 313 K. Regarding the compressor discharge temperature, for the R-1234yf refrigerant, it is lower than R-134a, and with R-152a, the discharge temperature is higher than the temperature of R-134a. Similar studies were carried out in [2, 4]. However, the studies carried out analyses with dimensions of components other than those of a car. On the other hand, no analysis of automotive air conditioning systems has been found under environmental conditions in the Ecuadorian mountains.

In the present study, the performance of various refrigerants in an automotive air conditioning system is thermodynamically analyzed. To do this, the vapor compression cycle is modeled, and simulations are carried out under the conditions of the Ecuadorian mountains to compare the refrigerants in terms of COP thermals, refrigeration capacity, compressor consumption, and discharge temperature.

2. Materials and Methods

Figure **1** shows the five basic components of the automotive air conditioning system. Condenser, expansion valve, compressor, evaporator, and receiver/drier.

The operation of the air conditioning system basically consists of four processes. First, the low-temperature, low-pressure liquid refrigerant evaporates, which must extract heat from the hot air in the cabin. Next, the refrigerant in a gaseous state is sucked into the compressor, in the compressor the pressure and temperature of the refrigerant is increased. This vapor is directed to the condenser where the refrigerant is condensed by heat transfer with cold air that is forced from the outside. Finally, the refrigerant in liquid state passes through the expansion device that is responsible for depressurizing the liquid to evaporate it, lower its temperature and introduce cooler air into the cabin.

The operating conditions for the components of the air conditioning system are represented by the input variables. The fixed parameters are associated with the geographical location and the physical properties of the refrigerants. Finally, the output variables provide points of comparison between the different refrigerant alternatives.

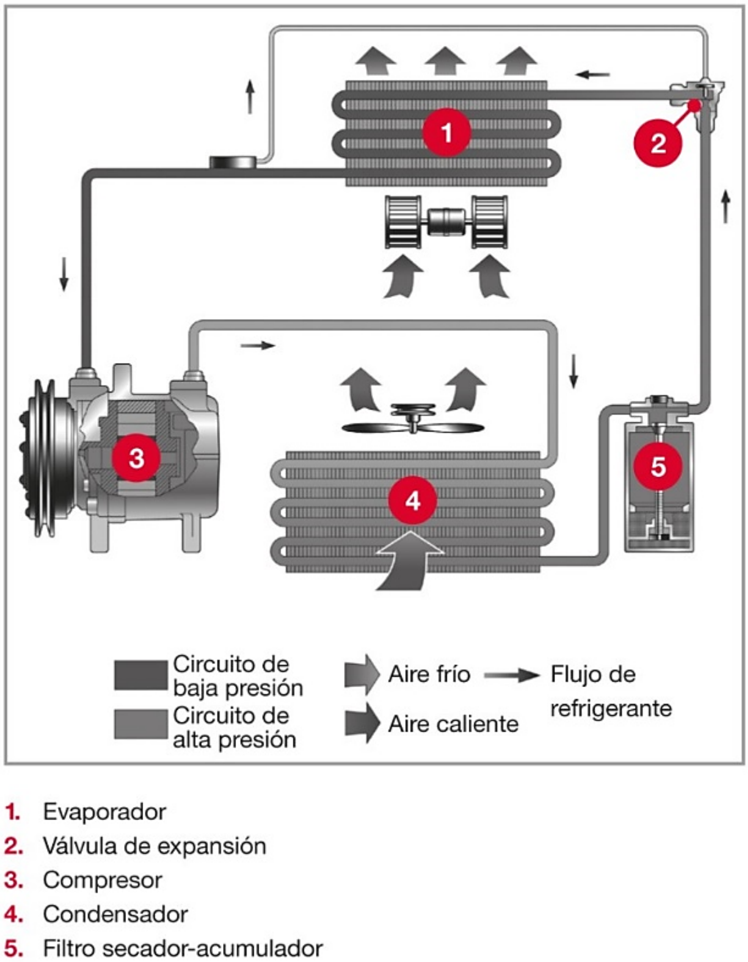

Figura 1

Constituent elements of the air conditioning system (Denso 2018).

2.1. Thermodynamic model of the vapor compression cycle

Figure **2** summarizes the scheme of the implemented thermodynamic model, describing the input variables, the fixed parameters, and the output variables. Each component of the vapor compression cycle was modeled with energy balance equations.

The condenser modeling is carried out by zones; for the initial calculations, the following equations, according to Nellis and Klein [5], are used:

$F_{z1,2,3} = \frac{L_{z1,2,3}}{L_{tubo}}$ (1)

Where:

$$F_{z1,2,3}{=}Exchanger\ fraction\ [-]$$

$$L_{z1,2,3}{=}\ Equivalent\ length\ for\ heat\ exchanger\ zones\ [m]$$

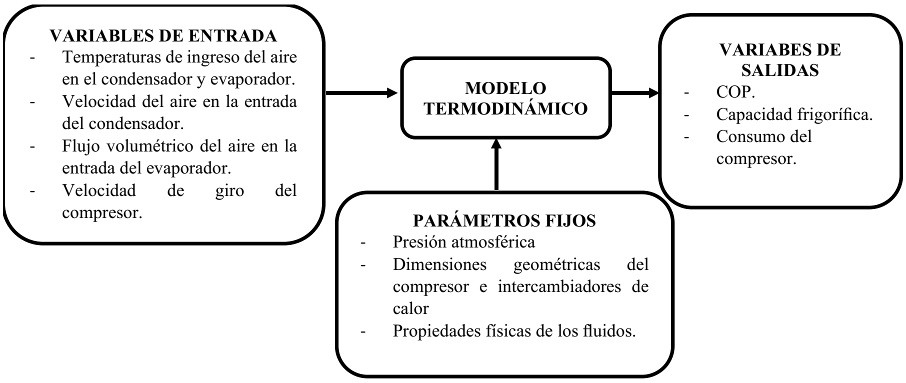

Figura 2

Variable definition for the model.

$$L_{tubo}= Exchanger\ total\ length\ [m]$$

Equation 1 determines the portion of the entire condenser allocated to each of the zones. With equation 2, the average temperature of the coolant is determined, which will allow its physical properties to be evaluated.

$T_{prom,R} = \frac{T_{R,V,sh}+T_{R,in,cond}}{2}$ (2)

Where:

$$T_{prom,R}= Average\ coolant\ temperature\ [℃]$$

$$T_{R,V,sh}= Coolant\ temperature\ in\ the\ desuperheat\ zone\ [℃]$$

$$T_{R,in,cond}= Refrigerant\ temperature\ at\ condenser\ inlet\ [℃]$$

Considering the thermal resistance for the three zones of the condenser, the thermal conductance is calculated using equation 3.

$UA_{1,2,3} = \frac{1}{R_{1,2,3}}$ (3)

Where:

$$U_{1,2,3}= Thermal\ Conductance\ \left[\frac{W}{m^2.℃}\right]$$

$$A_{1,2,3}= Heat\ transfer\ area\ [m^2]$$

$$R_{1,2,3}= Thermal\ resistance\ for\ heat\ exchanger\ zones\ \left[\frac{℃}{W}\right]$$

Using equation 4, the heat transferred in the desuperheating, condensation and subcooling zone, respectively, is determined.

$\dot{q}_{sh} = \dot{m}_R.[h_{R,in} - h_{R,V,sat}]$ (4)

$\dot{q}_{sat} = \dot{m}_R.[h_{R,V,sat} - h_{R,L,sat}]$(5)

$\dot{q}_{max,sc} = \dot{C}_{min}.[T_{R,sc} - T_{air,in}]$(6)

Where:

$$\dot{q}_{sh}= Heat\ transferred\ in\ the\ desuperheating\ zone.\ [kW]$$

$$h_{R,in}= Specific\ enthalpy\ of\ the\ refrigerant\ at\ the\ condenser\ inlet\ \left[\frac{kJ}{kg}\right]$$

$$h_{R,V,sat}=\ Specific\ enthalpy\ of\ refrigerant\ in\ saturated\ vapor\ \left[\frac{kJ}{kg}\right]$$

$$\dot{q}_{sat}=Heat\ transferred\ in\ the\ saturation\ zone\ [kW]$$

$$h_{R,L,sat}= Specific\ enthalpy\ of\ saturated\ liquid\ refrigerant\ \left[\frac{kJ}{kg}\right]$$

$$\dot{C}_{min}= minimum\ thermal\ capacitance\ \left[\frac{kJ}{℃}\right]$$

$$\dot{q}_{max,sc}= Heat\ transferred\ in\ the\ subcooling\ zone\ [kW]$$

$$T_{R,sc}= Subcooled\ coolant\ temperature\ [℃]$$

$$T_{air,in}= air\ temperature[℃]$$

$$\dot{m}_R= Mass\ flow\ of\ refrigerant\ in\ the\ heat\ exchanger\ \left[\frac{kg}{s}\right]$$

Finally, the total heat transfer rate is obtained from the sum of the heat transferred in each zone.

$\dot{q}_{R,condensador} = \dot{q}_{sh} + \dot{q}_{sat} + \dot{q}_{sc}$ (7)

Where:

$$\dot{q}_{R,condensador}=\ Total\ heat\ transferred\ in\ the\ condenser\ [kW]$$

In the case of the evaporator, the same sequence is used in modeling. The evaporator fraction for the reheating and evaporation zones is determined. The average temperature used to evaluate the properties of the refrigerants is calculated, and the thermal resistance in each of the zones is determined, with which the thermal conductance is determined.

With equations 8 and 9, the heat transferred in both zones of the evaporator is calculated.

$\dot{q}_{bif,evap} = \dot{m}_R.[h_{R,V,sat,ev} - h_4]$ (8)

$\dot{q}_{sh,ev} = \dot{q}_{max,sh,ev} * \epsilon_{sh,ev}$ (9)

The total heat transferred in the evaporator or also called refrigeration capacity is calculated by the sum of the heat transferred in both zones of the evaporator using equation 10.

$\dot{q}_{evaporador} = \dot{q}_{sh,ev} + \dot{q}_{bif,ev}$ (10)

The compressor of the air conditioning system was modeled based on the volumetric and isentropic efficiency curves, from which a linear regression was performed based on the pressure relations and the compressor rotation speed.

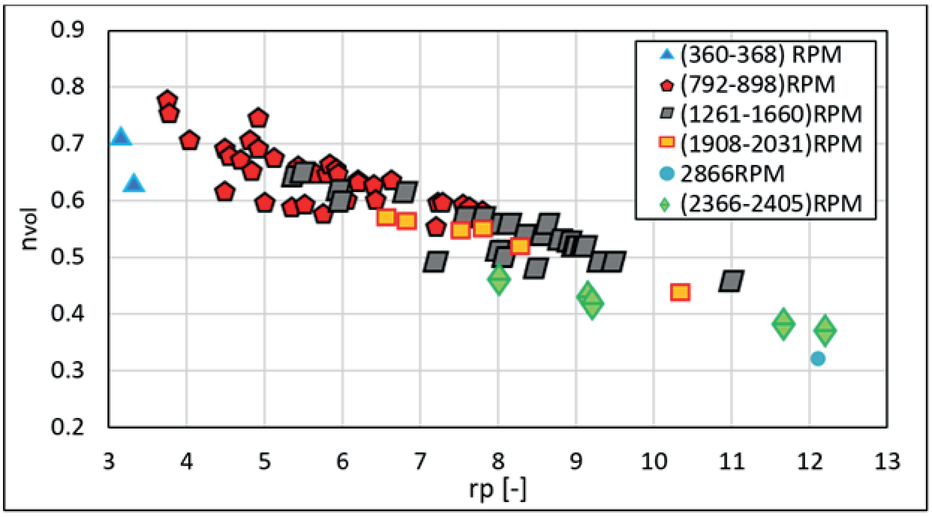

Figura 3

Compressor volumetric efficiency vs pressure ratio (6).

The volumetric efficiency is represented by equation 11.

$n_{vol} = 0,8537 - 0,02927 * rp - 0,000052 \ * n$ (11)

Where:

$$rp = \text{Pressure ratio } [-]$$

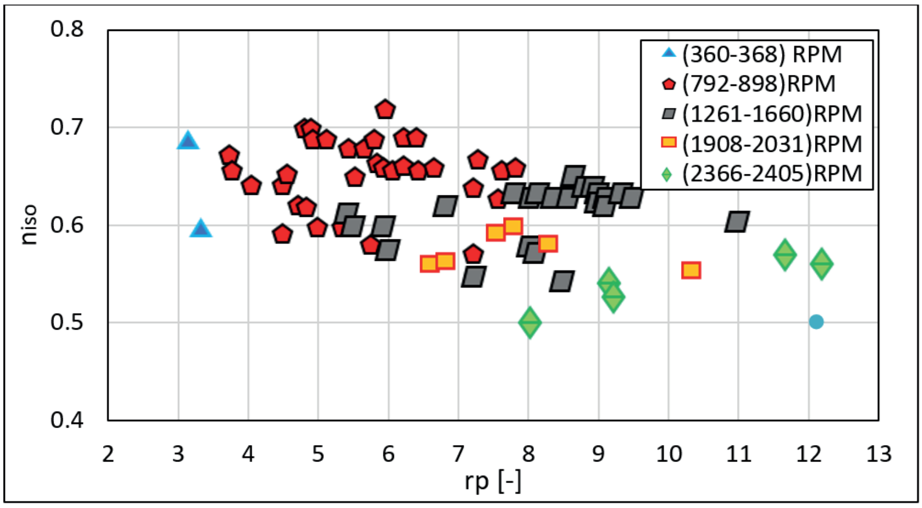

Figura 4

Compressor isentropic efficiency vs pressure ratio [6].

$$n = Compressor\ rotation\ speed\ \left[\frac{rev}{min}\right]$$

The isentropic efficiency is represented by equation 12.

$n_{iso} = 0,6848 + 0,00755 * rp - 0,00009 * n$ (12)

The volumetric flow is represented by equation 13.

$\dot{m}_r = n_{vol} * \rho_{in,comp} * \frac{n}{60} * d_v$ Where: $\rho_{in,comp}$= $Refrigerant\ density\ at\ the\ compressor\ inlet$ $\left[\frac{kg}{m^3}\right]$ d_v= $Compressor\ displacement\ \left[m^3\right]$ (13)

Using the definition of isentropic efficiency, the compressor output specific enthalpy (h_{out}) is calculated with equation 14.

$n_{iso} = \frac{h_{out,s} - h_{in}}{h_{out} - h_1}$ (14)

Where:

$$h_{in} = Specific\ enthalpy\ of\ the\ refrigerant\ at\ the\ compressor\ inlet \frac{kJ}{kg}]$$

$$h_{out,s} = Specific\ enthalpy\ of\ the\ refrigerant\ at\ the\ compressor\ outlet,$$

$$isentropic\ process\ \left[\frac{kJ}{kg}\right]$$

$$h_{out} = Specific\ enthalpy\ of\ the\ refrigerant\ at\ the\ compressor\ outlet,$$

$$considering\ a\ real\ process\ \left[\frac{kJ}{kg}\right]$$

Unifying the previous equations, equation 15 is proposed to calculate the compressor's energy consumption.

$$\dot{W}_{real} = \frac{\dot{m}_r * (h_{out,s} - h_{in})}{n_{iso} * (1-\epsilon)} \quad (15)$$

Where:

$$\dot{W}_{real} = Compressor\ energy\ consumption\ [kW]$$

$$\epsilon = \ compressor\ energy\ losses\ to\ the\ environment\ [\%]$$

The operating conditions that were used to simulate the operation of the air conditioning system were defined based on the rotation speed of the compressor. Table 1 shows the operating conditions at idle, Table 2 shows city driving, and Table 3 shows highway driving. The ambient temperatures were taken from the INHAMI annual report for 2020. The temperature in the cabin was taken from the experimental study [7], where the evolution of the interior temperature of the cabin of a vehicle was measured throughout the day in the city of Riobamba. Finally, the volumetric flow of air into the evaporator is a function of the blower knob levels on the front panel.

Tabla 1

Operating conditions for a compressor rotation speed of 800 rpm.

$N\ [rpm]$	800					
$V_{air}[m/s]$	1,92					
$T_{ambiente}$ [℃]	13,5		17		21	
$T_{habitâculo}$[℃]	25	35	25	35	25	35
$\dot{V}_{air,in,ev}[m^3/s]$	0,10					

Tabla 2

Operating conditions for a compressor rotation speed of 1600 rpm.

$N\ [rpm]$	1600														
$V_{air}[m/s]$	2,283														
$T_{ambiente}$ [℃]	13,5					17					21				
$T_{habitâculo}$[℃]	25	35	45	55	65	25	35	45	55	65	25	35	45	55	65
$\dot{V}_{air,in,ev}[m^3/s]$	0,15		0,10		0,08	0,15		0,10		0,08	0,15		0,10		0,08

Refrigerant alternatives are limited under the following considerations: that they work in a subcritical state and, at the same time, have a low level of ozone layer destruction (ODP). This implies that these refrigerants must not be toxic. Table 4 details the refrigerants used for the study and their classification in terms of safety.

Tabla 3

Operating conditions for a compressor rotation speed of 2400 rpm.

N $[rpm]$	2400											
$V_{air}[m/s]$	3,004											
$T_{ambiente}$ [℃]	13,5				17				21			
$T_{habitáculo}$[℃]	35	45	55	65	35	45	55	65	35	45	55	65
$\dot{V}_{air,in,ev}[m^3/s]$	0,15	0,12		0,1	0,15	0,12		0,1	0,15	0,12		0,10

Tabla 4

Alternative refrigerants to R-134a [8].

Refrigerant	Classification
R-134a	A1 (Non-toxic and non-flammable)
R-152a	A2 (Non-toxic and low flammability)
R-1234yf	A2L (Non-toxic and low flammable)
R-290	A3 (Non-toxic and flammable)
R-410A	A1 (Non-toxic and non-flammable)
R-32	A2L (Non-toxic and low flammable)

3. Results and Discussion

Firstly, it is verified that the implemented model does not return values that indicate temperature crossings in the heat exchangers, and that the heat exchange zones can be verified in the temperature profile. The criterion used for model adjustment is a maximum deviation of ±10% between the model results and the experimental values.

Figure **5** shows the temperature profiles for the heat exchangers, where the three zones of the condenser and two for the evaporator are identified.

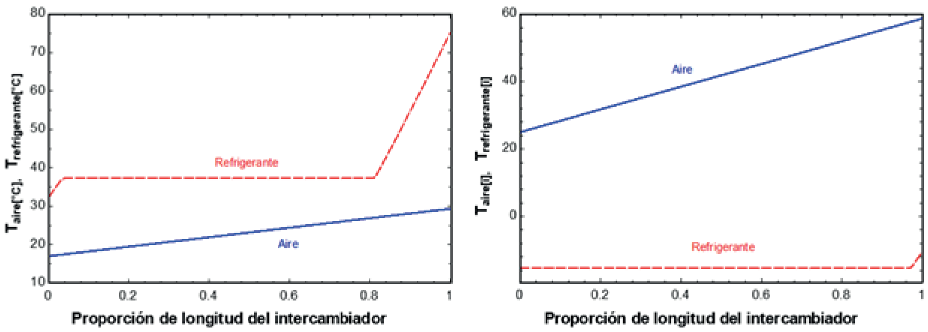

Figura 5

a) Condenser temperature profile. b) Evaporator temperature profile.

3.1. Thermodynamic model validation

Figure **6** shows the validation results of the thermodynamic model. In this figure, the data obtained from the model were contrasted with the experimental data. The model was validated with R2 correlation coefficients of 0.9742 for the COP, 0.8671 for cooling capacity, 0.9842 for compressor work, and 0.8998 for the air temperature at the evaporator outlet. These coefficients have a close approximation to unity. Therefore, the proposed model appropriately represents the thermodynamic and heat transfer phenomena for an automotive air conditioning system. The maximum deviation of the variables does not exceed ±10%.

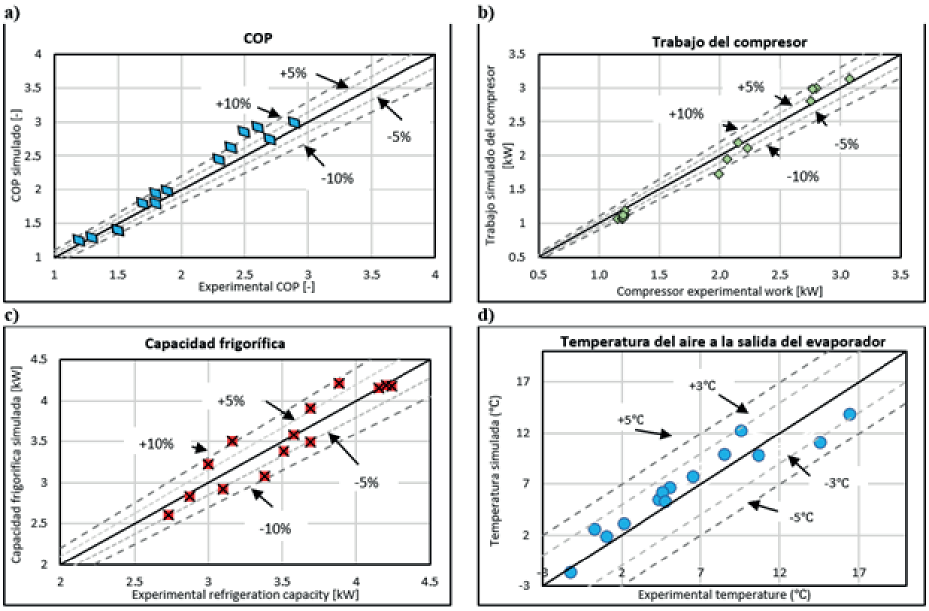

Figura 6

a) Validation of the COP, b) Validation of the compressor work, c) Validation of the refrigeration capacity, d) Validation of the air temperature at the evaporator outlet.

Table 5 shows the statistical analysis for the validation of the model variables. The root mean square error (RMSE), the maximum deviation, and the correlation coefficient R^2 are presented.

3.2. COP comparison, refrigeration capacity, and compressor consumption for various refrigerants.

To present the comparative results, a nominal operating point is defined that represents the intermediate of the system's operating conditions (nominal point): compressor rotation speed of 1600 rpm, ambient temperature of 17 °C, air temperature inside the cabin

Tabla 5

Statistical analysis of validated variables.

Statistical analysis	COP [-]	W_{comp} $[kW]$	Q_{ev} $[kW]$	$T_{air,out,ev}$ [℃]
RMSE	0,157	0,128	0,192	1,728
Maximum deviation	0,321	0,261	0,344	3,53
R^2	0,9742	0,9842	0,8671	0,8998

of 35 °C, air speed at the condenser inlet of 2,283 m/s, and volumetric air flow at the evaporator inlet of 0.1 m3/s. Figures 7, 8, 9, and 10 show the COP comparison, refrigeration capacity, compressor work, and compressor discharge temperature, respectively.

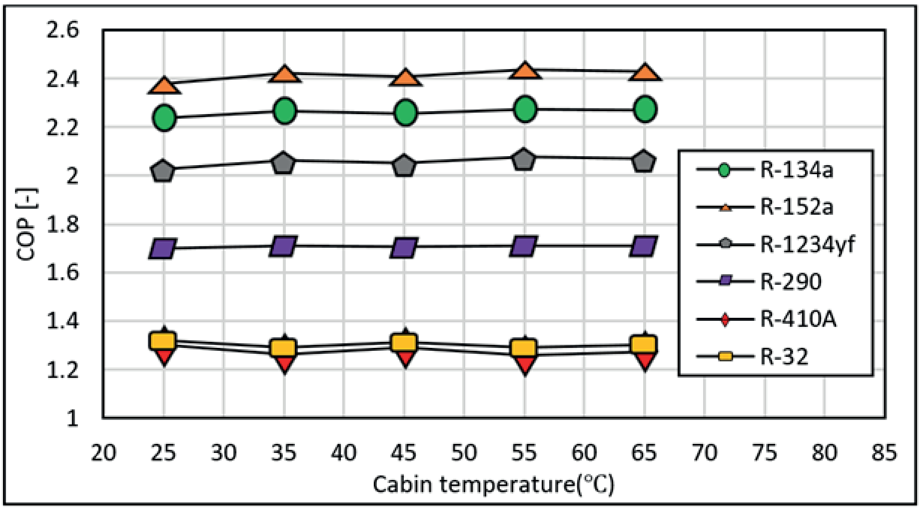

Figura 7

COP vs. cabin temperature working at the nominal point for various refrigerants.

Regarding the COP, compared to the refrigerant R-134a, R-152a presents an increase of 7%. The refrigeration capacity has decreased by 1% since the compressor work has decreased by 7%. Therefore, this refrigerant has the potential to be a possible substitute, using the same system components as R-134a. Consequently, by having a similar weight and dimension throughout the air conditioning system, an increase in the power demand of the internal combustion engine will not be needed; therefore, fuel consumption will not be affected. It must be considered that the application of this refrigerant will be limited by its flammability, which requires an in-depth study to optimize the refrigerant charge.

The results for R-1234yf show a decrease in COP of 9%, and the cooling capacity is reduced by 2%. However, the work of the compressor increases by 7% compared to R-134a. If R-134a is replaced with R-1234yf, a component called an internal heat exchanger

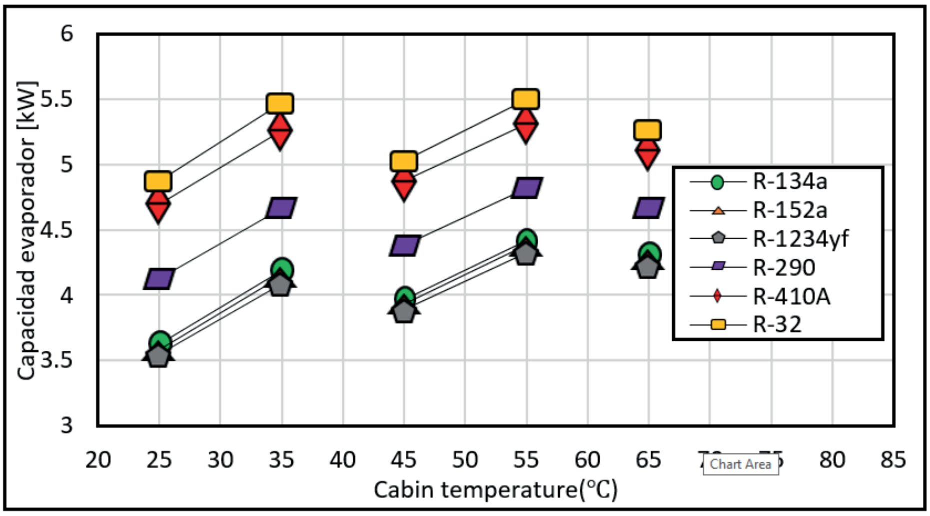

Figura 8

Evaporator capacity vs. cabin temperature working at the nominal point for various refrigerants.

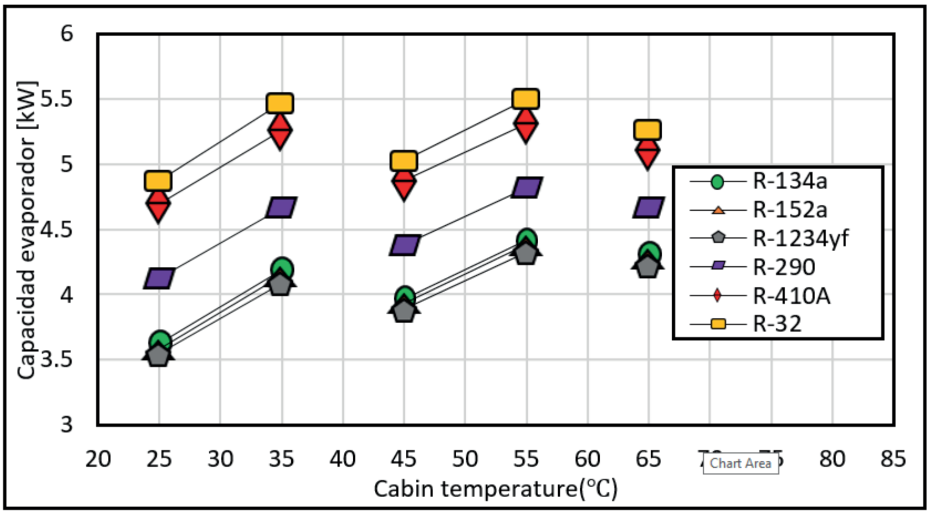

Figura 9

Compressor energy consumption vs. air temperature at the evaporator inlet working at the nominal point for various refrigerants.

(IHX) can be added, which produces a subcooling effect and helps compensate for the low cooling capacity of R-1234yf. Another alternative is to resize the system by adjusting the heat exchange areas in the evaporator and condenser. Finally, increasing the size of the compressor could offset the cooling capacity of this refrigerant.

Taking into consideration the work of the compressor, R-290, R-410A, and R-32 refrigerants have lower COP values than R-134a for use in automotive air conditioning systems. Consequently, the internal combustion engine presents a loss of power and

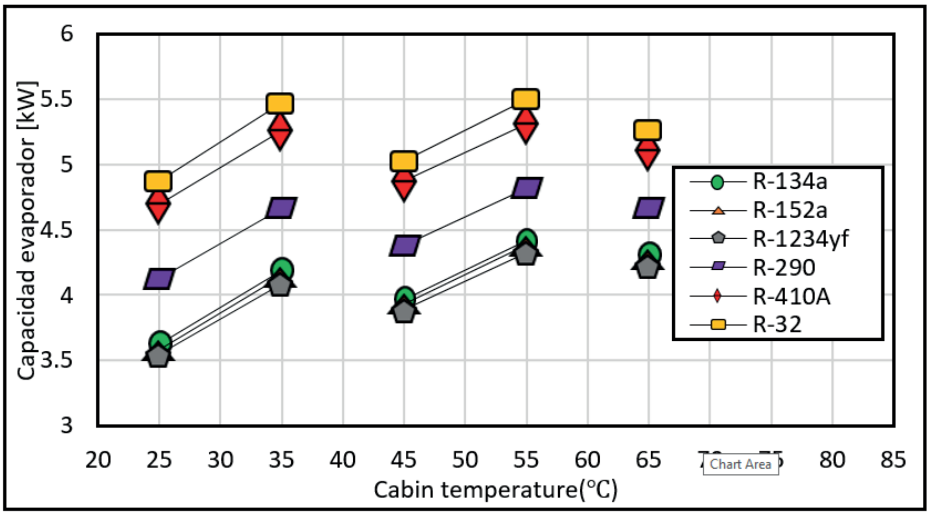

Figura 10

Refrigerant discharge temperature vs. air temperature at the evaporator inlet working at the nominal point for various refrigerants.

greater fuel consumption due to the increase in compressor work. This increase in compressor work is 47% for the R-290, 125% for the R-410A, and 129% for the R-32, compared to the R-134a.

With respect to the discharge temperature, the R-32 refrigerant has the highest temperature, reaching values at which the integrity of the compressor's lubricating oil is compromised. A similar case exists with the refrigerant R-410A, which suggests that these refrigerants would not work adequately for these operating conditions in a vehicle.

4. Conclusions

Based on the energy simulation, it is determined that the R-152a refrigerant has the best COP. Considering a direct replacement of R-134a in automotive air conditioning systems, using the current components of the system such as the compressor, condenser, evaporator, and expansion device. Furthermore, it is important to emphasize that the working pressures are lower when working with this refrigerant, which makes it possible to use the same system pipes.

If a replacement for R-1234yf is required, the addition of a heat exchanger is required to compensate for the low cooling capacity. This refrigerant turns out to be a good substitute since it has an ozone destruction potential of 0 and a global warming potential of 4. Indicating that this refrigerant remains only 11 days in the atmosphere compared to the 14 years that R-134a remains.

Regarding refrigerants R-290, R-410A, and R-32, their performance results are limited for use in automotive air conditioning systems due to their excessive demand on the compressor's work. However, these refrigerants are usually used in domestic and commercial refrigeration systems. In the case of R-290, being a natural refrigerant, it is usually used in small refrigeration systems such as domestic freezers, refrigerators, and heat pumps.

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