EXERGETIC OPTIMIZATION OF ABSORPTION REFRIGERATION

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ABSTRACT

Nowadays, several scientific studies aim to improve the refrigeration systems commonly used to reduce the consumption of electric energy as well as the environmental impact caused by this equipment. However, it is desired that this be done together with increased efficiency and reduced production cost of the system. Absorption refrigeration systems offer this opportunity to save energy, as they can use thermal energy to produce, residual heat and geothermal energy as primary energy. In addition, they use very ecological working fluids, drawing the attention of the scientific academic world in recent decades. Currently, thermodynamic analyzes based on exergy are increasingly being implemented to calculate the performance of thermodynamic systems, where just considering COP as an efficiency parameter is no longer sufficient. The exergetic analysis takes into account the irreversibility of the system and can indicate which components need to be improved to have a better system performance. Taking this into account, this paper presents the modeling and exergetic optimization of an absorption refrigeration system that uses ammonia and water as working fluids. The thermodynamic model of the refrigerator was developed based on the principles of mass and energy conservation under the steady-state, and was implemented using the Engineering Equation Solver (EES) software. Regarding the performance of the modeled refrigerator, a value of COP = 0.4571. A parametric analysis of the system was carried out with the results obtained numerically from the proposed model, where the relevance of some operating parameters for the performance coefficient and the exergetic efficiency of the system was evaluated. An exergetic analysis of the system was also carried out, where it was shown that the generator and the absorber are responsible for 56.4% and 29.2%, respectively of the total destroyed exergy. Moreover, based on the proposed thermodynamic model, an exergetic optimization of the cooling system was performed based on parameters such as generator temperature and absorber pressure. Thus, it can be concluded that the model developed can be used as a useful tool in the study of absorption chillers possible to predict the impact on the system performance, taking into account various operating conditions.

Keywords: heat transfer enhancement; optimal geometry; eccentricities

NOMENCLATURE

e	flow specific exergy, kJ/kg
ED	exergy destroyed, W
h	specific enthalpy, kJ/kg
$\dot{m}$	mass flow, kg/s
P	pressure, N/m^2
$\dot{Q}$	heat transfer rate, W
Q	quality,
s	specific entropy, kJ/kg.K
T	temperature, K
x	mass fraction
v	specific volume, m^3/kg
$\dot{W}$	potency, W
η	efficiency
COP	coefficient of performance

Subscripts

A	absorber
B	pump
C	condenser
E	evaporator

G generator
ger/ret generator/rectifier
R rectifier
VE1 expansion valve 1
VE1 expansion valve 1

INTRODUCTION

There are several types of cooling systems adopted today. Each one with its particularities and application objectives that will make it ideal for an application. In particular, the absorption system uses as its main source the energy received in the form of heat, which can be acquired in different ways, such as solar, geothermal, residual heat biomass, residual biomass in various ways. However, this type of system is not as used when compared to systems that operate based on the compression cycle by steam, which have greater efficiency, but whose main source is electrical energy.

The absorption refrigeration cycle is believed to be the oldest among the cycles that operate by steam compression, air, thermoelectricity and thermomagnetism, having its foundations dated in 1777 by the Scottish Nairm (Martinez, 2018). According to Stoecker and Jones (1985), the first patent was made in the United States in 1860 by the Frenchman Ferdinan Carré. The first use of the system in the United States was probably made by the Confederate States during the Civil War to supply natural ice that had been cut off by the North (Stoecker and Jones, 1985).

Absorption refrigeration systems operate according to a heat-fed refrigeration cycle, where a secondary or absorbent fluid in the liquid phase is responsible for absorbing the primary fluid or refrigerant, in the form of vapor (Martinho, 2013). Kim and Park (2007) also add that, in absorption systems, the mechanical process of the vapor compression system is replaced by a physical-chemical process, using energy in the form of heat instead of mechanical work. As sources of energy in the form of heat, absorption refrigerators can use solar, geothermal, biomass and waste heat energy. In addition, they are silent systems, free from mechanical vibrations, require little maintenance and are ecological (Ziegler, 1999).

Absorption refrigerators have the advantage of using a non-electrical energy source, ensuring flexibility to the designer/owner/operator (Herold et al., 2016). In other words, this type of refrigerator stands out precisely due to the development of new technologies that use other energy sources, such as waste heat from gas and steam turbines, solar, geothermal and biomass energy (Adewusi; Zubair, 2004). Vargas et al., 2000, also explain that the fact that the absorption refrigerator uses low-content heat sources draws attention both for economic reasons and for the current need for refrigeration systems with low environmental impact. However, this type of refrigeration still needs to compete with modern and conventional refrigeration systems and, in the In the United States, for example, the existence of reliable and low-cost electricity reduces the use of another type of refrigeration other than the commonly used one, which is vapor compression refrigeration (Herold; Radermacher; Klein, 2016).

When analyzing the National Energy Balance 2021, base year 2020, carried out by the Energy Research Company that is linked to the Ministry of Mines and Energy, the industrial, residential and commercial sectors account for almost 80% (79.9%) of the electricity consumed throughout the year 2020. It is important to note that during the year 2020, Brazil, like the whole world, was in the first year of facing sar-cov-2, so there was an increase in residential energy consumption. Industry leads consumption with around 36.6% of total consumption, followed by the residential sector (27.6%) and commercial (15.7%), as shown in Fig. 1.

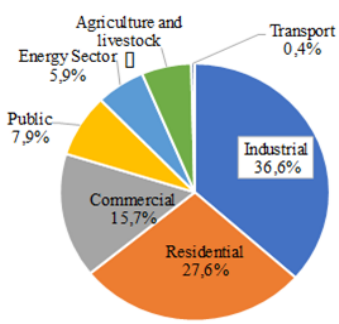

Figure 1. Sector participation in electricity consumption in Brazil.

The same behavior of energy consumption associated with electricity can be found in several other countries throughout the world. An important example is the United States of America (USA). Where according to U.S. Energy Information Administration, that the four sectors responsible for the most energy consumption in the US are: residential, commercial, industrial and transportation. The collected data are schematized in Fig.2.

Another important point that shows to be observed in the other point regarding American energy consumption is greater in production and in renewable energies such as solar, even more expressive for the year 2050, which is the consumption of fossil fuels such as oil and natural gas. remain dominant. This is demonstrated in Fig.3.

Thus, it is very important to study the development of equipment and devices that allow the reduction of energy consumption without reducing energy efficiency and their quality. In this context, absorption refrigeration systems offer an alternative to vapor compression refrigeration systems.

In addition, absorption refrigeration systems use working fluids that do not degrade the environment as much, thus reducing the environmental impact that conventional refrigeration systems cause. In this way, the probability of reducing electrical energy consumption together with the reduction of environmental impacts and the possible use of low-cost energy sources, aroused the interest of the scientific community in relation to absorption refrigeration systems in recent decades.

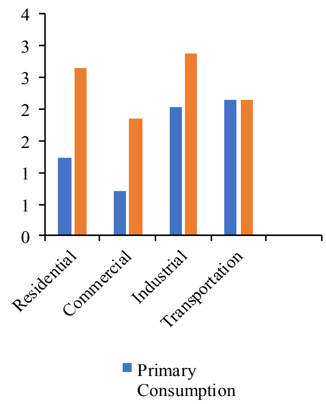

Fig.2. Energy consumption by sector in trillion Btu, January 2022.

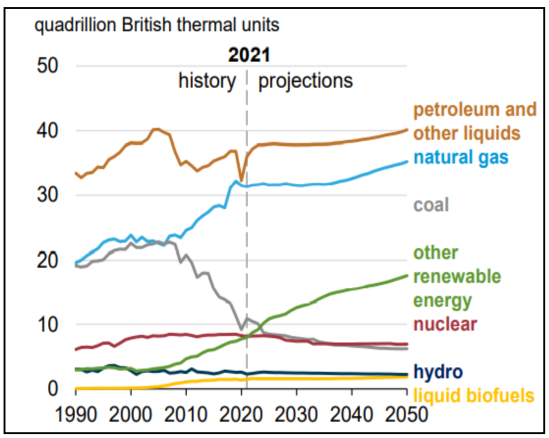

Fig.3. Energy Consumption by fuel.

This article proposes an optimization strategy based on the exergetic efficiency of na NH_3H_2O absorption refrigerator. The main contributions of this article are summarized below,

1) The simplified mathematical model is developed based on the thermodynamics and heat transfer theory;
2) Development of a computational code in line with the mathematical model capable of predicting system responses with low computational cost and response time;
3) Perform a parametric analysis with the numerical results obtained from the mathematical model;
4) Perform the exergetic optimization of the system for different operating conditions.

METHODOLOGY

Mathematical model of the refrigeration system by single stage of absorption

The mathematical model was mainly based on the application of the principles of conservation of mass and energy in steady-state for each previously defined control volume. Each component will be defined as a single control volume with uniform properties in its domain. To solve the equations and obtain the thermodynamic properties of the system, the Engineering Equation Solver (EES) software was used.

The physical problem of this work consists of an absorption refrigeration system composed of an absorber, a condenser, a pump, two expansion valves, a generator/rectifier, and an evaporator. Each component mentioned was defined as a single control volume, where the laws of conservation of mass and energy will be applied, as well as the exergy flows that enter and leave will be analyzed of each defined control volume. The generator/rectifier set was defined only as a control volume, that is, in the mathematical model developed in this work, at the top of the thermal generator there is a rectification column responsible for ensuring that only ammonia vapor enters the condenser. The thermodynamic cycle of the absorption refrigerator with all the components mentioned above is shown in Figure 4.

For carry out the modeling of the proposed system and described in Figure 8, the following initial considerations were made: i) The system operates in steady-state; ii) The pressure losses in the lines and in the heat exchangers due to friction are not considered; iii) The flows through the expansion valves are considered isenthalpic; iv) Variations in kinetic and potential energy will not be considered; v) The water and ammonia solutions in the generator/rectifier and absorber are in equilibrium at their respective temperatures and pressures.

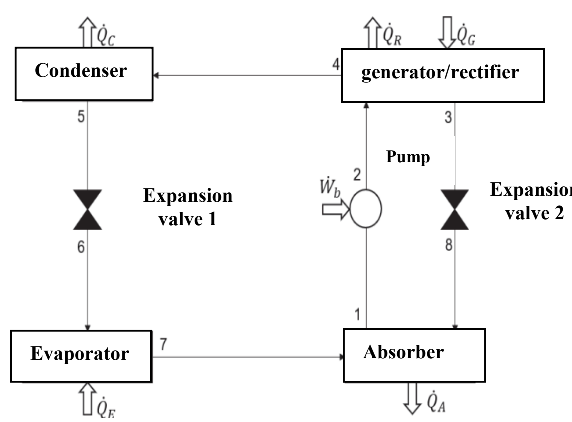

Figure 4. Single-stage absorption refrigeration cycle used in mathematical modeling.

Before starting the modeling, it is necessary to define some input parameters for the simulation. Thus, the mass flow through the pump, the temperature at which the mixture is leaving the absorber and the generator, the rate of heat transfer to the generator and the system pressures were chosen. The initial values chosen are shown in Table 1.

Table 1. Input parameters of the mathematical model of the absorption refrigerator.

Parameters	Symbol	Unit	Value
pump mass flow		kg/s	0.01
pressure at the outlet of the absorber		kPa	550
pressure at the generator inlet		kPa	1100
absorber temperature		ºC	30
generator temperature		ºC	100
heat transfer to the generator		kW	10
pump efficiency		-	1
mass fraction of ammonia at the generator/rectifier output		-	0.9996

Initially, a mass balance was made in relation to the flow that enters and leaves the pump. As the working fluid of this system is a two-phase mixture of ammonia and water, it is also necessary to balance the amount of ammonia in all control volumes. Thus, we have the following relations:

$$\dot{m}_1 = \dot{m}_2 \quad (1)$$

$$x_1 = x_2 \quad (2)$$

Where x_1 and x_2 correspond to the proportions of ammonia in the mixture before and after passing through the pump, respectively. The value of x_1 is calculated using the EES routine for the NH_3H_2O mixture as a function of T_1, P_1, and Q_1, with Q_1 corresponding to the titer at that certain point. It was decided that the mixture leaving the absorber is in the saturated liquid state, that is, $Q_1 = 0$. Already $\dot{m}_1$, the mass flow rate of the mixture entering the pump, had its value previously defined, and according to the balance of mass, $\dot{m}_2$ as well.

The specific work of the pump is defined by the product of the specific volume at point 1 by the pressure variation between points 1 and 2. Thus, we have:

$$w = \frac{v_1(P_2 - P_1)}{n_B} \quad (3)$$

The pressures and pump efficiency (η_B) have already been defined as input parameters of the system. The specific volume was defined using the routine of this property for the NH_3H_2O mixture, with the following inputs: T_1, P_1, and Q_1. The enthalpy point 1, h_1, was also defined using the same inputs, changing only the function to enthalpy.

Having determined the specific work of the pump and the specific enthalpy h_1, through the energy balance, it is possible to determine the specific enthalpy of point 2. That is:

$$h_2 = h_1 + w \quad (4)$$

Thus, the pump power can be determined through the following relationship:

$$\dot{W}_B = \dot{m}\left(h_2 - h_1\right) \quad (5)$$

Generator/Rectifier

When passing through the pump, the system pressure is increased from P1 to P2, a value initially determined in the model's input parameters. At the top of the generator, point 4 of the cycle, it was defined that the mass fraction of ammonia in the mixture is 0.996. Performing a mass balance on this component, we have:

$$\dot{m}_2 = \dot{m}_3 + \dot{m}_4 \quad (6)$$

The temperature at point 4 (T_4) is determined using x_4, P_4 and Q_4 as inputs to the EES function, where it was considered that $Q_4 = 1$, that is, the fluid leaving the liquid generator/rectifier component consists of saturated steam and as there was no change in the system pressure, we have that:

$$P_4 = P_2 \quad (7)$$

With the value of T_4 defined, together with the pressure P_4 and the quality Q_4, it is possible to obtain the enthalpy at that point (h_4).

Point 3 corresponds to the weak ammonia solution that leaves the generator and returns to the absorber. To determine the enthalpy at this point, the temperature T_3 was used, the input parameter already defined, and the mixture was considered in the state of saturated liquid, that is, $Q_3 = 0$. There was also no change in the pressure level at that point, that is:

$$P_3 = P_2 \quad (8)$$

For determine the mass flow at points 3 and 4, it is necessary, in addition to the mass balance, to take into account the amount of ammonia through an ammonia balance in the component, as follows:

$$\dot{m}_2 x_2 = \dot{m}_3 x_3 + \dot{m}_4 x_4 \quad (9)$$

Finally, performing the energy balance for the generator/rectifier, $\dot{Q}_R$ is determined, which corresponds to the necessary heat transfer provided by some source of heat to the system, through the following relationship:

$$\dot{m}_2 h_2 - \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{Q}_R - \dot{Q}_G = 0 \quad (10)$$

Condenser

Applying the mass and ammonia balance between the inlet and outlet of the condenser and knowing that the pressure in this component consists of the high pressure of the system, we have that:

$$\dot{m}_4 = \dot{m}_5 \quad (11)$$

$$x_4 = x_5 \quad (12)$$

$$P_4 = P_5 \quad (13)$$

$$h_6 = h_5 \quad (14)$$

With the values of pressure (P_5) and ammonia mass fraction (x_5) and, considering that the fluid leaves the condenser in the saturated liquid state, both the temperature at the exit of the condenser T_5 and the specific enthalpy for this is determined point, h_5. Applying the proper energy balance, $\dot{Q}_C$ is determined as follows:

$$\dot{m}_4 h_4 - m_5 h_5 + \dot{Q}_C = 0 \quad (15)$$

Expansion valve 1

When in operation, the cycle works under two pressure levels: a high one (condenser saturation pressure) and a low one (evaporator saturation pressure). When passing through the expansion valve, from points 5 to 6, the pressure is reduced to the low pressure of the system, that is:

$$P_6 = P_1 \quad (16)$$

Doing the mass and ammonia fraction balance, we have $\dot{m}_5 = \dot{m}_6$ and $x_5 = x_6$. Besides, considering that the throttling process is isentropic, the following relationship holds $h_5 = h_6$.

With this, the title of the mixture at the valve outlet (Q_6) is calculated from the values of pressure (P_6), enthalpy (h_6) and ammonia mass fraction (x_6). And later, with the value of Q_6, together with P_6 and x_6, the temperature at the outlet of the valve is determined expansion and inlet of the T_6 evaporator

$$\dot{m}_7 h_7 - \dot{m}_8 h_8 - \dot{m}_1 h_1 + \dot{Q}_A = 0$$

$$e_i = h_i - h_0 - T_0 \left(s_i - s_0 \right).$$

Evaporator

In this component, the operating pressure is the saturation pressure of the evaporator, that is, it consists of the low pressure of the system. It is in the low pressure region that the refrigerant fluid evaporates and with that the refrigeration of a desired fluid or environment is obtained. So, we have that $P_1 = P_7$. And, applying ammonia mass and mass fraction balance, we get $\dot{m}_6 = \dot{m}_7$ and $x_6 = x_7$.

To determine both the enthalpy h_7 and the temperature T_7, the values obtained for P_7 and x_7, in addition to considering that the mixture leaves the evaporator as saturated steam, that is, $Q_7 = 1$. Finally, energy balance applies in the evaporator, determining the heat transfer in this component, $\dot{Q}_E$, as follows:

$$\dot{m}_6 h_6 - \dot{m}_7 h_7 + \dot{Q}_E = 0 \quad (17)$$

Absorber

For this component, it is not necessary to perform a mass and ammonia balance, because as it is a cycle, the mass and ammonia fluxes are also cyclic and have already been determined for the other components. Knowing this, only an energy balance is applied to determine the heat transfer that must be rejected by the absorber, taking into account the ammonia vapor that comes from the evaporator, the ammonia-rich solution that goes to the pump, and the lean solution ammonia that returns from the generator, symbolized by state 8 in Fig. 4.

$$\dot{m}_7 h_7 - \dot{m}_8 h_8 - \dot{m}_1 h_1 + \dot{Q}_A = 0 \quad (18)$$

Expansion valve 2

This expansion valve corresponds to the component between the generator/rectifier and the absorber responsible for reducing the pressure to the working pressure of the absorber. Thus, there was a reduction in the pressure of the system and, similarly to another valve, the pressure reduction process was considered isoenthalpic, we have that $P_8 = P_1$ and $h_8 = h_3$. And, through the mass and ammonia balance, for this component, we have that $\dot{m}_8 = \dot{m}_3$. Thus, it is possible to determine both the Q_8 titer of the mixture leaving this component and the T_8 temperature through the appropriate EES functions using the enthalpy, mass fraction, and pressure values at this point.

Exergetic Analysis

For perform exergetic analysis, it is necessary to determine the exergy flows of the entire system. For this, Eq. (2) which calculates the flow-specific exergy at each point in the cycle. An exergy balance was performed for the Control Volumes considered, it is possible to obtain the main properties at points in the cycle, such as temperature, pressure, ammonia mass fraction of the fluid and enthalpy at each point in the cycle. Thus, for the use of $e_i = h_i - h_0 - T_0\left(s_i - s_0\right)$, it remains to define the entropies (s_i) and the properties of the dead state (h_0, s_0) for each point of the cycle in Figure 8. To determine the entropies of the system, the EES library was used from the thermodynamic properties calculated in each point. As for the dead state, the properties were defined as the following ambient conditions: T_0 = 20ºC and P_0 = 101.325 kPa. With this, the values of h0 and s0 are also determined.

Destroyed exergy from the absorption refrigeration cycle

Pontos	x_i	P_i (kPa)	T_i (°C)	h_i (kJ/kg)	$\dot{m}_i$ (kg/s)	s_i (kJ/kg.K)
1	0.5925	550	30	-93.82	0.01	0.2973
2	0.5925	1100	30.5	-93.12	0.01	0.2973
3	0.3339	1100	100	234.8	0.006115	1.263
4	0.9996	1100	35.6	1313	0.003885	4.375
5	0.9996	1100	28.03	131.9	0.003885	0.4779
6	0.9996	550	6.814	131.9	0.003885	0.4886
7	0.9996	550	19.71	1309	0.003885	4.666
8	0.3339	550	80.54	234.8	0.006115	1.273

To determine the exergy destroyed for each component, a steady-state exergy balance is applied to each control volume. The exergy balance takes into account the exergy flows into and out of each control volume, the exergy transfers associated with heat transfers, energy transfer by work, and exergy destroyed in the component. Knowing this and performing the exergy balance for each control volume, it is possible to calculate the rate of exergy destroyed for each component of the system as follows:

$$ED_B = \dot{m}_1.(e_1 - e_2) + W_B \tag{19}$$

$$ED_{ger/ref} = \dot{m}_2.e_2 - \dot{m}_3.e_3 - \dot{m}_4.e_4 + \dot{Q}_G\left[1-\left(\frac{T_0}{T_3}\right)\right] + \dot{Q}_R\left[1-\left(\frac{T_0}{T_4}\right)\right] \tag{20}$$

$$ED_C = \dot{m}_4.(e_4 - e_5 + \dot{Q}_C\left[1-\left(\frac{T_0}{T_5}\right)\right] \tag{21}$$

$$ED_{VE1} = \dot{m}_6.(e_5 - e_6) \tag{22}$$

$$ED_E = \dot{m}_6.(e_6 - e_7) + \dot{Q}_E\left[1-\left(\frac{T_0}{T_7}\right)\right] \tag{23}$$

$$ED_{VE2} = \dot{m}_3.(e_3 - e_8) \tag{24}$$

$$ED_A = \dot{m}_8.e_8 - \dot{m}_7.e_7 - \dot{m}_1.e_1 + \dot{Q}_A\left[1-\left(\frac{T_0}{T_1}\right)\right] \tag{25}$$

RESULTS AND DISCUSSION

The thermodynamic model was implemented in the EES software. With the given parameters, the program calculates the values of temperature, enthalpy, entropy, flow ammonia mass and mass fraction at all points in the cycle. The operational conditions of the absorption system obtained are described in Table 2.

Table 2. Operating conditions of the mathematical model of the absorption refrigerator.

For the absorption refrigeration system in question, the coefficient of performance was equivalent to 0.4571 (COP = 0.4571). The values of heat transfer rates for each component were obtained from the operating conditions applied to the mathematical model and can be seen in Table 3.

Table 3. Heat transfer rates of cycle components of the absorption refrigerator model.

Component	Symbols	Value (KW)
Absorber	$\dot{Q}_A$	7.49
Generator / Rectifier	$\dot{Q}_G/\dot{Q}_R$	10/2.53
Condenser	$\dot{Q}_C$	4.59
Evaporator	$\dot{Q}_E$	4.57

The parametric analysis of the system was performed in order to obtain the behavior and influence of some parameters on the efficiency of the refrigerator, as well as to observe that some parameters do not have decisive interference in others. For this, some of the main parameters of the developed model will be varied and the consequence of this variation will be analyzed in relation to the coefficient of performance of the generator. The Fig. 5 illustrates this behavior.

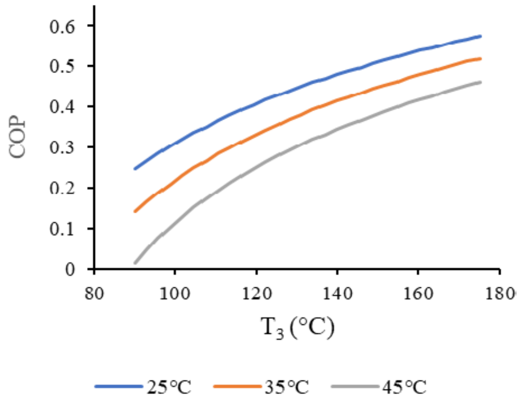

Figure 5. Generator temperature as a function of COP for different condenser temperatures.

With the objective of evaluating the influence of the condenser temperature on the performance of the absorption cooler, the generator temperature, T_3, was varied as a function of the COP for different condenser outlet temperatures, T_5. This can be seen in Fig. 5. For this, it was chosen as values of the model input parameters a temperature of the absorber of T_3 = 25 °C and for system low pressure P_1 = 300 kPa.

Observing Fig.5 it is noted that the system presented higher values of COP for the lowest values of condenser temperature. The COP varies between 0.25 and 0.58 for T_5 = 25 °C and between 0.01 and 0.46 for T_5 = 45 °C. This behavior can be explained because the lower the temperature of the condenser, the greater the transfer of heat from the condenser to the environment and, consequently, the greater the energy required for the evaporation of ammonia in the evaporator to occur. That is, it will be necessary to remove more energy in the form of heat from the refrigerated environment or the fluid to be cooled in the evaporator so that the ammonia leaves this component in the form of steam, thus increasing the efficiency of the refrigeration system.

To analyze the influence of the absorber temperature T_1 in the performance of the system, a constant temperature of the condenser was considered in the input parameters T_5 = 25 °C and P_1 = 300 kPa. Thus, by varying the temperature of the generator T_3 as a function of the COP of the system for different temperatures of the absorber, the behavior shown in Fig.6.

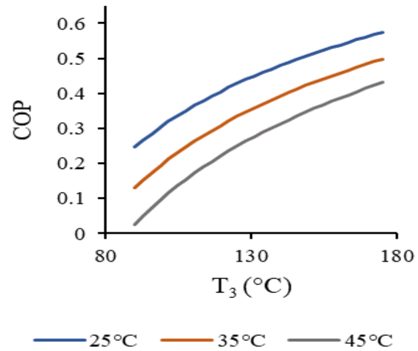

Figure 6. Generator temperature as a function of COP for different absorber temperatures.

Analyzing Fig. 6, it can be seen that the lower the temperature of the mixture that leaves the absorber, the higher the performance of the system. For T_1 = 25°C the system presented COP values between 0.25 and 0.58, as for T_1 = 45°C the COP value ranged between 0.03 and 0.43. This increase in the coefficient of performance for lower absorber temperatures is due to the fact that the amount of ammonia that can be dissolved in water is inversely proportional to the temperature in this component. That is, lower absorber temperatures indicate that there was greater heat transfer from the absorber to the cooling water of this component, thus resulting in greater absorption of ammonia in the water. This is observed in the value of the mass fraction of ammonia (x_1) of the mixture that leaves this component, which presents higher values at lower temperatures.

Regarding the influence of the evaporator temperature on the cycle performance, the low pressure value was used as inputs parameters of the model P_1 = 300 kPa, the temperature of the absorber, T_1= 20 °C and the temperature of the condenser, T_5 = 25 °C. Changing the generator temperature, T_3 as a function of the COP for different evaporator temperatures, the behavior described in Fig. 7.

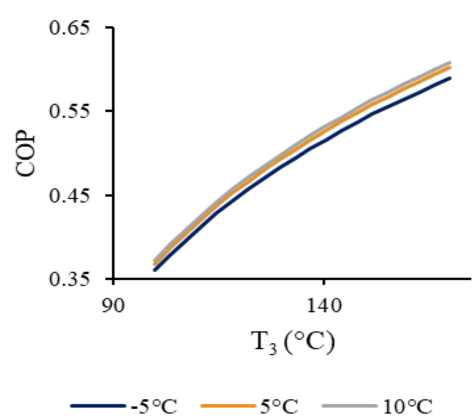

Figure 7. Generator temperature as a function of COP for different evaporator temperatures.

Analyzing the behavior of Fig. 7, it is noticed that the higher evaporator temperature showed higher COP values, ranging between 0.37 and 0.61. For lower temperatures, as in T_7 = -5 °C the COP varied between 0.36 and 0.59. When the ammonia vapor leaving the evaporator has higher temperatures, this indicates that there has been a greater heat transfer from the fluid to be cooled to the evaporator, that is, the greater the amount of heat removed from the fluid to be cooled so that the ammonia evaporates, consequently increasing the performance of the system.

For analysis of the Influence of pressure at the outlet of the absorber (P_1) on the COP of the system, the temperature of the generator T_3 was varied as a function of the COP of the system for different types of pressure P_1. With this, it was possible to investigate the impact of this parameter on the overall performance of the system. As initial parameters of model for this case, was defined as absorber temperature T_3 = 20 °C and for condenser temperature T_5 = 20 °C. The result of applying these parameters are shown in Fig. 8.

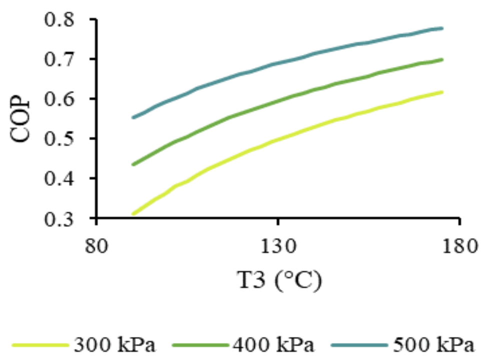

Figure 8. Generator temperature as a function of COP for different pressures.

Through Fig. 8, it is observed that the higher the value of the saturation pressure of the evaporator P_1, the higher the COP of the system, where for the pressure P_1 = 500 kPa, COP values between 0.55 and 0.77 were obtained. In this case, higher values of P_1 imply higher temperatures of the fluid at the outlet of the evaporator, that is, a greater amount of heat transfer was required from the fluid to be cooled to the evaporator. That is, the higher the pressure P_1 in the evaporator, the greater the amount of heat needed to evaporate the ammonia, increasing the efficiency of the cycle. Furthermore, the pressure P_1 it also influences the amount of ammonia that is dissolved in the water in the absorber. Greater values of P_1 imply higher amounts of ammonia mass fraction leaving the evaporator.

It is also important to mention that the greater the value of P_1, the greater the value of the specific volume of the liquid solution that leaves the evaporator and, consequently, the greater the drive power required by the pump. However, the mixture that passes through the pump is a liquid solution of NH_3H_2O, presenting much smaller values of specific volume if compared to the ammonia vapor. For this reason, the pump drive power for this type of refrigeration system is very low, having no relevant impact on the COP of the system. To prove this fact from the model developed, another value was calculated for the COP of the system, which does not take into account the power of the pump. Then, the temperature of the generator was varied T_3 as a function of the two performance coefficients, where COP_1 considers the pump power and COP_1 does not. The result of such an analysis is shown in Fig. 9

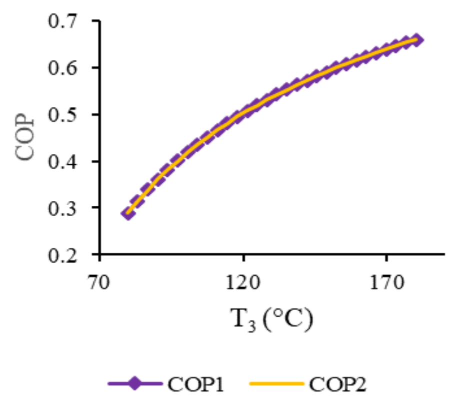

Figure 9. Effect of pump power on system COP.

As expected, there is no relevant difference when considering the pump drive power or not in the performance calculations of refrigeration systems by absorption.

The amount of ammonia mass fraction of the vapor leaving the generator/rectifier is an important factor in the cycle. Some absorption refrigeration cycles use a rectifier before the condenser precisely to remove any liquid existing in the fluid, causing only ammonia vapor to enter the condenser. In the mathematical model developed, this is represented by the fraction of dough at point 4 (x_4). To analyze the relevance that this property has on performance of the system, varied x_4 depending on the COP. The behavior obtained is shown in Fig 10.

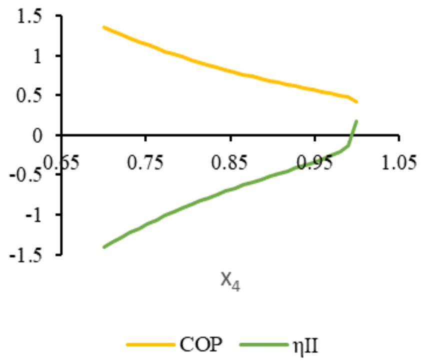

Figure 10. Influence of x_4 on system performance.

Through Fig. 10, it can be seen that the system has higher COP values when the ammonia mass fraction is further away from the unit. However, also plotting the exergetic efficiency curve, it is seen that for x_4 values lower than approximately 0.9995, the system presented negative exergetic efficiency, that is, the system loses more energy than the amount that enters the system, making the system impracticable.

For analyze the influence of the fluid temperature at the output of the generator T3 in relation to the exergetic efficiency of the system, a pressure P_1 = 350 kPa and the temperatures of the absorber and condenser of $T_1 = T_5$ = 30ºC were chosen as input parameters. Furthermore, it was considered that 10 kW is constantly supplied to the generator (Q_G = 10kW). The behavior of the exergetic efficiency of the system as a function of generator temperature is shown in Fig. 11.

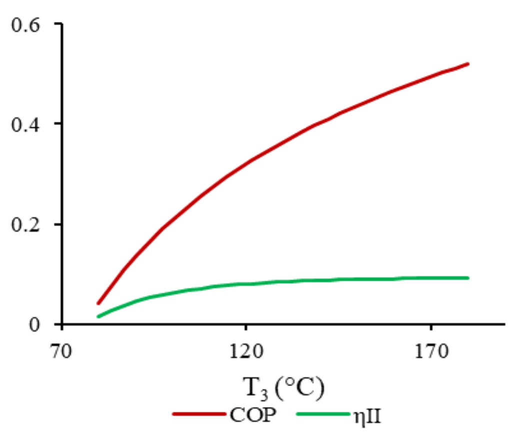

Figure 11. T_3 as a function of COP and exergetic efficiency.

From the behavior obtained in Fig. 11, it is noted that the higher the temperature T_3, the higher the corresponding value of the COP of the system. However, in relation to exergetic efficiency, even if higher values of T_3 result in a greater amount of steam of ammonia in the generator, there is also an increase in exergy losses both in the generator and in the absorber, reducing the exergetic efficiency of the system. It can be seen that from approximately T_3 = 140ºC, there is no significant variation in the second law efficiency, even for higher values of T_3, precisely due to the increase in exergy losses caused by the increase in temperature in this component. This increase in the destruction of exergy in the generator/rectifier as a function of the increase in temperature T_3 is shown in Fig. 12, where it can be seen that from approximately 112°C the destruction of exergy in this component increases significantly.

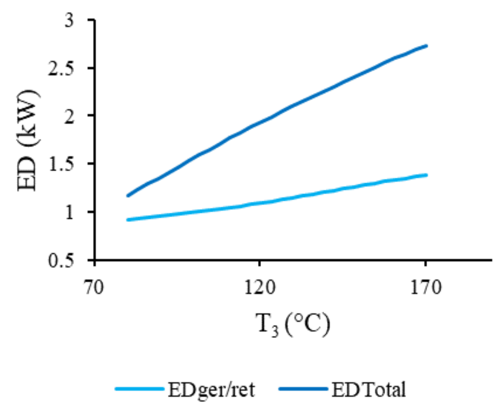

Figure 12. T_3 as a function of the exergy destroyed in the generator/rectifier and the total destroyed exergy of the system.

With this, it can be said that only a first law analysis, that is, through the behavior of the COP as a function of the generator/rectifier temperature, is not enough to correctly analyze the performance of a refrigeration system. It is also necessary to take into account, for a more consistent analysis, the rate of exergy destruction in each component and the exergetic efficiency of the system.

For obtain the influence of T_1 on the second law efficiency, P_1 = 500 kPa, T_5 = 25ºC was used as input parameters, and the heat transfer QG was kept constant for all cases in this section. The behavior obtained is shown in Figure 13, where the temperature of the generator T_3 was varied as a function of the exergetic efficiency for three values of temperature of the absorber T_1.

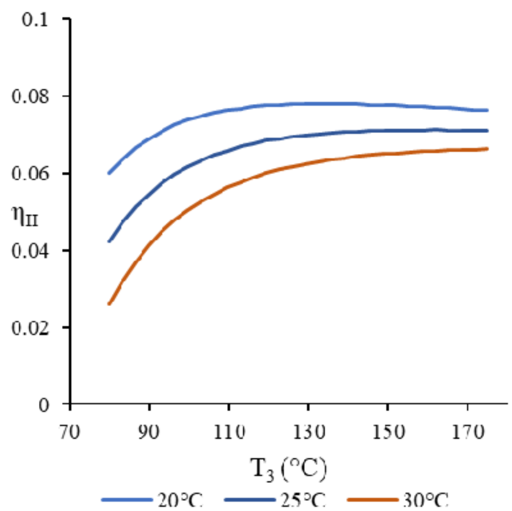

Figure 13. T_3 as a function of exergetic efficiency for different temperature values of the T_1 absorber.

From Figure 13, it is observed that the lower the absorber temperature, the higher the exergetic efficiency of the system, where for a temperature T_1 = 20 ºC the second law efficiency presented values between 0.0599 and 0.0762. This is explained by the fact that higher heat transfer in the evaporator results in higher temperature ammonia vapor entering the absorber. Thus, there is a greater heat transfer from the absorber for cooling water in this component so that the temperature of the mixture at the outlet of the absorber is decreased since in this component the temperature of the solution is inversely proportional to the amount of ammonia that can be dissolved in the water. Regarding the temperature of condenser T5, the following values were used as input parameters: P_1 = 300 kPa and T_1 = 30 ºC. With these parameters and varying the temperature of T_3 as a function of the exergetic efficiency, for different temperatures of T_5, the behavior is obtained as shown in Figure 14.

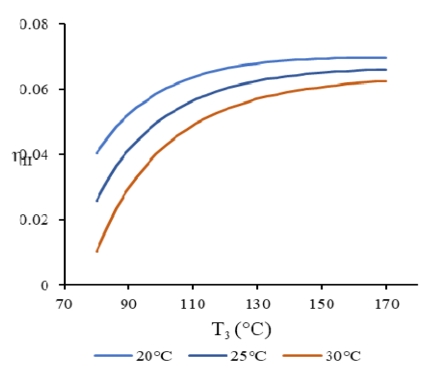

Figure 14. T_3 as a function of exergetic efficiency for different condenser temperatures T_5

Analyzing the behavior of the curves in Figure 14, it can be seen that the system presented higher values of exergetic efficiencies for lower values of condenser T_5 temperature, with values ranging between 0.0403 and 0.0691 for T_5 = 20 ºC. This is because higher condenser temperatures result in lower evaporator heat transfer rates as the liquid leaving the condenser will be at a higher temperature. Lower values for condenser temperature imply greater heat transfer in the evaporator, increasing the exergetic efficiency of the system. For analyze the influence of T_7 on exergetic efficiency, P_1 = 300 kPa, T_1 = 30 ºC and T_5 = 25 ºC were used as input parameters. The temperature of generator T_3 was then varied as a function of the exergetic efficiency of the system for different T_7 evaporator temperatures, as shown in Figure 15.

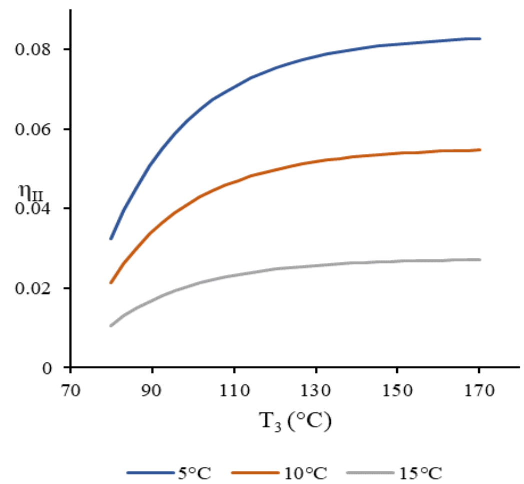

Figure 15. T_3 as a function of exergetic efficiency for different temperatures of the T_7 evaporator.

Analyzing Figure 15, it can be seen that the system has higher exergetic efficiency values when the temperature of the T_7 evaporator is lower, where the second law efficiency varied between 0.032 and 0.082 for T_7 = 5ºC and between 0.021 and 0.055 for T_7 = 10ºC. In the evaporator, lowering T_7 temperature has a greater impact on exergetic efficiency than increasing heat transfer Q_E. That is, the evaporator has greater cooling potential at lower temperatures.

The temperature in the generator T_3 was varied as a function of the exergetic efficiency for different values of heat transfer to the generator, as shown in Figure 16. The input parameters defined were P_1 = 300 kPA, T_1 = 20ºC and T_5 = 25ºC.

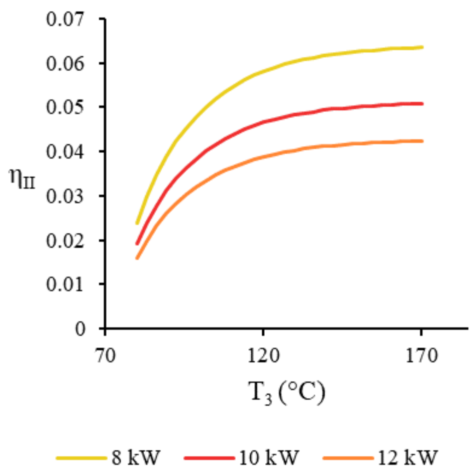

Figure 16. T_3 as a function of exergetic efficiency for different Q_G values.

Through the analysis of Figure 16, lower values of second law efficiency are observed for smaller values of Q_G, where for this case it presented values between 0.024 and 0.063 for Q_G = 8kW and for Q_G = 10 kW it presented values between 0.019 and 0.051. This happens because the more heat is supplied to the generator, the higher the temperature T_3 and the greater the exergy losses associated with both the generator and the absorber and condenser, reducing the exergy efficiency of the system.

For assessing the amount of exergy destroyed in each component, the input parameters in the model described in Table 03 were used. Applying Eqs. 19 – 25, the behavior illustrated in Figure 17 was obtained.

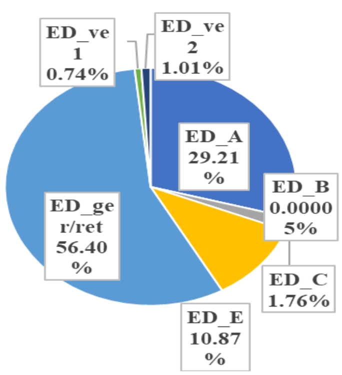

Figure 17. Share of exergy destruction of each component of the absorption refrigeration cycle.

The components that showed the highest rate of exergy destruction were the generator/rectifier set and the absorber, with 56.4% and 29.2%, respectively, of the total exergy destroyed in the system. These irreversibilities occur mainly due to the mixing process with greater temperature differences that takes place in these components. Knowing this, in terms of design improvement and optimization, these two components would require more attention in order to decrease the amount of total system exergy destruction, thus improving system performance.

In order to improve the performance of the refrigeration system model as a function of parameters such as temperature and pressure, values were sought to present exergetic efficiency optimizations for certain operating conditions. To calculate the maximum or minimum values of the parameters of this work, the Golden Section Search method, available at EES, was used. According to Bagheri et al. (2019), this method is more consistent than the quadratic approximation method, also available through the software. In addition, temperature and pressure values were sought that minimize the destruction of exergy in the generator/rectifier, sincewhich is the component with the highest exergy losses. For this, the Golden Section Search method, available at EES, was also applied.

Initially, a temperature value was sought in the T3 generator/rectifier that presented the highest exergetic efficiency with the input parameters described in Table 4.

Table 4. Model input parameters for exergetic optimization as a function of T_3.

Parameters	Symbol	Unit	Value
Pump Mass Flow		kg/s	0.01
Pressure at the outlet of the absorber		kPa	300
Absorber Temperature		ºC	20
Condenser Temperature		ºC	20
Heat transfer to the generator		kW	10
Mass fraction of ammonia at the generator/rectifier output		-	0.9996

Thus, T_3 is varied as a function of the second law efficiency in order to seek an optimal temperature value which results in the highest exergetic efficiency for these operating conditions, as shown in Figure 18.

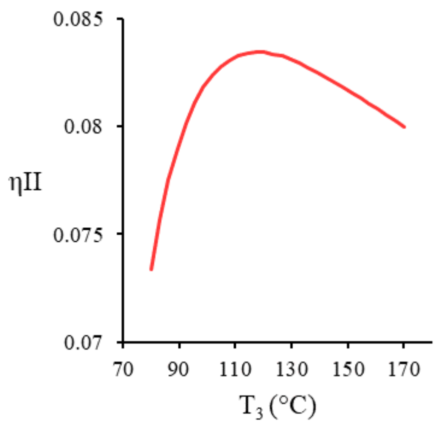

Figure 18. Exergetic optimization as a function of T_3 generator/rectifier temperature.

Note that the exergetic efficiency has an increasing behavior up to a certain temperature value and then starts to decrease with the increase of T_3. Therefore, it is possible to obtain a temperature value T_3 in which the system will present a value optimal exergetic efficiency. The values that optimize the model for this case were T_3 = 118.2ºC with an exergetic efficiency of 0.08344.

Optimization as a function of P_1, it was also sought, values of pressure P1 that resulted in greater exergetic efficiency in the developed model of cooling system. For this case, the input parameters described in Table 5 were used.

Table 5. Model input parameters for exergetic optimization as a function of P_1.

Parameters	Symbol	Unit	Value
Pump Mass Flow		kg/s	0.01
Condenser Temperature	T_5	ºC	30
Absorber Temperature	T_1	ºC	30
Heat transfer to the generator		kW	5
Mass fraction of ammonia at the generator/rectifier output	x_4	-	0.9996

Varying P_1 as a function of the second law efficiency, the behavior shown in Figure 19 is obtained.

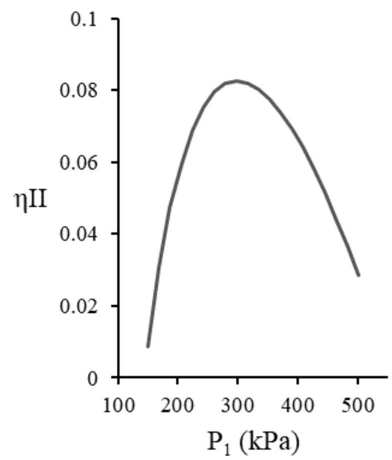

Figure 19. Exergetic optimization as a function of pressure at the outlet of the absorber P_1.

Analyzing Figure 19, it can be seen that between pressures of 150 kPa and approximately 297 kPa, there is an increase in exergetic efficiency. By increasing the pressure value, a maximum value for efficiency is reached and then this is reduced to the as P1 increases. The reduction in efficiency occurs because the system pressure in the output of the absorber is the same saturation pressure of the evaporator, that is, the increase of P1 results in the increase of the temperature of the evaporator T7, reducing the efficiency of second law. The optimal point found for this case was at P1 = 298.5 kPa and an exergetic efficiency of 0.08254.

Temperature and pressure values were obtained that minimize the exergy losses in the generator/rectifier, which was indicated by the exergy analysis as the component responsible for the greater destruction of exergy in the developed model. Minimizing this parameter is extremely important from the point of view of system performance, in which the generator/rectifier is the greatest source of irreversibility, as it presents a greater share of total system exergy destruction.

For this analysis, the input parameters shown in Table 6 were used, where the temperature of the T1 absorber was varied as a function of the exergy destroyed in the generator/rectifier, obtaining the behavior of Fig. 20.

Table 6. Model input parameters for calculating the minimum value of exergy destroyed in the generator/rectifier.

Parameters	Symbol	Unit	Value
Pump Mass Flow		kg/s	0.01
Condenser Temperature	T5	ºC	30
Pressure at the outlet of the absorber	P1	ºC	350
Heat transfer to the generator		kW	10
Mass fraction of ammonia at the generator/rectifier output	x4	-	0.9996

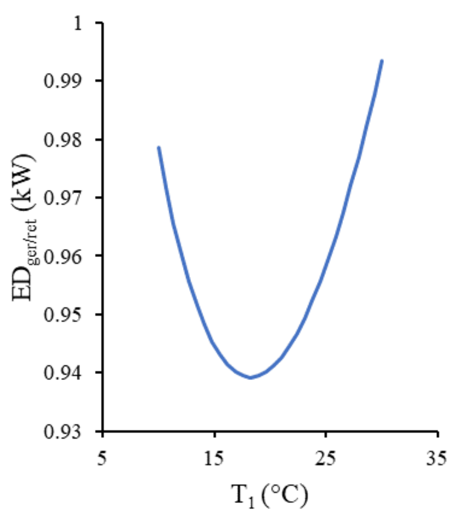

Figure 20. Reduction of exergy destroyed in the generator/rectifier as a function of absorber temperature.

From the behavior of the curve in Figure 20, it can be seen that there is a temperature value T1 where the value of exergy destruction in the generator/rectifier is the minimum possible. The minimum exergy destruction value obtained for this component was 0.9392 kW at T_1 = 18.3°C. By increasing the temperature of the absorber, the destruction of the exergy of the generator/rectifier also has its value increased, because the temperature of the solution that leaves the absorber is at a higher temperature, increasing the mixing losses in the generator/rectifier. Furthermore, it can be affirmed through the analysis of Figure 20, that solutions with a high concentration of ammonia also generate a greater destruction of exergy in the generator/rectifier. This is represented in the graph when the temperature values are less than 18.3ºC. Thus, it can be seen that for T_1 values <18.3ºC, the exergy destruction in the generator/rectifier increases, as well as the concentration of ammonia in the solution leaving the absorber. This occurs because the solution is making the absorber richer in ammonia, but at a low enough temperature to increase the losses in the generator/rectifier.

The minimum value of $ED_{ger/ret}$ as a function of P_1. Pressure at the outlet of the P1 absorber was also analyzed as a function of the exergy destruction in the generator/rectifier. For this, the values from Table 08 were also used, considering T1 = 20 ºC and varying P1 as a function of the exergy destroyed in the generator/rectifier, presenting the behavior described in Figure 21.

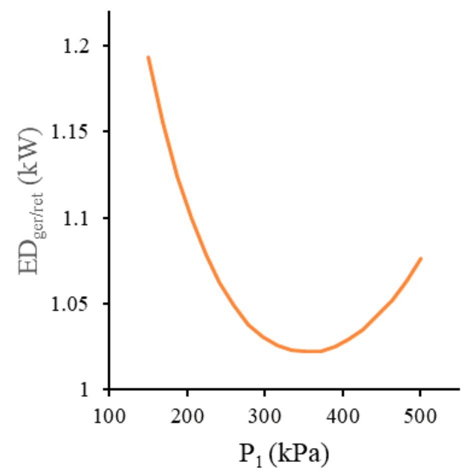

Figure 21. Reduction in the amount of exergy destroyed in the generator/rectifier as a function of the output pressure of the P_1 absorber.

Applying the Golden Section Search through the EES, the value of P1 = 353.5 kPa was determined, for which the exergy destruction presented a minimum value of 1.021 kW for this case. The behavior of the curve shown in Figure 21 is explained by the fact that the higher P1, the greater the absorption efficiency, that is, the greater the amount of ammonia in the solution that goes to the generator. However, the temperature of the absorber is constant, requiring greater heat transfer in the generator, increasing the losses in this component. And if the pressure drops below 353.5 kPa, the solution will get weaker and weaker in terms of ammonia concentration, also increasing the losses in the generator/rectifier.

CONCLUSIONS

The main conclusions of this article are summarized according to the specific objectives of this work, i) A steady-state mathematical model of an absorption refrigerator was developed to represent the operation of the main components of the refrigeration cycle as a function of the operating parameters, ii) The equation of the mathematical model using mass, energy and entropy balances was implemented in the Engineering Equation Solver (EES), iii) A parametric analysis of the system was performed, from which the importance of component temperatures as a function of COP and exergetic efficiency was studied. The impact of the pressure at the outlet of the absorber on the performance of the system was analyzed and also how the mass fraction of ammonia that enters the condenser must be greater than 0.9995, where values smaller than this resulted in negative exergetic efficiency of the system. In addition, the impact of the increase in temperature of the generator/rectifier on the destruction of exergy in the system was also shown, in which increasing this temperature increases the irreversible losses in this component, iv) An exergy analysis of the cooling system was also carried out, from which it was possible to assess that the generator/rectifier is responsible for approximately 56.4% of the total exergy destroyed in the system. Thus, the analysis indicated that the generator/rectifier is the most important component from the point of view of irreversibilities, with the greatest potential to improve system performance, v) and The exergetic optimization of the absorption refrigeration system was carried out under certain operating conditions, where an exergetic efficiency of 0.08344 was obtained for T_3 = 118.2 ºC, as well as an exergetic efficiency of 0.08254 for P_1 = 298.5 kPa . The values of T_1 and P_1 were obtained that resulted in the lowest possible value of exergy destruction of the generator/rectifier in certain cases, where T_1 = 18.3ºC presented $ED_{ger/ret}$ = 0.9392 kW and P_1 = 353.5 kPa resulted in $ED_{ger/ret}$ = 1.021 kW.

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