Revista Facultad de Ingeniería, Universidad de Antioquia, No.113, pp. 19-27, Oct-Dec 2024

Performance of ejector refrigeration cycle using a steady flow model

Desempeño de un ciclo de refrigeración con eyector usando un modelo de flujo estable

Leonardo Pacheco-Sandoval[1,2*], Carlos Alirio Díaz-González[1,2], Germán David Acebedo-Roncancio[1,2], Miguel Ángel Rodríguez-Camacho[1,2]

[1]Ingeniería en Energía, Universidad Autónoma de Bucaramanga. Avenida 42 48-11 Bucaramanga-Colombia.
[2]Grupo de investigación en Recursos Energía y Sostenibilidad -GIRES, Universidad Autónoma de Bucaramanga. Avenida 42 48-11 Bucaramanga-Colombia.

CITE THIS ARTICLE AS:

L. Pacheco-Sandoval, C. A. Díaz-González, G. D. Acebedo-Roncancio and M. Á. Rodríguez-Camacho. "Performance of ejector refrigeration cycle using a steady flow model", *Revista Facultad de Ingeniería Universidad de Antioquia*, no. 113, pp. 19-27, Oct-Dec 2024. [Online]. Available: https://www.doi.org/10.17533/udea.redin.20240305

ARTICLE INFO:

Received: September 25, 2023
Accepted: March 11, 2024
Available online: March 11, 2024

KEYWORDS:

Cooling Systems; Waste heat recovery

Sistema de Enfriamiento; Recuperación Calorífica

ABSTRACT: A study of an alternative ejector cooling system is proposed in this paper. A mathematical model without irreversibility is developed and simulated. The numerical results are validated with literature data for two working fluids (R134a and R152a). The performance and behavior of the system using conventional refrigerants (HFC's, HC's) and unconventional (R717, R718) is obtained; the process uses 100 kW of heat available in a waste stream. Finally, the paper explores the reduction of problems associated with the use of conventional refrigerants by introducing the use of R718 (water) as the working fluid of the proposed system.

RESUMEN: Un estudio de un sistema de refrigeración alternativo con eyector es propuesto en este artículo. Se desarrolla y simula un modelo matemático sin irreversibilidades. Se realiza una validación de los resultados numéricos con datos de la literatura para dos fluidos de trabajo (R134a y R152a). El rendimiento y el comportamiento del sistema usando refrigerantes convencionales (HFC, HC) y no convencionales (R717, R718) es estudiado, el proceso utiliza 100 kW de calor disponible en una corriente de desecho. Finalmente, se explora la reducción de los problemas asociados con el uso de refrigerantes convencionales introduciendo el uso de R718 (agua) como el fluido de trabajo del sistema propuesto

1. Introduction

The constant demand for thermal comfort rapidly increased the use of the cooling system. Reducing the electricity consumption used for refrigeration is an effective strategy for mitigating the constantly growing energy and environmental impact ($\mathrm{CO}_{2\mathrm{eq}}$ and greenhouse gas emission) [1–3].

* Corresponding author: Leonardo Pacheco-Sandoval
E-mail: Lpacheco560@unab.edu.co
ISSN 0120-6230
e-ISSN 2422-2844

Approximately 15% of total electricity consumption is used for refrigeration [2–4]. Research and development (R & D) of thermal energy refrigeration using low-grade heat may considerably decrease in energy consumption. Ejector refrigeration systems (ERS) seem a promising alternative to traditional compressor-based systems due to their reliability, low maintenance needs, and small initial and operational costs. Furthermore, ERS can help reduce greenhouse gas emissions due to the use of primary energy and avoidance of environmentally harmful refrigerants [5–10]. Nevertheless, ejector refrigeration has not been able to position itself in the market due to its low-performance coefficient and severe reduction in

BY NC SA OPEN ACCESS

performance when not operating under design conditions [6, 11, 12].

Many theoretical and experimental works of the ejector refrigeration systems are documented in the literature for various cooling fluids, including R113 [12], R123 [13, 14], R134a [15–17], R141b [14, 18], R152a [13–16, 19], R245fa [14, 18, 20], R290 [14, 16, 17, 19], R600 [14, 17], R600a [16, 18, 19, 21, 22], and ammonia [15, 20, 23].

This work aims to numerically study an ejector refrigeration system (ERS). This paper is organized as follows: Section 2 describes the ERS and its operation. The processes of the ejector, which is considered the principal unit, are detailed. A comprehensive mathematical model, along with a solution process and its comparison with the literature results, are presented in section 3. Finally, a comparative study of performance systems at several refrigerant fluids is presented in section 4. In this work, a comprehensible mathematical model is presented, and a comprehensible solution procedure is developed. The fluids analyzed are HFC´s (R134a, R152a), HC´s (R290, R600a), R717 (NH_3) and R718 (H_2O). The performance of the ejector cooling system is analyzed. The COP behavior is shown, and the maximum Qevap is calculated for all work-fluid. Using R718 as a working fluid provides many advantages: it has a high heat of vaporization, is inexpensive, and has minimal environmental impact; however, the cooling cycle temperature is limited to above 0 °C [24].

2. System description

The alternative ejector refrigeration system (ERS) is a closed system. ERS is composed of six elements: a generator heat exchanger, supersonic ejector, pump, expansion valve, condenser, and evaporator, as shown in Figure 1. The system comprises two circuits: the coolant loop (3-4-5-6) and the power loop (1-2-4-5). The coolant loop has the same functional principles as a conventional cooling system by compression [5–7, 11].

The heat exchanger generator recuperates the remainder of the heat. This thermal energy is absorbed by the working fluid that is in a liquid state in point 1, producing a high temperature, and high-pressure vapor in point 2. The high-pressure vapor (primary fluid) flows through the ejector, where it is accelerated as it passes through the supersonic nozzle. At the nozzle exit, the primary fluid pressure becomes lower than the entry flow pressure. At this point, the inside evaporator flow (point 3, second fluid) is suctioned into the ejector and is mixed with the primary fluid. The fluid mixture emerges from the mixing chamber and is compressed in two steps: firstly, the shock wave (supersonic – subsonic regime). And secondly, a diffuser increases the pressure. The pressure of the emerging flow (point 4) is the condenser pressure. The fluid inside the condenser becomes liquid by rejecting heat from the environment (point 5). A portion of the liquid is pumped to the generator to complete the power loop (point 1). The rest of the refrigerant fluid (point 4) is expanded through a throttling valve (point 6), and a mixture of vapor and liquid enters the evaporator. In the evaporator, all the fluid is transformed to vapor by absorbing heat from the cooled space (point 3) and enters the ejector, which completes the refrigeration cycle [11, 25]

The ejector is the heart of the system. Figure 2 shows a scheme diagram of a pressure and velocity ejector profile. The primary fluid (2) is expanded through the convergent-divergent nozzle in the ejector to produce high-velocity vapor (I). At this point, the inside nozzle flows out with supersonic speed (Match>1) to create a very low-pressure region. Consequently, the high-pressure fluid called secondary fluid (3) can be entrained from the evaporator into the mixing chamber (ii). The fluid mixture emerges from the mixture chamber with supersonic speed (iii). Due to the downstream high pressure of the ejector, at the constant area, a shock wave occurs at the end of the contact area zone, when regime flow is changed to subsonic (Mach<1) (iv): the shock wave occurs before the diffuser (v). This shock wave contributes to a major compression effect. A low contribution of compression effect occurs in the diffuser (iv) [26–28]. The ejector takes advantage of the kinetic energy present in fluids (gases) to be accelerated and decelerated by changes in the cross-sectional area of a pipe. Figure 3 shows a T-s diagram of the ejector refrigeration cycle. The primary fluid (point 1) is expanded in the isentropic process (Figure 2, point (ii)). The process of mixture in the diffuser and pump is also isentropic.

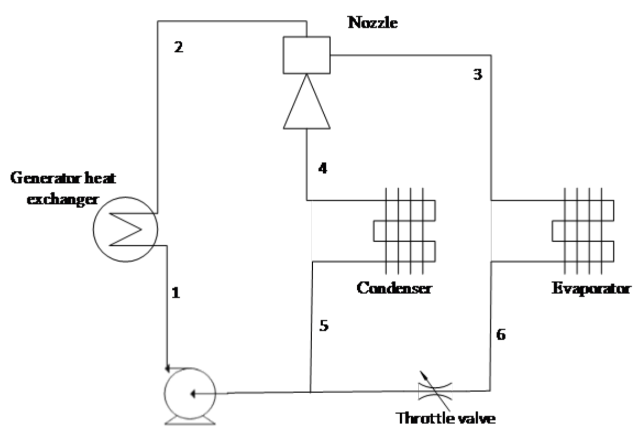

Figure 1 Scheme diagram of an ejector cooling system

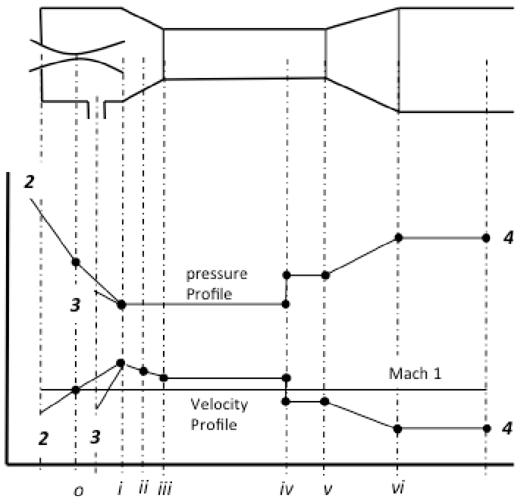

Figure 2 Scheme diagram of pressure and velocity ejector profile

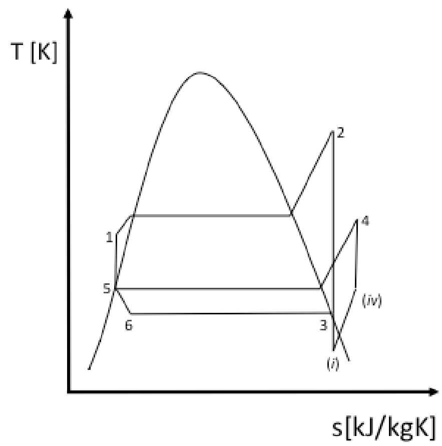

Figure 3 Schematic T-s Diagram of an ejector refrigeration cycle

3. Mathematical model

An ideal model (without irreversibility) under steady conditions is developed. The mathematical model applied the equations of mass and energy conservation. Several assumptions on the physical conditions are made:
a) Flow is axisymmetric and steady,
b) Physical properties are thermo-dependent,
c) adiabatic condition at the ejector,
d) Pressure losses negligible at the generator, condenser, evaporator, and connection tube,
e) Kinetic energy negligible out of the ejector,
f) The sizing equation model is for the nominal charge,
g) Fluid acceleration processes in the nozzle, in the diffuser, and in the pump are adiabatic and reversible, Equations 1-2-3-4. Therefore, they are isentropic processes,

$$s_5 = s_1 \tag{1}$$

$$s_2 = s_3 \tag{2}$$

$$s_3 = s_{(\mathrm{i})} = s_{(ii)} = s_{(iii)} \tag{3}$$

$$s_4 = s_{(\mathrm{v})} = s_{(vi)} \tag{4}$$

h) isobaric processes in the generator, Equation 5; in the condenser, Equation 6, and in the evaporator, Equation 7,

$$P_1 = P_2 = P_{gen} \tag{5}$$

$$P_4 = P_5 = P_{con} \tag{6}$$

$$P_3 = P_6 = P_{eva} \tag{7}$$

i) steam and liquid saturate at condenser exit, Equation 8 and evaporator exit, Equation 9, respectively,

$$x_5 = 0 \tag{8}$$

$$x_3 = 1 \tag{9}$$

The physical model consists of the following set of equations for energy balances applied in the generator Equation 10, the condenser Equation 11, and evaporator Equation 12.

$$Q_{\text{gen}} = \dot{\mathrm{m}}_1 (h_2 - h_1) \tag{10}$$

$$Q_{\text{con}} = \dot{\mathrm{m}}_4 (h_5 - h_4) \tag{11}$$

$$Q_{eva} = \dot{\mathrm{m}}_3 (h_3 - h_6) \tag{12}$$

The kinetic energy contribution changes the thermodynamic state of the primary fluid, Equation 13, and the secondary fluid, Equation 14. The enthalpies at the into, Equation 13 - 14 and exit ejector, Equation 15, are shown in the following equations,

$$h_2 = h_i + 0,5 * V_{2i}^2 \tag{13}$$

$$h_3 = h_i + 0,5 * V_{3i}^2 \tag{14}$$

$$h_4 = h_v + 0,5 * V_v^2 \tag{15}$$

The mixing relation of the primary and secondary fluid emerges from the mixture chamber, Equation 16,

$$(P_i - P_v) A_v = \dot{\mathrm{m}}_2 V_{2i} + \dot{\mathrm{m}}_3 V_{3i} - (\dot{\mathrm{m}}_2 + \dot{\mathrm{m}}_3) V_i \tag{16}$$

Overall material balance and entrainment ratio (w), Equation 17-18, between the primary and the secondary fluid,

$$\dot{\mathrm{m}}_2 + \dot{\mathrm{m}}_3 = \dot{\mathrm{m}}_4 \tag{17}$$

$$w = \frac{\dot{\mathrm{m}}_2}{\dot{\mathrm{m}}_3} \tag{18}$$

An isentropic expansion for the primary and secondary fluid (at the nozzle and suction chamber) is expressed in terms of the Mach number at the nozzle exit of the primary fluid and the secondary fluid, Equations 19 and 20 respectively [5, 25].

$$M_{2i} = \sqrt{\frac{2}{y-1} * \left(\left(\frac{P_2}{P_i}\frac{y-1}{y}\right) - 1\right)} \tag{19}$$

$$M_{3i} = \sqrt{\frac{2}{y-1} * \left(\left(\frac{P_3}{P_i}\frac{y-1}{y}\right) - 1\right)} \tag{20}$$

The evaluation of the critical Mach number of the mixer was carried out for the conditions of the primary and secondary fluids, see Equation 21. The relation between the Mach number and critical Mach number is given for Equation 22. The Mach number after the wave shock is evaluated with Equation 23

$$M^*{}_{iv} = \frac{\left(M^*{}_{2i} + wM^*{}_{3i}\right) * \sqrt{\frac{T_3}{T_2}}}{\sqrt{(1+w)\left(1+w\frac{T_3}{T_2}\right)}} \tag{21}$$

$$M^* = \sqrt{\frac{M^2(y+1)}{M^2(y-1)+2}} \tag{22}$$

$$M_v = \frac{M_{iv}^2 + \frac{2}{y-1}}{\frac{2y}{y-1}M_{iv}{}^2 - 1} \tag{23}$$

The pressure increase in the shock wave is shown in Equation 24, and the pressure increase in the diffuser is shown in Equation 25:

$$\frac{P_i}{P_{iv}} = \frac{1+yM_{iv}^2}{1+yM_v^2} \tag{24}$$

$$\frac{P_{con}}{P_v} = \left(\frac{\eta_{iv}(y-1)}{2}M_v^2 + 5\right)^{\frac{y}{y-1}} \tag{25}$$

Internal areas of the ejector in relation to the conditions of pressure and temperature are given by Equations 26, 27 and 28:

$$A_o = \frac{\dot{m}_1}{P_{gen}}\sqrt{\frac{RT_2}{y*\eta_t} * \left(\frac{y+1}{2}\right)^{\frac{y+1}{y-1}}} \tag{26}$$

$$\frac{A_i}{A_o} = \sqrt{\frac{1}{M_{2i}^2}\left[\frac{2}{y+1}\left(1 + \frac{(y-1)}{2} * M_{2i}^2\right)\right]^{\frac{y+1}{y-1}}} \tag{27}$$

$$\frac{A_o}{A_v} = \frac{P_{con}}{P_{gen}}\left(\frac{1}{(1+w)\left(1+w\frac{T_3}{T_2}\right)}\right)^{\frac{1}{2}} * \left(\frac{\frac{P_{\text{gen}}^{P_{\text{con}}}}{\frac{1}{y}}\left(1-\frac{P_{\text{gen}}^{P_{\text{con}}}}{\frac{y-1}{y}}\right)^{\frac{1}{2}}}{\left(\frac{2}{y+1}\right)^{\frac{1}{y-1}}\left(1-\frac{2}{y+1}\right)^{\frac{1}{2}}}\right) \tag{28}$$

The coefficient performance system (COP) is given by the Equation 29:

$$COP = \frac{Q_{\text{eva}}}{Q_{gen} + W_{\text{pump}}} = \frac{Q_{\text{eva}}}{\dot{m}_1\left(h_2 - h_5\right)} \tag{29}$$

3.1 Solution process model

The numerical simulations were carried out using the Engineering Equation Solver (EES) software. The calculation steps are described below:

Step 1: Specify the refrigerant and operating pressure P_{gen} , P_{con} , P_{eva}

Step 2: Specify the available heat flux in the generator, Q_{gen}

Step 3: Obtain T_5, h_5, s_5 from the P_{con} and $x_5 = 0$

$s_1 = s_5$ and P_{gen} obtain T_1, h_1.

$h_6 = h_5$ and P_{eva} obtain T_6, S_6.

P_{eva} and $X_3 = 1$ obtain T_3, h_3, S_3.

$s_2 = s_3$ and P_{gen} get T_2, h_2.

Step 4: Calculate $\dot{m}_1$ clearing $Q_{\text{gen}} = \dot{m}_1\left(h_2 - h_1\right)$.

Step 5: Calculated $\dot{m}_3 = \frac{Q_{\text{evap}}}{(h_3-h_6)}$ and calculate the entrainment ratio $w = \frac{\dot{m}_1}{\dot{m}_3}$

Step 6: $h_4 = \frac{h_2 - wh_3}{1+w}$ and from P_{con} get T_4, s_4

Step 7: Using Equations 18 – 22, the pressures within the ejector are calculated. The calculating process is iterative assuming $P_{\text{eva}} > P_i$ until an exit pressure from the ejector is equal to the condenser pressure. Finally obtaining P_i y P_v

Step 8: P_i and $i = s_2$ it can be obtained i, h_i, ρ_i

Step 9: Calculate A_o, A_i and A_{iv}.

Step 10: Applying the first law of thermodynamics in the ejector is obtained V_{2i} and V_{3i}

Step 11: From $\left(P_i - P_v\right)A_v = \dot{m}_1 V_{2i} + \dot{m}_3 V_{3i} - \left(\dot{m}_1 + \dot{m}_3\right)V_v$ is calculated V_e.

Step 12: $h_4 = h_v + 0.5 * V_v{}^2$ allows us to find the enthalpy h_v.

Step 13: Since P_v and h_v get s_v.

Step 14: Repeat stage 7 until $s_v = s_4$.

Step 15: Calculate Q_{con}

Step 16: Get representative values of operation as COP and $W_p 4$

3.2 Simulation condition and model validation

Simulations are carried out for the conditions presented by Dahmani *et al.* (2011) [16]. Two cooling fluids (HFC) are used in model validation: R134a and R152a. Table 1 shows the properties of the cooling fluids. The heat flux in the evaporator was fixed: $Q_{\text{eva}} = 5[\,\text{kW}]$. The operation pressures by Dahmani *et al.* (2011) [13] are:

- Generator: $P_{\text{gen}} = 2400 - 3200[kPa]$,
- Condenser: $P_{\text{con}} = 665.8[\text{kPa}]$
- evaporator: $P_{\text{eva}} = 349.9[kPa]$

Figures 4 and 5 show comparison of the numerical and literature data for the R134a. In Figure 4, the mass flow in the generator for the different operating pressures is shown. Less than 2% of errors are observed. As the heat flux in the evaporator $\mathrm{Q_{exa}}$ is fixed, pressure in the generator increases, and the mass flow rate and the heat flux in the generator $\mathrm{Q_{gen}}$ decrease. Due to less $\mathrm{Q_{gen}}$ and pump work, the COP increases, Figure 5. Errors of less than 5% are observed. Model validation with the refrigerant R152a causes a similar behavior, showing a slight increase in error rates, as shown in Figures 6 and 7 from mass flow in the generator and COP, respectively. Errors of less than 7% are obtained. Errors lower than 15% are obtained in the calculation of the COP, as shown in Figure 7.

Table 1 Properties of R134a and R152a refrigerants [29]

	R134a	R152a
chemical formula	CF_3CH_2F	CH_3CHF_2
molecular weight [kg/kmol]	102.03	66.05
Boiling at normal condition T[°C]	-26.11	-24.7
critical P[kPa]	4068	4760
critical T[°C]	101	114
Toxicity	A1	A2
ODP	0	0
GWP	1300	120

The trend of the mass flow ratio in the generator is maintained for the two refrigerants (R134a and R152a) as shown in Figures 4 and 6.

The results obtained for the COP using R134a and R152a refrigerants fall within the ranges reported by various researchers. The reported COP ranges in the scientific literature were (0.1 - 0.65) and (0.25 - 0.7) for R134a and R152a, respectively [30–32].

4. Results discussion

The performance of the ejector cooling system (COP, mass flow rate, pump work) was evaluated using different conventional and non-conventional refrigerants: HFC's (R134a, R152a), HC's (R290, R600a), R717 (ammonia) and R718 (water). Table 2 shows the properties of each refrigerant used. In this process, the operating parameters are constant in the generator, the condenser, and the evaporator, $Q_{gen} = 100\,\text{kW}$ from a diesel engine that has a generation of 318 kW and 31.45 % of performance; $\mathrm{P_{eva}}$ and $\mathrm{P_{con}}$ are considered saturated pressures at 5ºC and 25ºC respectively.

Table 2 Properties of the refrigerants [29]

Cooling Fluid	Tboiling @ 1 [bar]	Critical conditions		Security	
		P [kPa]	T [ºC]	Toxicity y	Inflammability
R134a	-26.14	4059	101	A	1
R152a	-24.05	4520	113.3	A	2
R290	-42.8	4247	96.68	A	3
R600a	-10.2	3647	135	A	3
R717	-33	11333	132.3	B	2
R718	100	22064	374	A	1

Table 3 Work pressure in the condenser and evaporator, and latent heat of vaporization at $\mathrm{P_{eva}}$ for refrigerants

Cooling Fluid	Pressure [kPa]		$\mathrm{h_{fg}}$@ P_{evap} [kJ/kg]
	Condenser	Evaporator	
R134a	665.8	349.9	194.7
R152a	597.2	315.2	301.2
R290	952.2	551.2	367.4
R600a	350.8	187.5	345.8
R717	1003	516	1244
R718	3.169	0.87	2460

Table 3 shows the condenser and evaporator pressure for each refrigerant used. The pressure range in the generator for each refrigerant is fixed, ensuring the heating of the primary fluid and the vacuum pressure in the suction chamber in the ejector.

4.1 COP

Figure 8 shows the behavior of the COP for each refrigerant based on the pressure relation between the generator and the condenser: $\mathrm{P_{gen}/P_{con}}$. In Figure 8, the different ranges of $\mathrm{P_{gen}}$ are shown to which the system must operate for each studied refrigerant. A trend of increasing COP is observed when the pressure of $\mathrm{P_{gen}}$ increases.

The R718 works at high pressures in the generator and at very low pressures in the condenser and evaporator (see Table 3 and Figure 8). The low-pressure condition in the condenser and evaporator causes operational problems in the system due to the air filtration. The R718 is recognized for its ability to support a wide range of operating pressures in contrast with the others, where the operating range is limited. The refrigerant with a higher COP observed is the R717. Table 4 shows the COP at the maximum value of $\mathrm{P_{gen}/P_{con}}$ for each working fluid. The refrigerants that obtain a higher COP are the R717, the

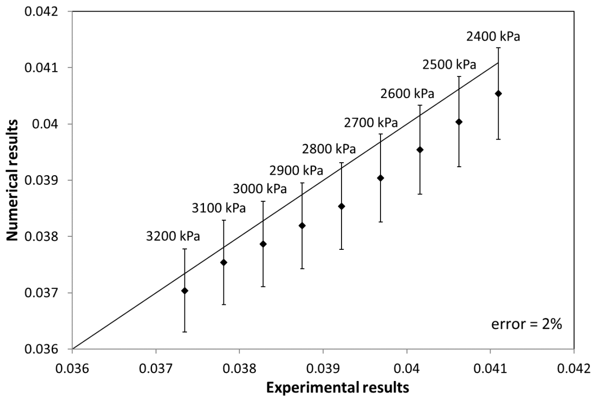

Figure 4 R134a mass flow rate of the generator for experimental data [16] and present numerical predictions.

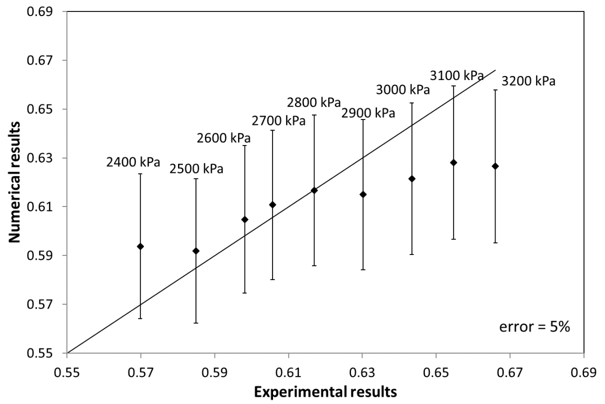

Figure 5 R134a COP for experimental data [16] and present numerical predictions

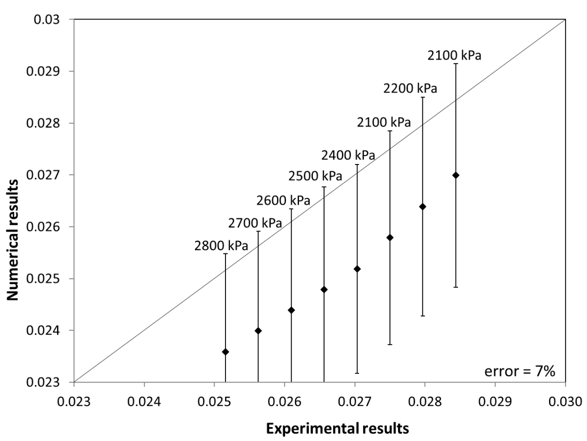

Figure 6 R152a mass flow rate of the generator for experimental data [16] and present numerical predictions

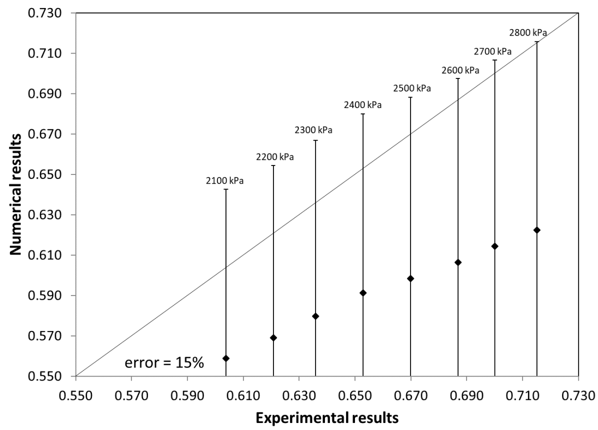

Figure 7 R152a COP for experimental data [16] and present numerical predictions

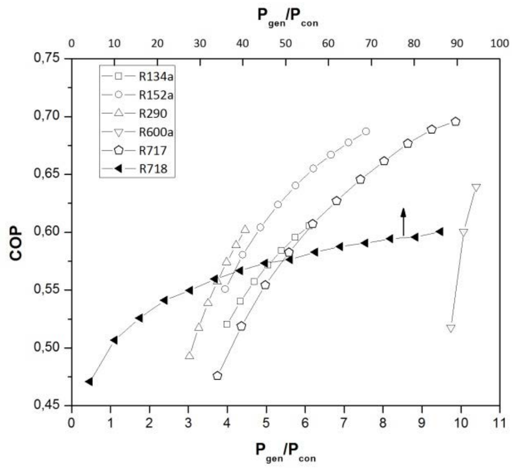

Figure 8 Pressure range in the generator

R125a, and the R600 with a COP of 0.6766, 0.669, and 0.6391, respectively.

4.2 Mass flow

Table 4 shows the mass flow of each cooling fluid for $Q_{gen} = 100\,kW$ and COP maximum. Table 4 shows that R718 and R717 have a lower cooling mass flow rate, with a flow of 0.05202 kg/s and 0.1197 kg/s, respectively. The lower cooling mass flow rate is due to the high vaporization heat latent of cooling fluids R718 and R717, as shown in Table 3. The lower flow used represents a reduction of the environmental impacts and system cost due to the use of refrigerants and the scale of the system.

4.3 Pump

The main advantage of the proposed system lies in substituting the mechanical compressor with a thermo-compressor, significantly reducing the power consumption of the cooling system. However, it is necessary to use a hydraulic pump, which ensures the refrigerant flow in the generator. Table 4 compares the pump work from COP maximum for each cooling fluid. Table 4 shows that the pump work used in the R718 is very low and negligible in the energy balance. This is due to the low mass flow required when R718 is used, 0.05202 kg/s. Refrigerants R134a, R152a, R290, and R717 have power consumption in the pump (0.8-1.5 kW) close between them due to the similar pressure range in the generator. R600a has a high consumption in the pump due to the high pressure that it operates in the generator (2.6 kW), 10 to 12 times the pressure in the condenser; see Figure 8.

5. Conclusions

The study presents a mathematical model that calculates the thermodynamic states, the operating conditions for the operation, and the sizing of the constituent elements. An

Table 4 COP, Mass flow rate, W_{pump}, and Q_{evap}

	COP	mass flow [kg/s]	W_{pump} [kW]	Q_{evap} [kW]
R134a	0.6053	0.8201	1.289	61.31
R152a	0.6871	0.5373	1.220	69.55
R290	0.6022	0.4354	1.636	61.21
R600a	0.6394	0.4702	2.569	65.58
R717	0.6957	0.1203	0.822	70.14
R718	0.6005	0.0500	0.006792	60.05

Nomenclature

Symbols		
A	m^2	*area*
h	$\mathrm{J/kg}$	*enthalpy*
M		*Mach number*
$\dot{m}$	$\mathrm{kg/s}$	*mass flow rate*
P	Pa	*pressure*
Q	J	*heat*
S	$\mathrm{J/kg}.K$	*entropy*
T	K	*temperature*
x		*vapor quality*
V	m/s	*speed*
w		*entrainment ratio*
y		*compressibility relation*
Subindex		
1, 2, 3, ...	*position in the system*	
o, i, ii, iii,...	*position in the ejector*	
con	*condenser*	
evap	*evaporator*	
gen	*generator*	
Superindex		
*	*critical conditions*	

acceptable validation of the model was performed, finding an adequate deviation between the results obtained and the experimental results reported in the literature, which were nearly 5% and 15% for R134a and R152a, respectively.

Six types of refrigerants were studied (HFC's (R134a, R152a), HC's (R290, R600a), R717 (ammonia), and R718 (water) with their respective operating conditions. The R717 and R152a have a high COP, 0.6957 and 0.6871, respectively. However, the R717 requires a lower mass flow rate than the R125a and, consequently, a lower power consumption in the pump.

Lower pump consumption (0.006792 kW) is presented by R718 due to a lower mass flow (0.05 kg/s). R718 is considered a natural refrigerant (working fluid) because of its non-toxicity, non-flammability, and very low cost.

6. Declaration of competing interest

The Authors declare that they have no significant competing interests, including financial or non-financial, professional, or personal interests interfering with the full and objective presentation of the work described in this manuscript.

7. Acknowledgements

The authors acknowledge the Universidad Autónoma de Bucaramanga (UNAB) and Research Group "GIRES".

8. Funding

This work was developed for projects I34001 and I56040 supported by Universidad Autónoma de Bucaramanga.

9. Author contributions

Leonardo Pacheco Sandoval Conceived and designed the analysis and wrote the paper, C.A. Díaz G. contributed to the modeling and critical manuscript revision. G.D. Acevedo R. contributed to analysis of results and wrote the paper. M.A. Rodriguez C. contributed to carried out the simulation and analysis results.

10. Data availability statement

The authors confirm that the data supporting the findings of this study are available within the article [and/or] its supplementary materials.

References

[1] S. Xu, X. Huang, and R. Du, "An investigation of the solar powered absorption refrigeration system with advanced energy storage technology," *Solar Energy*, vol. 85, no. 9, Apr. 20, 2011. [Online]. Available: https://doi.org/10.1016/j.solener.2011.04.022

[2] S. Zhang, Z. Lin, and Y. Cheng, "Optimizing the set generating temperature to improve the designed performance of an ejector cooling system with thermal pumping effect (ecstpe)," *Solar Energy*, vol. 157, Aug. 10, 2017. [Online]. Available: https://doi.org/10.1016/j.solener.2017.08.027

[3] G. Besagni, R. Mereu, and F. Inzoli, "Ejector refrigeration: A comprehensive review," *Renewable and Sustainable Energy Reviews*, vol. 53, Aug. 04, 2015. [Online]. Available: https://doi.org/10.1016/j.rser.2015.08.059

[4] A. Mota-Babiloni, J. Navarro-Esbrí, A. Barragán-Cervera, F. Molés, B. Peris, and G. Verdú, "Commercial refrigeration – an overview of current statusfroid commercial. vue d'ensemble du statut actue," *International Journal of Refrigeration*, vol. 57, Apr. 15 2015. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2015.04.013

[5] I. Sarbu, "A review on substitution strategy of non-ecological refrigerants from vapour compression-based refrigeration, air-conditioning and heat pump systems," *International Journal of Refrigeration*, vol. 46, Apr. 27, 2014. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2014.04.023

[6] A. Kasaeian, A. Shamaeizadeh, and B. Jamjoo, "Combinations of rankine with ejector refrigeration cycles: Recent progresses and outlook," *Applied Thermal Engineering*, vol. 211, Mar. 19, 2022. [Online]. Available: https://doi.org/10.1016/j.applthermaleng.2022.118382

[7] G. Besagni, R. Mereu, and F. Inzoli, "Ejector refrigeration: A comprehensive review," *Renewable and Sustainable Energy Reviews*,

vol. 53, Aug. 04, 2015. [Online]. Available: https://doi.org/10.1016/j.rser.2015.08.059

[8] R.Fingas, M. Haida, J. Smolka, G. Besagni, J. Bodys, M. Palacz, and *et al.*, "Experimental analysis of the air-to-water ejector-based r290 heat pump system for domestic application," *Applied Thermal Engineering*, vol. 239, Oct. 15, 2023. [Online]. Available: https://doi.org/10.1016/j.applthermaleng.2023.121800

[9] M. Morid and M. H. Khoshgoftar-Manesh, "6e evaluation of an innovative polygeneration system consisting of gas turbine cycle with co2 capture, ejector refrigeration cycle, steam rankine cycle, solar tower and medad unit," *Thermal Science and Engineering Progress*, vol. 46, Oct. 20, 2023. [Online]. Available: https://doi.org/10.1016/j.tsep.2023.102234

[10] K. Abbady, N. Al-Mutawa, and A. Almutairi, "The performance analysis of a variable geometry ejector utilizing cfd and artificial neural network," *Energy Conversion and Management*, vol. 291, Sep. 01, 2023. [Online]. Available: https://doi.org/10.1016/j.enconman.2023.117318

[11] S. He, Y. Li, and R. Wang, "Progress of mathematical modeling on ejectors," *Renewable and Sustainable Energy Reviews*, vol. 13, no. 8, Sep. 30, 2008. [Online]. Available: https://doi.org/10.1016/j.rser.2008.09.032

[12] N. Al-Khalidy, "An experimental study of an ejector cycle refrigeration machine operating on r113," *International Journal of Refrigeration*, vol. 21, no. 8, Mar. 27, 1998. [Online]. Available: https://doi.org/10.1016/S0140-7007(98)00030-9

[13] R. Y. F. Akkurt, "Experimental investigation on ejector cooling system performance at low generator temperatures and a preliminary study on solar energy," *Journal of Mechanical Science and Technology*, vol. 26, Jun. 13, 2012. [Online]. Available: https://doi.org/10.1007/s12206-012-0841-5

[14] K. Megdouli, M. Elakhdar, E. Nahdi, L. Kairouani, and A. Mhimid, "Journal of physics: Conference series," in *Tunisia-Japan Symposium: R&D of Energy and Material Sciences for Sustainable Society*, Gammarth, Tunisia, 2014.

[15] K. Cizungu, A. Mani, and M. Groll, "Performance comparison of vapour jet refrigeration system with environment friendly working fluids," *Applied Thermal Engineering*, vol. 21, no. 5, May. 15, 2000. [Online]. Available: https://doi.org/10.1016/S1359-4311(00)00070-3

[16] A. Dahmani, Z. Aidoun, and N. Galanis, "Optimum design of ejector refrigeration systems with environmentally benign fluids," *International Journal of Thermal Sciences*, vol. 50, no. 8, Feb. 20, 2011. [Online]. Available: https://doi.org/10.1016/j.ijthermalsci.2011.02.021

[17] R. Roman and J. I. Hernandez, "Performance of ejector cooling systems using low ecological impact refrigerants," *International Journal of Refrigeration*, vol. 34, no. 7, Mar. 12, 2011. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2011.03.006

[18] J. Chen, H. Havtun, and B. Palm, "Parametric analysis of ejector working characteristics in the refrigeration system," *Applied Thermal Engineering*, vol. 69, no. 1-2, Apr. 19, 2014. [Online]. Available: https://doi.org/10.1016/j.applthermaleng.2014.04.047

[19] A. Selvaraju and A. Mani, "Analysis of an ejector with environment friendly refrigerants," *Applied Thermal Engineering*, vol. 24, no. 5-6, Aug. 29, 2023. [Online]. Available: https://doi.org/10.1016/j.applthermaleng.2003.08.016

[20] R. Yen, B. Huang, C. Chen, T. Shiu, C. Cheng, S. Chen, and *et al.*, "Performance optimization for a variable throat ejector in a solar refrigeration system," *International Journal of Refrigeration*, vol. 36, no. 5, Apr. 05, 2013. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2013.04.005

[21] D. Butrymowicz, K. Śmierciew, J. Karwacki, and J. Gagan, "Experimental investigations of low-temperature driven ejection refrigeration cycle operating with isobutane," *International Journal of Refrigeration*, vol. 39, Oct. 27, 2013. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2013.10.008

[22] D. Śmierciew, J. Gagan, D. Butrymowicz, and J. Karwacki, "Experimental investigations of solar driven ejector air-conditioning system," *Energy and Buildings*, vol. 80, May. 24, 2014. [Online]. Available: https://doi.org/10.1016/j.enbuild.2014.05.033

[23] T. Sankarlal and A. Mani, "Experimental investigations on ejector refrigeration system with ammonia," *Renewable Energy*, vol. 32, no. 8, May. 29, 2006. [Online]. Available: https://doi.org/10.1016/j.renene.2006.05.008

[24] G. Besagni, R. Mereu, and F. Inzoli, "Ejector refrigeration: A comprehensive review," *Renewable and Sustainable Energy Reviews*, vol. 53, Aug. 04, 2015. [Online]. Available: https://doi.org/10.1016/j.rser.2015.08.059

[25] M. Zeyghami, D. Yogi-Goswami, and E. Stefanakos, "A review of solar thermo-mechanical refrigeration and cooling methods," *Renewable and Sustainable Energy Reviews*, vol. 51, Jul. 06, 2015. [Online]. Available: https://doi.org/10.1016/j.rser.2015.07.011

[26] H. González-Bravo, R. Dorantes-Rodríguez, J. Hernández-Gutiérrez, R. B. y Brown, R. Román-Aguila, and H. Terres-Peña, "State of art of simple and hybrid jet compression refrigeration systems and the working fluid influence," *International Journal of Refrigeration*, vol. 35, no. 2, Oct. 26, 2011. [Online]. Available: https://doi.org/10.1016/j.ijrefrig.2011.10.012

[27] L. Boumaraf, P. Haberschill, and A. Lallemand, *Ejectors and their usefulness in the energy savings*. New York: Nova Science Publ, 2010.

[28] H. El-Dessouky, H. Ettouney, I. Alatiqi, and G. Al-Nuwaibit, "Evaluation of steam jet ejectors," *Chemical Engineering and Processing: Process Intensification*, vol. 41, no. 6, Sep. 27, 2001. [Online]. Available: https://doi.org/10.1016/S0255-2701(01)00176-3

[29] B. Poling, J. Prausnitz, and J. O'Connell, *The properties of gas and liquids*, 5th ed. New York: Mcgraw Hill, 2000.

[30] Z. Zhang, X. Feng, D. Tian, J. Yang, and L. Chang, "Progress in ejector-expansion vapor compression refrigeration and heat pump systems," *Energy Conversion and Management*, vol. 207, Jan. 21, 2020. [Online]. Available: https://doi.org/10.1016/j.enconman.2020.112529

[31] B. M. Tashtoush, M. A. Al-Nimr, and M. A. Khasawneh, "A comprehensive review of ejector design, performance, and applications," *Applied Energy*, vol. 240, Jan. 19, 2019. [Online]. Available: https://doi.org/10.1016/j.apenergy.2019.01.185

[32] Z. Aidoun, K. Ameur, M. Falsafioon, and M. . Badache, "Current advances in ejector modeling, experimentation and applications for refrigeration and heat pumps. part 1: Single-phase ejectors," *Inventions*, vol. 4, no. 1, Feb. 26, 2019. [Online]. Available: https://doi.org/10.3390/inventions4010015

[33] B. M. Tashtoush, M. A. Al-Nimr, and M. A. Khasawneh, "A comprehensive review of ejector design, performance, and applications," *Applied Energy*, vol. 240, Jan. 19, 2019. [Online]. Available: https://doi.org/10.1016/j.apenergy.2019.01.185

[34] Z. Aidoun, K. Ameur, M. Falsafioon, and M. Badache, "Current advances in ejector modeling, experimentation and applications for refrigeration and heat pumps. part 1: Single-phase ejectors," *Inventions*, vol. 4, no. 1, Feb. 26, 2019. [Online]. Available: https://doi.org/10.3390/inventions4010015