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ANALYSIS OF THE EFFECT OF PRESSURE DROP IN EJECTORS ON THE PERFORMANCE OF VAPOR COMPRESSION REFRIGERATION SYSTEMS

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Abstract. *The ejector is a device known to operate minimizing the energy losses caused by the expansion process in vapor compression refrigeration systems, thus resulting in an increase in the coefficient of performance (COP). The pressure drop in the suction nozzle (ΔP), condensation and evaporation temperature are parameters that directly influence the performance of these systems. However, although there is an optimal ΔP that makes the COP of a refrigeration system reach its maximum value for each operating condition, to determine it is necessary to perform a search within a wide range of ΔP or through the use of optimization methods, which can be exhausting. Thus, this article presents an alternative that seeks to facilitate the determination of ΔP that maximizes the COP for different refrigerants and operating conditions (condensation and evaporation temperatures), by specifying an equivalent temperature difference* ΔT_{eq} *determined by the saturation pressure. Thus, the mathematical modeling is established and from the simulations carried out in the EES software, with the refrigerants R134a, R290 and R1234ze(E), it is possible to conclude that the performance of the system improves with the increase of the evaporation temperature and with the decrease of the condensation temperature. Furthermore, the results showed that a value of 0.5K for* ΔT_{eq} *is ideal to meet all operating ranges and evaluated fluids.*

Keywords: *Ejector,Refrigeration,Pressure Drop*

1. INTRODUCTION

The need to save energy and preserve the environment are frequent topics of discussion around the world. Refrigeration systems and air conditioning are responsible for around 7.8% of greenhouse gas emissions globally, and these emissions can occur from leaking refrigerants or associated with the generation and transmission of electricity, according to Cabello et al. (2022).

Therefore, many researchers have studied new refrigerants in order to replace those that degrade the ozone layer and intensify global warming, such as: chlorofluorocarbons (CFC), hydrochlorofluorocarbons (HCFC) and hydrofluorocarbons (HFC). In this respect, natural refrigerants, hydrocarbons (HC) and hydrofluorolefins (HFO) have become options and are gaining importance in the Brazilian market, especially in the commercial refrigeration market (Ubukata, 2015) and (Makhnatch et al., 2019).

In recent studies, it is possible to note the search for alternative refrigerants with good thermodynamic properties and environmentally safe. Ojeda (2021) evaluated in an experimental mode, refrigerants that can replace HFC R134a in cascade refrigeration systems. The results showed that the HFO R1234yf proved to be the most suitable in terms of thermal efficiency and environmental safety for this replacement. Śmierciew et al. (2017) concluded, in their research, HFO R1234ze(E) is a promising fluid for applications in systems that have the ejector in their configuration.

In terms of energy consumption, vapor compression refrigeration cycles (VCRC) stand out for their high consumption. Thus, Tsimpoukis et al. (2021) and Sanaye et al. (2019) affirmed that the development of technologies and strategies that reduce the use of energy in these systems is of vital importance, since the reduction of electricity consumption implies a reduction in the use of fossil fuels for this purpose and in the indirect emissions (associated with the generation of electric energy) which resulted from this process.

A technology that has been the subject of several researches, and that was invented by Charles Parsons at the beginning of the twentieth century (Ojeda et al., 2021), is the ejector. This device can help reduce the work required by the compressor in refrigeration systems, becoming a promising alternative for reducing energy consumption. In addition, it is a device with no moving parts with low operating costs and has the ability to receive a two-phase flow without causing erosion damage. The ejector, whose schematic view can be seen in Fig. 1, has four main components, namely the converging-diverging nozzle, the suction chamber, the mixing chamber and the diffuser (Tashtoush et al., 2019) and (Besagni et al., 2016).

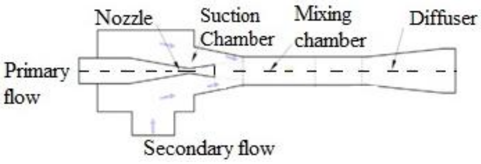

Figure 1. Schematic view of the structure of an ejector.

Ejectors can be used as expansion devices in modified vapor compression refrigeration cycles called ejector expansion refrigeration cycle (EERS). In these systems, the expansion or throttling process leads to a dissipation of energy from the refrigerant fluid, and it is at this point that the ejector acts, helping to recover the losses that occurred by the process, which leads to an increase in the coefficient of performance (COP) (Li et al., 2014) and (Tashtoush et al., 2019). These devices have been used in a wide variety of applications, especially in refrigeration systems. In this regard, Aghazadeh Dokandari et al. (2014) compared a cascade refrigeration cycle with ejector using CO_2 and NH_3 with a cascade system without ejector, and concluded that the first had its COP increased by 7% in relation to the second. Bai et al. (2017) carried out a simulated study with a double-temperature transcritical CO_2 refrigeration cycle with two cascaded ejectors, and also concluded that when using a cycle with two ejectors, efficiency can increase by up to 25.5% compared to systems with only one ejector and up to 28% with a cycle without ejectors.

The experimental study by Ersoy; Bilir Sag (2014) compared the performance of an EERS and VCRC under identical working conditions, such as: evaporation temperature of 10 °C and condensation temperature ranging between 50 and 60 °C. The authors found that the COP of one EERS can be 6.2 to 14.5% higher than the other cycle. Lawrence; Elbel (2014) also experimentally compared the aforementioned cycles, and obtained as a result, a 6% increase in the performance of the EERS in relation to the VCRC.

Several parameters influence the performance of refrigeration cycles. Among them, it is possible can highlight the condensation temperature (T_c), the evaporation temperature (T_e) and the pressure drop in the suction nozzle (ΔP), a common parameter in the mathematical modeling of refrigeration systems that use a two-phase ejector, mainly the EERS, which is given by subtracting the evaporator outlet pressure to the suction nozzle inlet pressure.

Rostamnejad; Zare (2019) proposed a new configuration of an EERS, in which a booster compressor is added to the system and six refrigerants are used for simulation. The COP of the system was analyzed in relation to the ΔP that varied in a range from 0.5 to 50 kPa, thus, the authors noticed that the maximum value of the coefficient of performance (6.26) is reached for an optimal ΔP of 7 kPa for fluid R1234ze under working conditions with T_e = 5 °C and T_c = 40 °C. Another point observed is that as the evaporation temperature increases and condensation decreases, the COP will be higher.

A similar discussion was conducted by Ersoy; Bilir (2012) about ΔP and its influence on COP, but applied to a transcritical EERS system using CO_2. To analyze these parameters, the authors assumed fixed values of T_e and T_c and varied the ΔP between 19 kPa and 2,320 kPa, with optimal results of 3.378 performance and approximately 500 kPa of pressure drop. They also concluded that for each operating condition there is an optimal pressure drop to reach the maximum COP of the system.

It is possible to observe that the procedure to determine the optimal pressure drop at the suction nozzle (ΔP) could be reduced to the search for a maximum performance of the refrigeration system for a given pressure drop range, that is, generally the optimal value of this parameter is obtained within a wide range determined by the authors as a function of the operating conditions of the system.

An alternative that seeks to facilitate the determination of the ΔP that maximizes the COP for different refrigerants and operating conditions (condensation and evaporation temperatures) is the specification of an equivalent temperature difference (ΔT_{eq}) and, thus, through the saturation pressure corresponding to this temperature difference, the ΔP can be determined (Lawrence, 2012) and (Atmaca et al., 2019).

Therefore, the objective of the present study is to evaluate, through the mathematical modeling of an EERS, the use of an equivalent temperature difference to determine the pressure drop in the ejector suction nozzle that maximizes the COP of the system, considering different refrigerants and different condensation and evaporation temperatures.

2. METHODOLOGY

A vapor compression refrigeration system with an ejector (used to recover losses incurred in the expansion process), based on the model proposed by Li et al. (2014) was implemented in the Engineering Equation Solver (EES) software. The main parameters used to simulate the cycle are shown in Tab. 1 and the schematic view of the system is represented by Fig. 2.

The first step was the determination of the thermal cycle. Then, thermodynamic analyzes were performed following the premises proposed by Li et al. (2014), and thus, the mathematical model for the EERS can be established considering the equations of conservation of mass, energy and momentum. After implementing the model, the equations for determining the pressure drop in suction nozzle (ΔP) as a function of ΔT_{eq}, obtained through the saturation pressure, are added to the algorithm and the simulation is performed using the selected fluids R134a, R290 and R1234ze(E).

Table 1. Simulation parameters.

Parameters	**Values [°C]**
Condensing Temperature	30; 40; 50
Evaporation Temperature	-8; -5; 0; 5 e 10

2.1 Operating principles of the proposed cycle

The vapor compression refrigeration cycle is widely studied in the literature due to its range of applications in refrigeration systems and air conditioning. In this context, ejectors, which are introduced in this system with the aim of reducing the losses generated by the expansion valve and increasing the COP, have been the subject of several studies in recent years (Tashtoush et al., 2019) and (Besagni et al., 2016). With this device, the cycle is now called ejector-expansion refrigeration system (EERS) and its most usual configuration, based on Li et al. (2014), is shown in Fig. 2.

In general, the operating principle of the ejector are described as follows: The primary flow from the condenser expands in the ejector and accelerates to supersonic velocity through the converging-diverging nozzle as its pressure decreases. This induces the entry of a secondary flow from the evaporator, which when entering the ejector, increases in velocity. The pressure at the suction nozzle outlet is equal to that of the constant area mixing chamber. When they reach the same velocity at the entrance of the chamber, the flows mix at constant pressure, and entering the diffuser its pressure is increased while the velocity reduces and after that, the flow leaves the ejector towards the liquid separator.

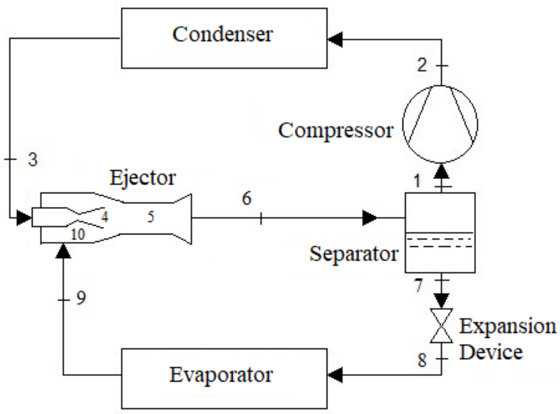

Figure 2. Ejector-expansion refrigeration system.

2.2 Mathematical model

To establish the mathematical model, some assumptions about the studied system are made below in order to simplify the analysis:

1. Steady state;
2. The fluids leaving the evaporator and condenser are saturated;
3. The vapor and liquid lines leaving the separator are saturated;
4. Pressure drop in lines and heat exchangers is negligible;
5. The expansion process is considered isoenthalpic;
6. The heat transfers with the environment are ignored, except for the condenser;

7. The velocities at the inlet and outlet of the ejector are negligible;
8. The primary and secondary streams only mix when they acquire the same pressure, that stays the same throughout mixing;
9. It is considered one-dimensional homogeneous equilibrium flow at all points of the ejector.

Initially the working fluid is selected and some operating conditions such as condensation (T_c) and evaporation (T_e) temperatures are given according to Tab. 1. Thus, the analyzes begin by calculating the saturation properties at the evaporator outlet given at point 9, and the condenser at point 3. In Eqs. (1) to (6), *p, h, s* are the pressure, specific enthalpy and specific entropy, respectively.

$$p_3 = p(T_c, x = 0) \tag{1}$$

$$h_3 = h(T_c, x = 0) \tag{2}$$

$$s_3 = s(T_c, x = 0) \tag{3}$$

$$p_9 = p(T_e, x = 1) \tag{4}$$

$$h_9 = h(T_e, x = 1) \tag{5}$$

$$s_9 = s(T_e, x = 1) \tag{6}$$

For the ejector, an initial value of entrainment ratio (ω) is estimated, this parameter is defined as the ratio between the mass flow rates of the secondary flow to the primary flow, and the efficiencies of the primary nozzle (η_{mn}), suction nozzle (η_{sn}), mixing section (η_{ms}), and diffuser (η_d) are given. Besides the mentioned parameters, the pressure drop in the suction nozzle (ΔP) and the primary and secondary streams properties during the flow in the ejector are presented by Eqs. (7) to (22). The equations referring to the outlet of the primary nozzle are comprised between Eq. (7) and Eq. (11), at the outlet of the suction nozzle are between Eqs. (12) and (16), to the mixing section are Eqs. (17) to (19) and referring to the diffuser are Eqs. (20) to (22). Where *A* and *c* refer, respectively, to the area and velocity at each point shown in the diagram in Fig. 2.

$$p_4 = p_9 - \Delta P \tag{7}$$

$$h_{4,s} = h\left(p_4, s_3\right) \tag{8}$$

$$h_4 = h_3 - \eta_{mn}\left(h_3 - h_{4,s}\right) \tag{9}$$

$$c_4 = \sqrt{2\left(h_3 - h_4\right)} \tag{10}$$

$$A_4 = \frac{1}{\left(1+\omega\right) p_4 c_4} \tag{11}$$

$$p_{10} = p_9 - \Delta P \tag{12}$$

$$h_{10,s} = h\left(p_{10}, s_9\right) \tag{13}$$

$$h_{10} = h_9 - \eta_{sn}\left(h_9 - h_{10s}\right) \tag{14}$$

$$c_{10} = \sqrt{2\left(h_9 - h_{10}\right)} \tag{15}$$

$$A_{10} = \frac{\omega}{\left(1+\omega\right) p_{10} c_{10}} \tag{16}$$

$$c_5 = \sqrt{\eta_{ms}\left(\frac{1}{1+\omega}c_4 + \frac{\omega}{1+\omega}c_{10}\right)} \tag{17}$$

$$h_5 = \frac{1}{1+\omega}\left(h_4 + \frac{c_4^2}{2}\right) + \frac{\omega}{1+\omega}\left(h_{10} + \frac{c_{10}^2}{2}\right) - \frac{c_5^2}{2} \tag{18}$$

$$s_5 = s\left(p_5, h_5\right) \tag{19}$$

$$h_6 = h_5 + \frac{c_5^2}{2} \tag{20}$$

$$p_6 = p\left(h_{6s}, s_5\right) \tag{21}$$

$$x_6 = \frac{1}{1+\omega} \tag{22}$$

Equations (23) and (24) are introduced at this point, the first refers to the equivalent temperature difference (ΔT_{eq}), which is the difference between the evaporation temperature, here called T_9, and the suction nozzle outlet temperature (T_k). The second equation concerns the calculation of the pressure drop that is evaluated as a function of the saturation pressure corresponding to ΔT_{eq}.

$$T_k = T_9 - \Delta T_{eq} \tag{23}$$

$$\Delta P = p\left(T_9, x=1\right) - p\left(T_k, x=1\right) \tag{24}$$

After calculating the parameters related to the ejector, the other components of the cycle also have their properties determined based on the equations below. For the liquid separator, at the saturated vapor outlet:

$$h_1 = h\left(p_6, x=1\right) \tag{25}$$

$$p_1 = p_6 \tag{26}$$

The flow passes through the compressor, which has the following properties and efficiency:

$$p_2 = p_3 \tag{27}$$

$$h_{2,s} = h\left(p_2, s_1\right) \tag{28}$$

$$h_2 = h_1 + \frac{h_{2,s} - h_1}{\eta_c} \tag{29}$$

$$\eta_c = 0{,}874 - 0{,}0135\left(\frac{p_3}{p_9}\right) \tag{30}$$

At the saturated liquid outlet from the liquid separator and to the expansion valve and evaporator:

$$p_7 = p_6 \tag{31}$$

$$h_7 = h\left(p_6, x=0\right) \tag{32}$$

$$h_7 = h_8 \tag{33}$$

$$p_8 = p_9 \tag{34}$$

The compression work (W_c), cooling capacity (q_e) and the system performance coefficient (COP) are presented in sequence, as follows:

$$W_c = \frac{h_2 - h_1}{1+\omega} \tag{35}$$

$$q_e = \frac{(h_9 - h_8)\,\omega}{1+\omega} \tag{36}$$

$$COP = \frac{q_e}{W_c} \tag{37}$$

3. RESULTS AND DISCUSSIONS

Based on the mathematical model detailed in Section 2, the simulation is performed for an ejector-expansion refrigeration system (EERS), varying the values of condensation (T_c) and evaporation (T_e) temperatures according to Tab. 1. In the analysis, the efficiencies at the primary nozzle (η_{mn}), suction nozzle (η_{sn}), mixing section (η_{ms}), and diffuser (η_d) are respectively 0.90; 0.90; 0.85 and 0.80 according to Atmaca et al. (2019). Furthermore, as T_e is varied, T_c remains constant in the calculation of the coefficient of performance (COP), ΔP and other parameters.

Figures 3, 4 and 5 shows the variation of ΔT_{eq} in relation to the COP obtained through optimization by quadratic approximations, for the condensation and evaporation temperature conditions selected. In this sense, each point of the curves of the three fluids presented represents the variation of T_e between -8°C and 10°C, decreasing in the graphs. Thus, when observing the figures mentioned, it is clear that by keeping the condensation temperatures fixed, the COP tends to decrease as T_e decreases.

In Figure 3, T_c is fixed at 30°C and when analyzing R1234ze(E) it is noticed that the equivalent temperature difference increases from 0.13 °C to approximately 0.33 °C, and it is at this value point maximum of ΔT_{eq}, in which T_e = -8°C, that the coefficient of performance obtained its minimum value of about 4.93. At the same evaporation temperature, fluids R134a and R290 presented COP values appoximately 4.62 and 4.58, respectively. It is also noted that the fluid R1234ze(E) has the highest COP among the fluids analyzed at all points of the curves presented, reaching a maximum value of approximately 10.30, as show in Fig. 3.

The same trend is observed in Fig. 4, in which the three studied refrigerants presents COP values in decreasing curves, confirming that this parameter has its values decreased with the reduction of T_e. Thus, it can be seen that the values of ΔT_{eq} , between the extremes of the curves, increased by about 80% for refrigerants R290 and R134a and 77% for R1234ze(E) according to the determined temperature parameters.

When T_c is set at 50°C (Fig. 5), the maximum coefficiente of performance values are lower in relation to the other graphs analyzed, such a situation was already expected, since it is known that when the condensation temperature increases, the COP of refrigeration systems decreases according to Rostamnejad; Zare, 2019.

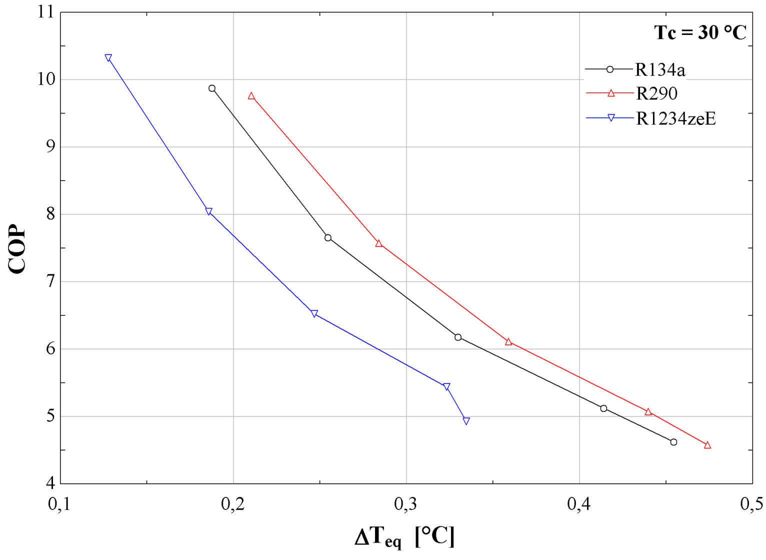

Figure 3. COP versus ΔT_{eq} optimized for T_c = 30°C and variable T_e.

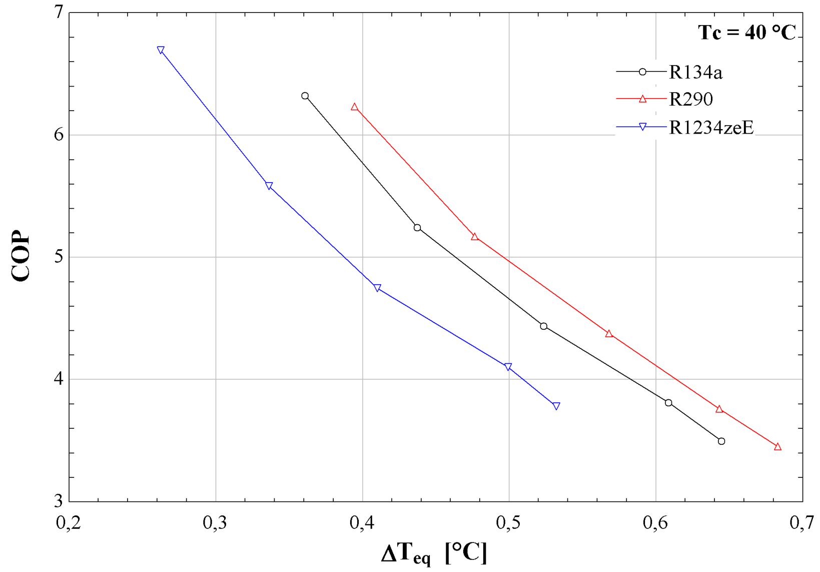

Figure 4. COP versus ΔT_{eq} optimized for T_c = 40°C and variable T_e.

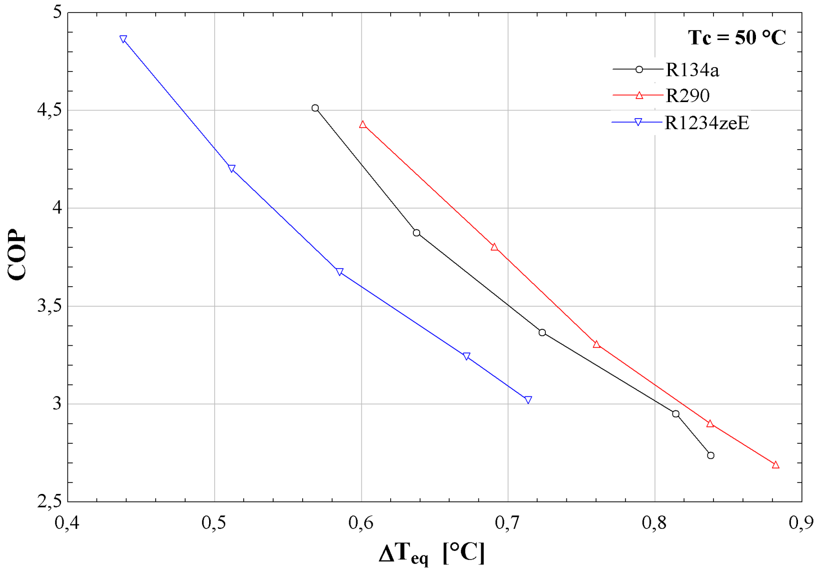

Figure 5. COP versus ΔT_{eq} optimized for T_c = 50°C and variable T_e.

From Figs. 3, 4 and 5, it is possible to notice that the optimized ΔT_{eq} varies between approximately 0.13 °C and 0.88 °C. Despite this variation, it is observed that a single ΔT_{eq} that maximizes the COP and that meets all operating conditions, as well as all studied refrigerants, can be adopted without loss of precision. The value adopted for this parameter is 0.5 K, obtained by calculating the average of ΔT_{eq} values. Thus, a comparison about the optimized COP and the COP referring to the fixed value of ΔTeq = 0.5 K is presented in Tab. 2 for the refrigerants R290, R134a and R1234ze(E). Through Eq. (38), the percentage errors between the performance coefficients found are calculated and presented in Tab. 3.

$$error(\%) = \frac{COP - COP_{\Delta Teq0.5}}{COP} x100 \tag{38}$$

In analyzing Tab. 3, it is noted that the percentage error did not reach the value of 1%, with the maximum found being 0.5029% and the maximum average value of 0.0814%, both values referring to the fluid R1234ze(E). Moreover, in Tab. 2, it is possible to observe that the combination of T_c = 40 °C and T_e = 0 °C for R134a presented the same maximum values of COP, both for the optimized and for ΔT_{eq} = 0.5 K.

The same behavior is observed for the refrigerants R290 and R1234ze(E), which in some points presented identical COP values, as in T_c = 40 °C and T_e = 5 °C referring to R290 and T_c = 40 °C and T_e = -5 °C for hydrofluorolefin. This indicates that the error is 0% when calculating the coefficient of performance of the two proposed ways, both in the points mentioned and in those highlighted in bold in Tab.2. Therefore, it can be seen that the value of 0.5 K adopted for the equivalent temperature difference (ΔT_{eq}) to find the maximum COP in a simplified way, was satisfactory and had an accuracy above 99.0%, since the average percent error for all studied temperature ranges does not reach 0.1% according to Tab.3. This shows that there is no need to use an optimal ΔT_{eq} for each studied point, as this would take more effort with minimal or no improvement in the COP value.

Table 2. Comparison between the optimized COP and COP values for ΔT_{eq} = 0.5K for the tested refrigerants.

		R290			R134a			R1234ze(E)		
T_c (°C)	**T_e (°C)**	**ΔT_{eq} (K)**	**COP**	**COP $\Delta T_{eq=0.5K}$**	**ΔT_{eq} (K)**	**COP**	**COP $\Delta T_{eq=0.5K}$**	**ΔT_{eq} (K)**	**COP**	**COP $\Delta T_{eq=0.5K}$**
30	-8	0.4738	**4.5814**	**4.5814**	0.4545	4.6259	4.6258	0.3345	4.9269	4.9255
	-5	0.4395	5.0735	5.0732	0.4139	5.1243	5.1236	0.3232	5.4393	5.4363
	0	0.3589	6.1141	6.1118	0.3298	6.1782	6.1748	0.2466	6.5224	6.5139
	5	0.2840	7.5732	7.5648	0.2546	7.6559	7.6449	0.1856	8.0407	8.0196
	10	0.2103	9.7650	9.7389	0.1875	9.8756	9.8436	0.1277	10.3207	10.2688
40	-8	0.6829	3.4536	3.4525	0.6446	3.4983	3.4976	0.5321	**3.7797**	**3.7797**
	-5	0.6430	3.7610	3.7603	0.6085	3.8108	3.8104	0.4991	**4.1020**	**4.1020**
	0	0.5679	4.3772	4.3771	0.5233	**4.4372**	**4.4372**	0.4101	4.7477	4.7472
	5	0.4764	**5.1713**	**5.1713**	0.4373	5.2444	5.2441	0.3361	5.5795	5.5771
	10	0.3945	6.2323	6.2310	0.3608	6.3229	6.3206	0.2623	6.6905	6.6830
50	-8	0.8822	2.6910	2.6881	0.8378	2.7384	2.7360	0.7136	3.0198	3.0187
	-5	0.8374	2.9006	2.8980	0.8140	2.9524	2.9503	0.6717	3.2422	3,2414
	0	0.7602	3.3069	3.3049	0.7232	3.3673	3.3659	0.5852	3.6731	3.6728
	5	0.6907	3.8050	3.8038	0.6377	3.8761	3.8754	0.5117	**4.2012**	**4.2012**
	10	0.6008	4.4295	4.4290	0.5684	4.5139	4.5137	0.4378	4.8628	4.8625

Table 3. Percentage errors between the optimized COP and the COP for ΔTeq = 0.5K for the tested refrigerants.

	Maximum error (%)	Average error (%)
R290	0.2673	0.0531
R134a	0.3240	0.0556
R1234ze(E)	0.5029	0.0814

When comparing the numerical results obtained in this study with those of Atmaca et al. (2019) under the same operating conditions in an EERS, it was noticed that the coefficient of performance values are similar in the two studies. Thus, when analyzing the point at which T_c= 40°C and T_e= 5°C, which according to the authors are usual values in applications of refrigeration systems, it was noticed that the optimal COP obtained in the mentioned study reached the maximum value of approximately 5.59 for the refrigerants R134a and R1234ze(E), while for the present research, the maximum values of this parameter were approximately 5.24 and 5.58, respectively, for the mentioned refrigerants. Therefore, the maximum error between the performance values was 6.21% for R134a.

Furthermore, in the same study by Atmaca et al. (2019) in which the value of ΔT_{eq} = 5 K was used to determine the pressure drop in the suction nozzle, it was noted that the ΔP curve remained constant when setting the evaporation temperature at 5 °C and varying the condensation temperature between 40 °C and 65 °C, this fact was also observed in the present study, when analyzed under the same parameters and refrigerants tested.

4. CONCLUSIONS

This study evaluated, through the mathematical modeling of an ejector vapor compression refrigeration system (EERS), the use of an equivalent temperature difference to determine the pressure drop at the ejector suction nozzle (ΔP) that maximizes the system COP. The selected refrigerants were R134a, R290 and R1234ze(E) and the condensing temperature was varied between 30 °C and 50 °C and the evaporation temperature between -8 °C and 10 °C.

The results showed that as the condensation temperature increased, the COP of the system drops for all fluids worked, reaching a minimum value of approximately 2.69, which refers to the refrigerant R290, when T_c = 50 °C . In contrast, R1234ze(E) showed higher COP values than the other analyzed refrigerants, thus, it shows a promising refrigerant for application in refrigeration systems that contain an ejector, in terms of thermal performance.

In addition, although there is a variation of ΔT_{eq} that maximizes the COP for the fluids worked, it is observed that ΔT_{eq} = 0.5 K satisfactorily met all the operating conditions studied, with maximum errors smaller than 0.6% in relation to the COP values found in the two ways discussed in this work.

Therefore, it is concluded that the use of a ΔT_{eq} to determine the pressure drop at the ejector suction nozzle (ΔP) proved to be an effective procedure to maximize the COP of an EERS, since in this way it is possible to meet, for same temperature range, some refrigerant fluids that work under different pressure ranges without having to adjust this parameter.

5. ACKNOWLEDGEMENTS

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