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Exergetic Study of a Ammonia Vapor Refrigeration System: Effect of the Compressor Efficiency, Compressor Inlet Pressure and Temperature

[1]Roberto J. Carrascal Berdugo, Eugenio Yime Rodriguez[2], Guillermo Valencia Ochoa[3]

[1]Research Assistant in Mechanical Engineering/ Faculty of Engineering/Universidad Del Atlántico, Colombia.
[2]Associate Professor in Mechanical Engineering Program/ Faculty of Engineering/ Universidad Del Atlántico, Colombia.
[3]Assistant Professor in Mechanical Engineering Program/ Faculty of Engineering/ Universidad Del Atlántico, Colombia.

Abstract : This article presents the exergistic analysis of a refrigeration cycle, by means of which it is sought to analyze the behavior of the exergy destroyed by each component of the system, besides the analysis of the exergistic efficiency, the COP and the work carried out by the compressor, by means of the comparison of these results for different adiabatic coefficients of the compressor. The study was conducted for a refrigeration cycle, which is composed of a compressor, condenser, heat exchanger, valve and evaporator. In which the compressor inlet pressure varied between 100kPa and 600kPa. Being in 500 kPa, the value of the pressure where a greater exegetical efficiency and a greater operation coefficient is produced. And in addition, the initial system temperature from 240K to 380K. Being in 350K, the temperature value where there is more exergetic efficiency and less exergy destroyed by the compressor.
Keywords : Compressor, Exergetic efficiency, Coefficient of operation.

1. Introduction

Refrigeration cycles are an application of thermodynamic cycles widely used in homes and industry to maintain food comfort or conservation. The increase in the use of these has generated the need for improvements in their performance, which has motivated a great deal of research with which great advances have been achieved in recent years in terms of energy and energy efficiency and energy efficiency[1].

For a single-stage absorption cycle, improvement has been sought by carefully designing the geometry of the distillation column within the generator and it has been proven that the reorganization of the flow lines of the liquid solution is more effective compared to the flow lines of the refrigerant[2].Likewise, we have

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experimented with absorption systems coupled to solar systems to seek an increase in efficiency[3-5]. With this technology a cooling system with three ejectors and three different design configurations that work with ammonia water was analyzed, achieving a greater cooling effect in the cycle of the evaporation tank with three ejectors[6]. Another improvement with the help of solar systems was the storage of thermal energy[7].

Similarly, the study of exergy in these cycles has been of great importance[8-10]. With this premise in mind, an analysis of a water and lithium bromide absorption cooling system has been carried out, in which the effect of the input temperature on the energy and exergetic efficiency was examined[11]. Similarly, this criterion was also applied in Kalina power cycles[12,13], to power combined cycle systems with absorption[14].

Taking into account the above, the contribution of this work is directed towards the realization of an exergetic analysis in which the exergetic efficiency and the destroyed exergy of a compression refrigeration cycle will be studied. For this purpose, parameters such as the efficiency of the components of the cycle and their initial temperatures will vary in order to evaluate each of the operating points of the cycle.

2. Methodology

In order to obtain the data for the case studies, a spreadsheet was used, in which by varying the parameters of the input data, different results were obtained with respect to the exergetic efficiency of the cycle, the work carried out by the compressor, the coefficient of performance of the cycle (COP) and the exergy destroyed by it.

2.2. Description of the Cooling Cycle

An ideal vapour compression refrigeration cycle was modified to include a counterflow heat exchanger, as shown in Figure 1. The ammonia leaves the evaporator as saturated vapour at 1.0 bar and is heated to constant pressure at 5 ° C before entering the compressor after isentropic compression at 18 bar, the refrigerant passes through the condenser, exiting at 40 °C, 18 bar. The liquid then passes through the heat exchanger, entering the expansion valve at 18 bar. The mass of the refrigerant flow rate is 12 kg/min.

The following operating conditions were considered for the analysis of the refrigeration cycle operating with ammonia:

- The cycle operates in a stationary state.
- The changes in kinetic and potential energy in the fluid were neglected.
- The compressor operates isentropically.
- The working fluid is ammonia.
- The state at the output of the condenser is in compressed liquid.

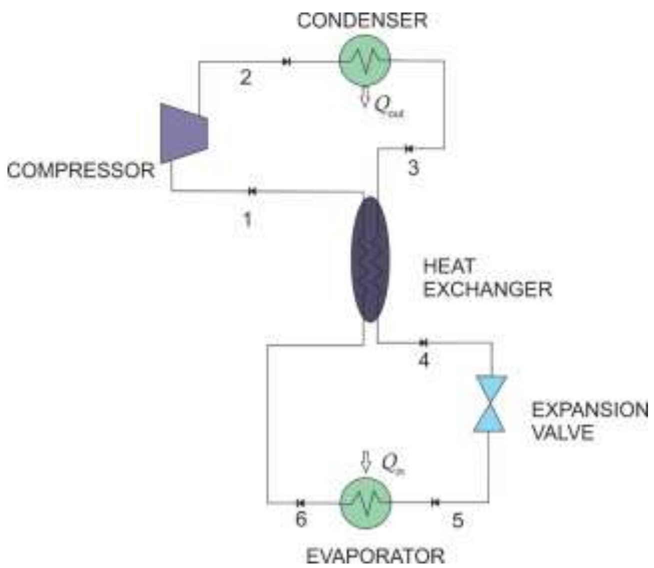

Figure 1: Gas Cycle Diagram.

The input data of the refrigeration cycle is shown in table 1. In which he observes that pressure 2 is where there is the highest enthalpy for a pressure of 1800 kPa.

Table 1. Thermodynamic properties of each state.

Status	Pressure [kPa]	Temperature [K]	Entalpy (*h*) *[kJ/kg]*	Entropy (*s*) *[kJ/kg-K]*
1	100	278,15	1483,25	6,1676
2	1800	-	2025,54	6,1676
3	1800	313,15	371,35	1,3569
4	1800	-	286,51	9,6009
5	100	-	286,51	9,6009
6	100	239,55	1398,41	6,1616

Table 2. Factors found for the cycle.

Factors	was calculated
A	COP
B	W comp
C	β

On the other hand, the following factors were determined for the cooling cycle with ammonia as the working fluid, shown in Table 2.

This table shows the cycle cooling coefficient (COP), the work done by the compressor (W comp) and the coefficient of performance (β), which are the factors found from the fundamental equations. In addition, the enthalpies of the compressor, evaporator, condenser, valve and heat exchanger inputs and outputs were manually determined to calculate the input and output heat, cooling capacity, compressor power and coefficient of performance, using thermodynamic tables.

2.1 Fundamental equations

A set of fundamental equations were used to find the results obtained for the realization of the case studies, which have been studied in the literature.

The cooling capacity in tons of cooling ($\dot{Q}_{in}$), was determined for the evaporator by means of an energy balance as shown in equation 1

$$\dot{Q}_{in} = \dot{m}(h_6 - h_5),(1)$$

Where$\dot{Q}_{ent}$is the cooling capacity and dot$\dot{m}$the mass flow of the cooling cycle.

The compressor power input($\dot{W}_{comp}$), was determined for the compressor by means of an energy balance, resulting in equation 2

$$\dot{W}_{comp} = \dot{m}(h_2 - h_1),(2)$$

The coefficient of performance (β), was determined by the ratio of the evaporator enthalpies to the compressor enthalpies as shown in equation 3

$$\beta = (h_6 - h_5)/(h_2 - h_1),(3)$$

The values of the destroyed exergy per component were obtained from an exergy balance sheet, applying equation 4

$\dot{X}_{in} - \dot{X}_{out} - \dot{X}_{destroyed} = \frac{dX_{system}}{dt}$,(4)

The amount of exergy entering a stationary flow system in all forms (heat, work, mass transfer) must be equal to the amount of exergy leaving plus the exergy destroyed, therefore, the exergy balance is obtained by equation 5

$\dot{X}_{in} - \dot{X}_{out} = \dot{X}_{destroyed}$,(5)

The exergy destroyed by the heat exchanger ($\dot{X}_{dest.int.calor}$), was determined from an exergy balance resulting in equation 6

$\dot{X}_{dest.int.calor} = \dot{m}\{[(h_6 - h_1) - T_o(s_6 - s_1)] + [(h_3 - h_4) - T_o(s_3 - s_4)]\}$, (6)

WhereT_o, the initial temperature of the refrigeration cycle.

The exergy destroyed by the capacitor ($\dot{X}_{dest.cond}$), was determined from an exergy balance, using equation 7

$\dot{X}_{dest.cond} = \dot{m}[(h_2 - h_3) - T_o(s_2 - s_3)] - \left(1 - \frac{T_o}{T_{cond}}\right)\dot{Q}_{cond}$, (7)

WhereT_{cond}, is the condenser temperature.

The exergy destroyed by the evaporator ($\dot{X}_{dest.evap}$), was determined from an exergy balance sheet, using equation 8

$\dot{X}_{dest.evap} = \dot{m}[(h_5 - h_6) - T_o(s_5 - s_6)] - \left(1 - \frac{T_o}{T_{evap}}\right)\dot{Q}_{evap}$, (8)

WhereT_{evap}, is the evaporator temperature.

The exergy destroyed by the compressor ($\dot{X}_{dest.comp}$), was determined from an exergy balance sheet, using equation 9

$\dot{X}_{dest.comp} = \dot{W}_{comp} + \dot{m}\{[(h_1 - h_2) - T_o(s_1 - s_2)]\}$, (9)

Where$\dot{W}_{comp}$, is the work of the compressor obtained by equation 10.

The exergy destroyed by the valve ($\dot{X}_{dest.valv}$), was determined from an exergy balance sheet, using equation 10

$\dot{X}_{dest.valv} = \dot{m}[(h_4 - h_5) - T_o(s_4 - s_5)]$, (10)

To determine the exergetic efficiency(η_{exe}),two equations can be used. The first way to determine the exergetic efficiency is through equation 11 where the total destuida exergy is given by equation 12 and the one spent by equation 13

$\eta_{exe} = 1 - \frac{x_{destroyed,total}}{x_{used}}$,(11)

$x_{destroyed,total} = \sum x_{destroyed\ by\ component}$, (12)

$X_{used} = X_{cond} + X_{evap} + \dot{W}_{comp}$, (13)

3. Results and Discussion

For Mass Flow and Input data the rest of the properties are obtained using a scheduled spread sheet and their results are shown in Table 3.

Table 3. Thermodynamic properties of each state.

Status	**Pressure [kPa]**	**Temperature [K]**	**Entalpy (*h*) *[kJ/kg]***	**Entropy (*s*) *[kJ/kg-K]***
1	100	278,15	-2725,7414	12,1341
2	1800	522,42	-2173,6488	12,1342
3	1800	313,15	-3818,3157	10,0575
4	1800	297,65	-3897,3161	9,6009
5	100	239,44	-3897,3161	7,2552
6	100	239,55	-2804,7418	11,8284

The case studies were carried out using these properties. It is observed that state 2 has the highest temperature and that the entropy of state 1 and 2 are the same.

Tables 4 and 5 show the heat of the compressor and evaporator and their temperatures, as well as the work carried out by the compressor and the initial temperature at which the cycle operates.

Table 4. Evaporator and condenser heat and work done by the compressor.

Heat and work *[kW]*	
Q evap, entrada	218,5149
Q cond,salida	328,9334
W comp, entrada	110,4185

Table 5. Initial cycle temperature (To) and evaporator and condenser temperature.

Temperatures*[K]*	
To	239
T evap	239,5
T cond	239

Table 4 shows the heat of the evaporator and the compressor, respectively, and the work of the compressor, obtained with the use of the spreadsheet, which were determined by means of the fundamental equations, previously written.

Table 5 shows the temperatures given by the spreadsheet, by means of which the exertions destroyed for each component were calculated using the fundamental equations previously written.

On the other hand, using the data obtained, the exergy destroyed by each component was calculated, as shown in figure 2.

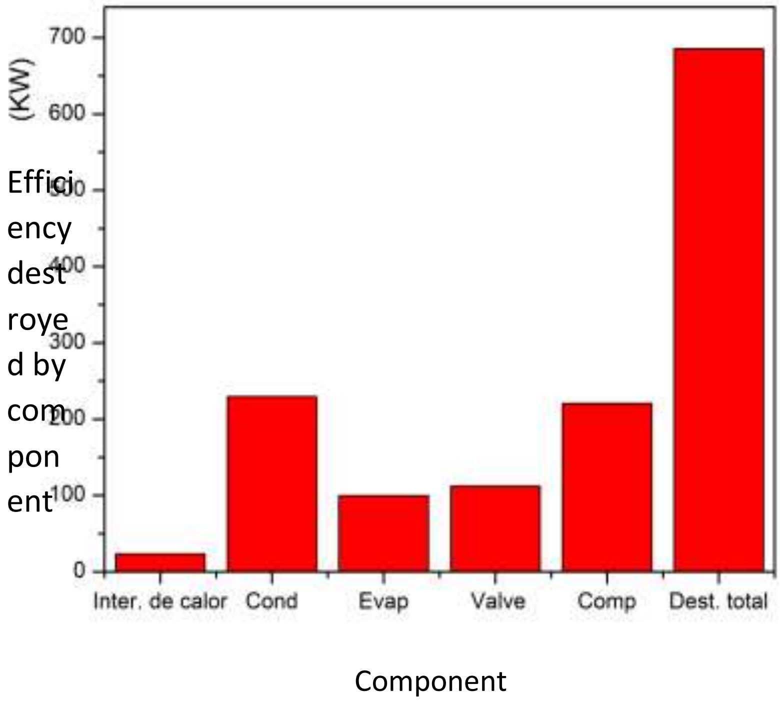

Figure 2. Exergy destroyed by each component.

The exergy destroyed by each component is observed, it being noted that the exergy destroyed by the condenser and the compressor are almost equal, in the same way the exergy destroyed by the valve and the evaporator are almost similar while the exergy destroyed by the heat exchanger is the smallest of all the components.

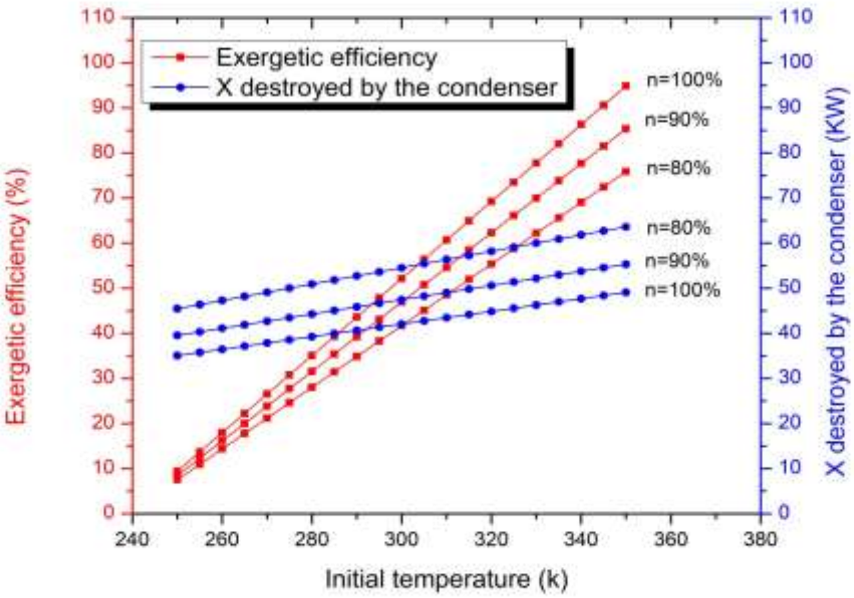

Figure 3. Exergetic efficiency and Exergy destroyed by the compressor as function of the Initial temperature.

In Figure 3, we analyzed how the exergy destroyed by the condenser and the initial temperature of the cycle affect the exergetic efficiency of the refrigeration cycle. Varying the initial temperature (To), from 240 K to 380 K, analyzing the destroyed exergy from 0 KW to 110 KW and with a range for the exergetic efficiency from %0 to 100%, the following results were obtained.

The adiabatic efficiency of the compressor was varied for 80, 90 and 100%, studying the behaviour of the exergetic efficiency and the exergy destroyed by the condenser, for a compressor that is more efficient than another one.

It is observed that the exergetic efficiency increases as the initial temperature increases and in the same way that the exergy destroyed by the condenser increases as the initial temperature of the cycle increases.

As is well known, for a compressor with an adiabatic efficiency of 100%, the destroyed exergy decreases and the exergetic efficiency increases.

On the other hand, in figure 4, it was analyzed how the change in the compressor inlet pressure (Pressure 1) affects the compressor work and the operating coefficient (COP). Varying the compressor inlet pressure from 0 KPa to 700 KPa and analysing the behaviour of the work carried out by the compressor from 0 KW to 650 KW and varying the COP from 0 to 7.

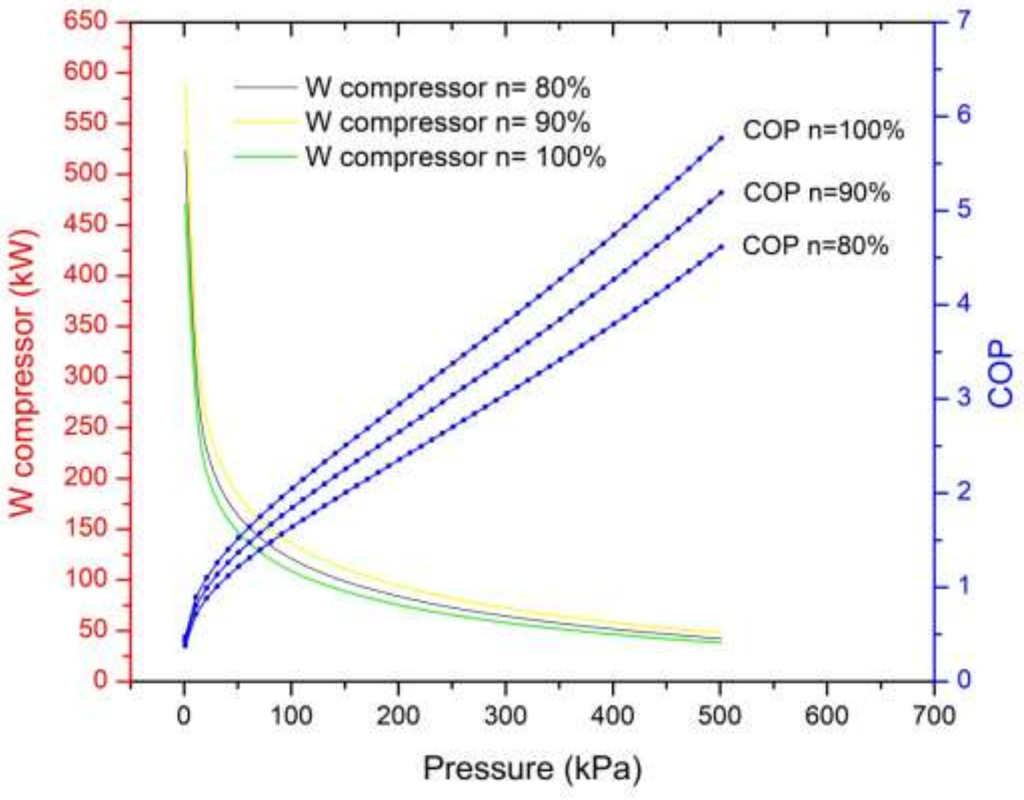

Figure 4: COP and Compressor energy consumption as function of the Compressor inlet pressure.

The adiabatic efficiency of the compressor was varied for 80, 90 and 100%, studying the work behavior of the compressor and the COP, for a compressor that is more efficient than another and the following results were obtained.

It can be seen that the work carried out by the compressor increases gradually as the inlet pressure to the compressor decreases and that for a compressor with an adiabatic efficiency of 100% more work is obtained.

Similarly, for a compressor with 100% efficiency, a higher COP value is obtained. Also, in figure 5, we analyzed how the change in the compressor inlet pressure affects the exergetic efficiency and COP. Varying the pressure from 0 KPa to 600 KPa, analyzing the behavior of the exergetic efficiency from 0% to 160% and studying the COP from 0 to 10, the following results were obtained.

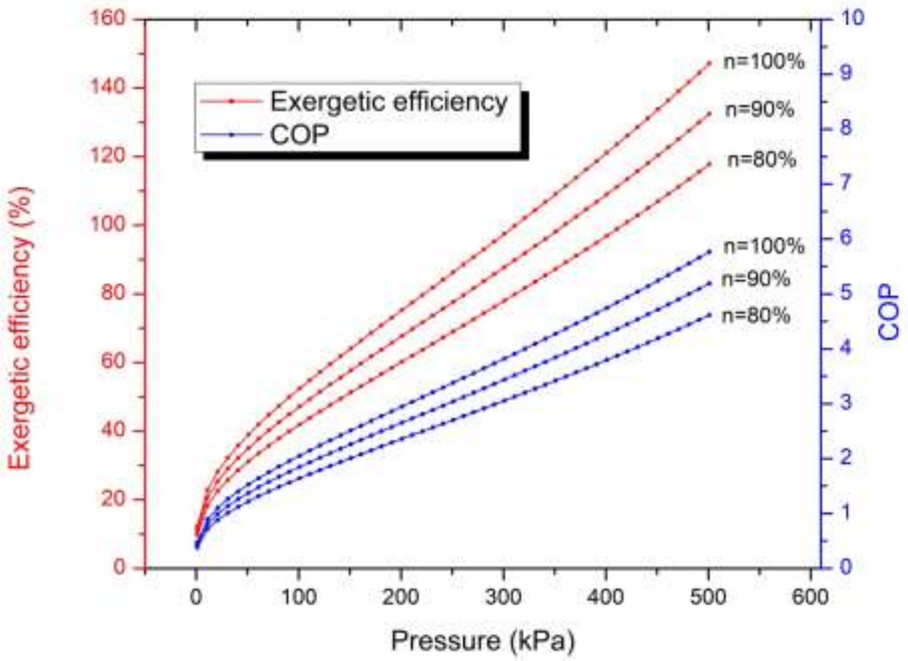

Figura 5: Exergetic efficiency and COP as function of the inlet compressor pressure.

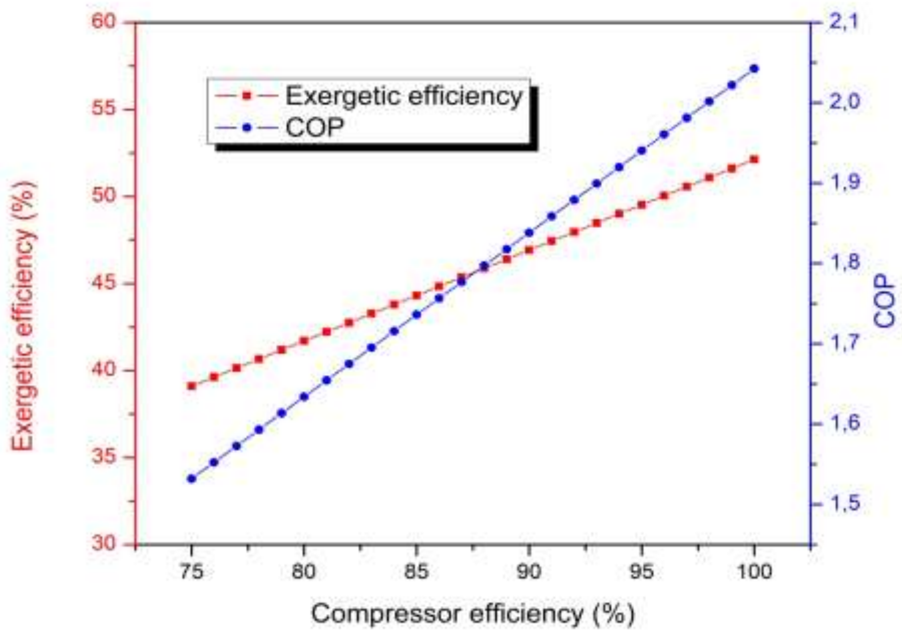

Figura 6: COP and Exergetic efficiency as function of the compressor efficiency.

The adiabatic efficiency of the compressor was varied for 80, 90 and 100%, studying the behavior of the exergetic efficiency and the COP. It can be seen that for a compressor with 100% efficiency, higher exergetic efficiency and higher COP.

Finally, in figure 6, the relationship between the coefficient of performance (COP) and the exergetic efficiency and the efficiency of the compressor was analyzed and the following results were obtained.

The relationship between POPs and exergetic efficiency is observed. It is notorious that for a compressor that works at 100% of its adiabatic efficiency, the exergetic efficiency of the cycle will be higher (53%), and that we will have a COP of 2.05.

4. Conclusions

In conclusion, it was possible to observe how, as the inlet pressure to the compressor increases, for different values of adiabatic efficiency of the compressor, the exergy destroyed by it for an efficiency of 100% decreased. In the same way, for a compressor with this aidabatic efficiency, the exergetic efficiency of the cycle is also improved and the operating coefficient is increased. Due to these efficiency gains, the results of this study are expected to serve as the basis for further research into cooling cycles that operate with ammonia as the working fluid.

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