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EXERGETIC ANALYSIS OF A R717/R744 CASCADE REFRIGERATION SYSTEM

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Abstract. *There is a growing search for the development of technologies that operate with the highest efficiency level. At the same time, it is desirable that these technologies have the lowest operational cost. Inside this context are the cascade refrigeration systems that use refrigerants R-717 and R-744. Their ideal operational condition is the maximum efficiency combined with the minimum cost. However, in general, increasing the efficiency implies increasing the cost. Thus, the objective of this study is to investigate the relationship between the maximum exergy efficiency and the minimal operating cost for a R717/R744 cascade refrigeration system. The methodology consists on the development of a exergetic model to describe the behavior of the system. The model includes equations about the operational cost and the exergetic efficiency, both used in the simulations to maximize the exergy efficiency and minimize operating cost. The results show that, for the system, the maximization of exergy efficiency provides the minimum operational cost. Analogously, the minimization of cost results in maximum efficiency. Based on the results, it is conclusive that, for the operation of a R717/R744 cascade refrigeration systems, the maximum exergetic efficiency and minimum operational cost occur simultaneously.*

Keywords: *Refrigeration, Simulation, Thermodynamics, Exergetic Analysis, Operational Cost.*

1. INTRODUCTION

Food preservation has always been one of the main dilemmas of humanity. Currently, there are numerous technologies that can overcome this adversity. Among them are residential refrigerators and cold rooms whose basic operational principles are based on thermodynamic cycles. However, in the early days of mankind, food preservation was done through using ice applied directly on the food. This technique, known as natural cooling, caused the reduction in food temperature (cooling effect) and delayed the growth of microorganisms, responsible for food decomposition.

With the technological advancement provided by the second industrial revolution that occurred in the mid 19th century, they were developed the first refrigeration equipments with industrial applications and thus began the artificial cooling age (Briley, 2004). The main equipments developed operated according to thermodynamic cycle by mechanical vapor compression and they utilized ammonia (NH_3) as the primary refrigerant. This substance was and remains widely used due to its excellent properties of heat exchange but, today, its toxicity limits its application to industrial systems (Çengel and Boles, 2006).

Subsequently, the need for obtaining lower operating temperatures and consequently increase cooling effect forced the development of new technologies to meet this demand. Among those technologies are the cascade cooling systems. Basically, these cooling systems are composed of two vapor compression refrigeration cycles interconnected by an intermediate heat exchanger. Thus, the increase in cooling effect is achieved by reducing the compression power input in each one and therefore it achieves lower operational temperatures. Typically, these systems employ two separate refrigerants, one for each thermodynamic cycle of the system.

Parallel to the technological development of refrigeration systems, important events happened in recent decades of human history, which have impacts reflected today. Among them are the energy crisis provided by the oil crisis, in the begining of 1970s. This conjecture demanded that companies must develop technologies more energy efficiency.

Additionally, in the late 1980s and early 1990s, it happened the phenomenon of globalization responsible for increase the competitiveness among companies, especially in economic field. In this way, companies wanted to increase their profits by reducing their operating costs. So, this scenario consisted by energy and economic factors extends to the present days and covers the current technologies including cascade refrigeration systems used in cold rooms. Thus, companies have sought to develop refrigerators which operate simultaneously with the highest level of energy efficiency and the lowest operational cost. Thus, this work carries out an exergetic analysis about a cascade refrigeration system which uses the pair CO_2 (R-717) and NH_3 (R-744) as refrigerants.

In general, it is known that increase in system efficiency often entails increasing in the costs. Most often it is due to using resources to develop new technologies or improve the existing one. Regarding the cost, there are three possibilities associated with cascade cooling systems. They are: acquisition, operational and the environmental one, related to using refrigerants that cause environmental impacts.

Then, in the present work, an exergetic model was developed in order to determine an optimum operational condition for a cascade cooling system combining high efficiency and low operational cost. In this sense, the objective of this article is to investigate the relationship between the maximum exergy efficiency and the minimal operating cost for a cascade cooling systems that uses R-717 and R-744 as refrigerants.

2. LITERATURE REVIEW

Basically there are three types of studies or articles. They are: experimental analysis, theoretical or computational models and studies that combine experimental analysis and computational models with economic models. About the first category, experimental analysis, they were analyzed the papers published by da Silva et al., 2011, Dopazo and Fernandez-Seara (2011), Sans-Cock et al., 2014 and Bingming et al., 2009.

da Silva et al., 2011 did an experimental analysis comparing the performance of three cooling systems applied in refrigerator sector. They were compared a CO_2 and a R404A cascade refrigeration system and two simple systems, one of them of R22 and the other using R404A. The results of this research show that CO_2 and R404A cascade system has a better performance than the others two systems. The other authors, Dopazo and Fernandez-Seara (2011), Sans-Cock et al., 2014 and Bingming et al., 2009 evaluated experimentally the performance of a CO_2 and NH_3 cascade refrigeration system according to different parameters such as condensation and evaporation temperatures.

In relation to theoretical or computational studies, they were analyzed those published by Lee et al., 2006, Getu and Bansal (2008), Dopazo et al., 2009, Dokandari et al., 2014 and Dubey et al., 2014. These authors developed models consisting mainly by equations related to mass conservation, energy, exergy and thermodynamic properties used to describe system behavior. Dubey et al., 2014 studied a CO_2 and propylene (R744-R1270) cascade system while others authors modeled a CO_2 and NH_3 system. Theses studies have sought to determine the conditions of optimum efficiency of a cascade cooling system according to parameters such as evaporation and condensation temperatures. Overall, the results of these studies show efficiency values according to a certain combination of those parameters.

Finally, related to thermoeconomic models, they were studied those published by Rezayan and Behbahaninia (2011), Mumanachit et al., 2012 and Aminyavari et al., 2014. These authors included thermodynamic models to describe a CO_2 and NH_3 cascade cooling system behavior and also assessed the costs related to them. Among the analyzed costs are the equipment acquisition, maintenance and operation beyond the environmental one. To address this problem, involving simultaneously efficiency and cost, the authors used special tools such as genetic solution algorithm and multi-objective optimization. Additionally, Zhang and Xu (2011) and Jain et al., 2015 also analyzed energy, exergy and costs related to refrigeration systems. As a general behavior, the results of all theses studies present a point of system that combines efficiency and cost due to a certain parameters combination such as evaporation and condensation temperatures. These models, particularly those presented by Rezayan and Behbahaninia (2011) and Aminyavari et al., 2014 are the basic models adopted to develop this article.

3. MATHEMATICAL MODEL

This section is divided into three parts. The first part describes a cascade refrigeration cycle. The second one covers the basic energy and exergy equations used to model the system. Finally, in the third part is presented the economic model used to analyse a cascade refrigeration system.

3.1 Describing cascade cooling system

The cascade cooling system studied consists of two thermodynamic cycles, one of them using CO_2 as refrigerant and the other using NH_3. The cycle that uses NH_3 is defined as high temperature circuit or high pressure zone and the cycle that uses CO_2 is defined as low temperature circuit or low pressure zone. Each cycle operates according to specific thermodynamic states that occur in each equipment of the cycle.

Each cycle consists of a heat exchanger, an expansion valve and a compressor. In NH_3 cycle the heat exchanger operates as a condenser, rejecting heat from the refrigerant to the external environment. Similarly, in CO_2 cycle the heat exchanger acts as an evaporator, removing heat from the external environment and providing it to the refrigerant and thus producing the cooling effect. Additionally, linking the two cycles, there is an intermediate heat exchanger named cascade one, which acts as an evaporator to NH_3 cycle and as a condenser to CO_2 circuit. The fluids, NH_3 and CO_2, seep in cross flow inside this heat exchanger.

The arrangement of the equipments in the cycle can be seen schematically in Fig. 1. Subsequently, they are described its basic operational principles.

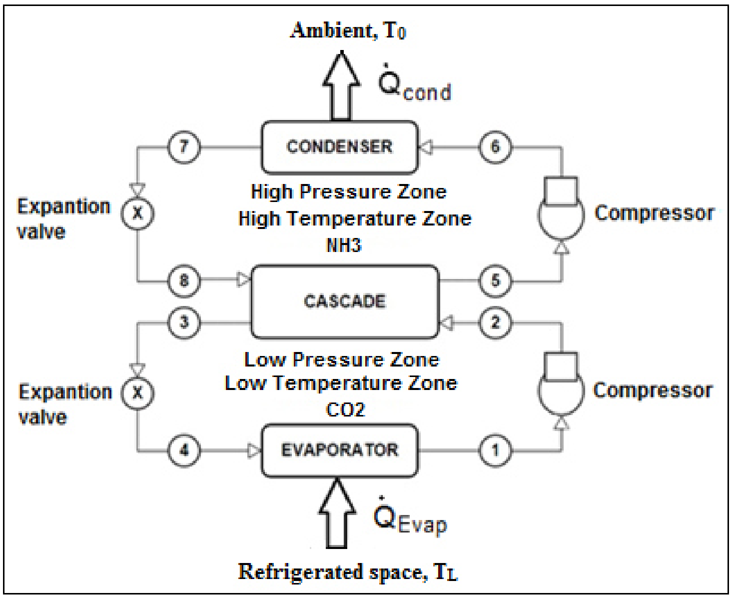

Figure 1. Diagram of a cascade refrigeration system

Initially, in low temperature circuit, point 1, CO_2 is as saturated steam at $T_{evap,CO2}$ (CO_2 evaporation temperature). Similarly, in high temperature circuit, NH_3 is as saturated vapor (point 5) at $T_{evap,NH3}$ (NH_3 evaporation temperature). From there, CO_2 leaves the evaporator and is directed to the compressor where it is compressed until superheated vapor state (point 2). Similarly, point 5, in high temperature circuit, NH_3 is compressed until superheated vapor state (point 6).

Continuing in low pressure zone, CO_2 leaves the compressor as superheated steam and it is directed to the cascade heat exchanger where it seeps in cross flow with NH_3 that enters from the other side of the circuit (point 8). This flow causes heat transfer from CO_2 to NH3. So, CO_2 leaves cascade heat exchanger as saturated liquid (point 3) at $T_{cond,CO2}$ (CO_2 condensation temperature). Related to NH_3, this fluid absorbs heat from CO_2 at $T_{evap,NH3}$, and passes from state 8 to state 5. The processes that happen in cascade heat exchanger occur at constant pressure and the temperature difference between $T_{cond,CO2}$ and $T_{evap,NH3}$ is named ΔTcascade.

Further, in low temperature circuit, CO_2 that leaves the cascade heat exchanger is throttled isoentalpically until its pressure reaches the value required in the evaporator (state 4). Finally, ending the process in low pressure zone, CO_2 passes from state 4 to state 1, evaporating and removing heat from the external environment and consequently cooling the conditioned space.

Meanwhile, in high temperature circuit, NH_3 that passed through the compressor will release heat in the condenser to the outside environment. This process occurs at constant pressure and NH_3 leaves the condenser as saturated liquid (point 7) at $T_{cond,NH3}$ (NH_3 condensation temperature). Closing the cycle, NH_3 as saturated liquid is isoentalpically throttled in expansion valve until its pressure reaches the value required in cascade heat exchanger (point 8).

3.2 Energy and exergy analysis

This part of the article is devoted to introduce the energy and exergy equations used to model the cascade cooling system. The main considerations used to model the system are:

- There is no pressure loss in refrigerant lines;
- There is no refrigerant overheating in evaporator outlet;
- There is no refrigerant subcooling in condenser outlet;
- There is no heat transfer in compressors and expansion devices;
- Are disregard power input fans in evaporator and condenser.

First, related to energy equations, the amount of energy absorbed by evaporator or the cooling load served by the system is calculated by Eq. (1).

$$\dot{Q}_{Evap} = \dot{m}_{CO_2}\left(h_1 - h_4\right) \tag{1}$$

The value of isentropic efficiency for both compressors are calculated by Eq. (2) and Eq. (3) presented by Dopazo et al., 2009.

$$\eta_{isent,CO_2} = 1 - (0{,}04(\frac{P_2}{P_1})) \tag{2}$$

$$\eta_{isent,NH_3} = 1 - (0{,}04(\frac{P_6}{P_5})) \tag{3}$$

To determine the power input in each compressor was applied the first law of thermodynamics to a control volume, resulting in Eq. (4) and Eq. (5).

$$\dot{W}_{Comp,CO_2} = \dot{m}_{CO_2} \frac{\left(h_1 - h_4\right)}{\eta_{isent,CO_2}} \tag{4}$$

$$\dot{W}_{Comp,NH_3} = \dot{m}_{NH_3} \frac{\left(h_6 - h_5\right)}{\eta_{isent,NH_3}} \tag{5}$$

Similarly, it was applied the first law of thermodynamics for each expansion valve and they were found Eq. (6) and Eq. (7).

$$h_3 = h_4 \tag{6}$$

$$h_7 = h_8 \tag{7}$$

Finally, it was applied the first law of thermodynamics in cascade heat exchanger, in condenser and to the cycle respectively and they were obtained Eq. (8), Eq. (9) and Eq. (10) .

$$\dot{Q}_{cascade} = \dot{m}_{CO_2}\left(h_2 - h_3\right) = \dot{m}_{NH_3}\left(h_5 - h_8\right) \tag{8}$$

$$\dot{Q}_{Cond} = \dot{m}_{NH_3}\left(h_6 - h_7\right) \tag{9}$$

$$\dot{Q}_{Cond} = \dot{Q}_{Evap} + \dot{W}_{Comp,NH_3} + \dot{W}_{Comp,CO_2} \tag{10}$$

In addition, the coefficient of performance of a cascade cooling system is defined by Eq. (11).

$$COP = \frac{\dot{Q}_{Evap}}{\dot{W}_{Comp,NH_3} + \dot{W}_{Comp,CO_2}} \tag{11}$$

The exergy destroyed in each equipment of the system is calculated by equations numbered from 12 to 18.

$$\dot{E}_{D,Evap} = \left(1 - \frac{T_0}{T_L}\right)\dot{Q}_{Evap} + (\dot{E}_4 - \dot{E}_1) \tag{12}$$

$$\dot{E}_{D,Comp,CO_2} = (\dot{E}_1 - \dot{E}_2) + \dot{W}_{Comp,CO_2} \tag{13}$$

$$\dot{E}_{D,Val,CO_2} = (\dot{E}_3 - \dot{E}_4) \tag{14}$$

$$\dot{E}_{D,casc} = (\dot{E}_2 + \dot{E}_8) - (\dot{E}_3 + \dot{E}_5) \tag{15}$$

$$\dot{E}_{D,Comp,NH_3} = (\dot{E}_5 - \dot{E}_6) + \dot{W}_{Comp,NH_3} \tag{16}$$

$$\dot{E}_{D,Val,NH_3} = (\dot{E}_7 - \dot{E}_8) \tag{17}$$

$$\dot{Q}_{D,Cond} = \left(\frac{T_0}{T_0} - 1\right)\dot{Q}_{Cond} + (\dot{E}_6 - \dot{E}_7) \tag{18}$$

The exergy entering and leaving the system are respectively calculated by Eq. (19) and Eq. (20).

$$\dot{E}_{in} = \dot{W}_{Comp,NH_3} + \dot{W}_{Comp,CO_2} \tag{19}$$

$$\dot{E}_{out} = \dot{Q}_{Evap}\left(\frac{T_0}{T_L} - 1\right) \tag{20}$$

The total exergy destroyed in the system is calculated by Eq. (21)

$$\dot{E}_{D,total} = \dot{E}_{in} - \dot{E}_{out} = \sum \dot{E}_D \tag{21}$$

Finally, the exergétic efficiency of the system is based on the second law of thermodynamics and it is calculated by Eq. (22).

$$\eta_{II} = \left(\frac{\dot{E}_{out}}{\dot{E}_{in}}\right) = \left(1 - \frac{\dot{E}_{D,total}}{\dot{E}_{in}}\right) \tag{22}$$

3.3 Economic analysis

The economic analysis developed here is based only on operational cost (or annual cost), calculated by Eq. (23) and Eq. (24). The variable "H" representes how many hours during a year the system works.

$$\dot{C}_{op} = (\dot{W}_{Comp,NH_3} + \dot{W}_{Comp,CO_2})\frac{C_{elec}}{3600} \tag{23}$$

$$\dot{C}annual = (\dot{C}_{op})3600 * H \tag{24}$$

4. SIMULATIONS

At the present stage, it was used the ESS software (Engineering Equation Solver) to solve the equations used to model the proposed system. Two simulations were performed. The first one related to maximizing exergy efficiency and the second simulation related to minimizing operating cost. They were inserted into EES equations about energy, exergy and cost. Input source data necessary to solve the problem are presented in Tab. 1 that was used too by Aminyavari et al., 2014. These values were chosen as a reference to validate the proposed model. Additionally, to obtain the thermodynamic properties related to each thermodynamic state of the system was used a specific library contained in ESS. To perform the simulations was used an internal routine of EES based on numerical method called "Conjugate Directions Method" with an convergence error of 1.0 x 10^{-4} and a maximum of 300 iterations. The results of these simulations are shown in Tab. 2.

Table 1. Input variables

Inputs Variables	Values
CO_2 evaporation temperature ($T_{evap,CO2}$)	-56 ºC < $T_{evap,CO2}$ < -47 °C
CO_2 condensation temperature ($T_{cond,CO2}$)	-11 °C < $T_{cond,CO2}$ < 1 °C
Cascade temperature difference ($\Delta T_{cascade}$)	2 °C < $\Delta T_{cascade}$ < 10 °C
NH_3 condensation temperature ($T_{cond,NH3}$)	40°C < $T_{cond,NH3}$ < 65 °C
Cooling capacity (Q_{evap})	5 kW < Q_{evap} < 100 kW
Ambient temperature (T_0)	25 ºC
Ambient pressure (P_0)	1 atm
Refrigerated space temperature (T_L)	-45 ºC
Cost of electricity (C_{elec})	0.06 US$/(kWh)
Annual operational hours (H)	7000 h

In addition, it was studied the influence of some parameters such as $T_{evap,CO2}$, $T_{cond,CO2}$, $\Delta T_{cascade}$ and $T_{cond,NH3}$ in operational cost and in exergetic efficiency. They were fixed three of the four variables and the fourth one was varied. The three set values refer to the values found in the simulations results and the variable value refers to the temperature range shown in Tab. 1.

5. RESULTS AND DISCUSSION

The results obtained in the simulations are presented in Tab. 2. These results are compared with those obtained by Aminyavari et al., 2014. Both results are very close each other so the proposed model is valid.

Tabela 2. Results – output variables

Results - Outputs	Multi-objective optimization (Aminyavari et al. 2014)	Cost minimization and Efficiency maximization
CO_2 compressor isentropic efficiency	0.8400	0.8472
NH_3 compressor isentropic efficiency	0.7927	0.8002
CO_2 mass flow rate (kg/s)	0.1998	0.2012
NH_3 mass flow rate (kg/s)	0.0610	0.0604
CO_2 compressor work (kW)	14.7721	14.2
NH_3 compressor work (kW)	18.6755	17.86
Overall COP	1.4949	1.56
Total exergy efficiency (%)	45.89	47.89
Annual operational cost (US$/year)	14,048	13,464
$T_{evap,CO2}$ (°C)	-48.68	-47
$T_{cond,CO2}$ (°C)	-7.06	-6.301
$\Delta T_{cascade}$ (°C)	2.0	2.0
$T_{cond,NH3}$ (°C)	40.1	40
Q_{evap} (kW)	50	50

Still, it is noticeable that they always are obtained the same values for the output quantities, presented in Tab. 2, when it maximizes exergetic efficiency or it minimizes operational cost. As the operational cost is a function only of compressors power input, when it maximizes the efficiency exergetic or it minimizes the operation cost, the system should consume the smaller amount of electric power to perform the same amount of work. Thus, the maximum exergetic efficiency or the minimum operational cost occur simultaneously.

Confirming these results, they are presented in Fig. 2 to 5 for Qevap = 50kW the influence of evaporation and condensation temperatures of refrigerants in exergetic efficiency and in operational cost. As shown in these figures, the lower annual operating cost of the system always occurs parallel to its maximum exergetic efficiency.

In addition, the values obtained for $T_{evap,CO2}$, $T_{cond,CO2}$, $T_{cond,NH3}$ and $\Delta T_{cascade}$ always remain the same, as those found in Tab. 2, regardless of thermal load variation either to maximize the efficiency or minimize the operational cost.

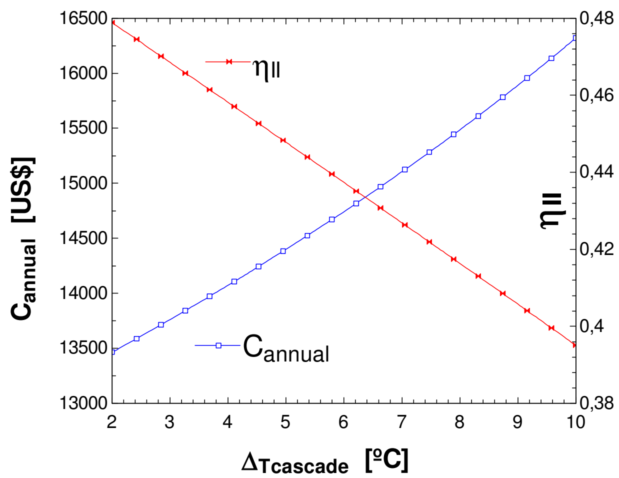

Figure 2. Diagram of C_{annual} and η_{II} versus $\Delta T_{cascade}$ ($T_{cond,CO2}$ = -6.301 °C , $T_{evap,CO2}$ = -47 °C, $T_{cond,NH3}$ =40 °C)

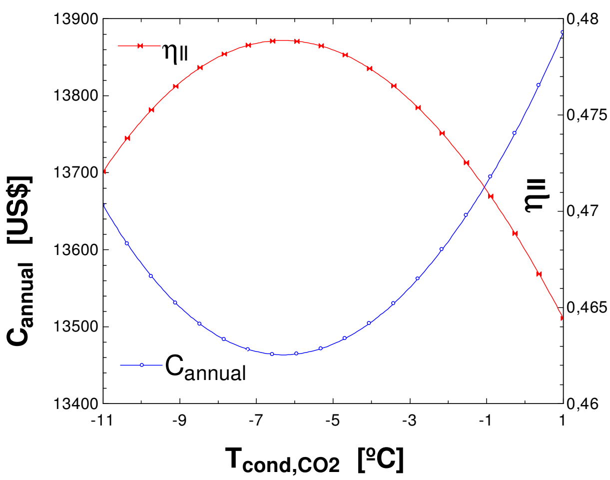

Figure 3. Diagram of C_{annual} and η_{II} versus $T_{cond,CO2}$ ($\Delta T_{cascade}$ = 2 °C , $T_{evap,CO2}$ = -47 °C, $T_{cond,NH3}$ =40 °C)

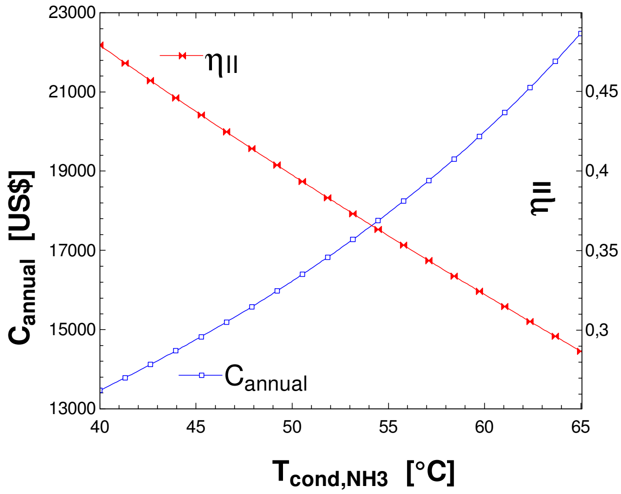

Figure 4. Diagram of C_{annual} and η_{II} versus $T_{cond,NH3}$ ($\Delta T_{cascade}$ = 2 °C, $T_{evap,CO2}$ = -47 °C, $T_{cond,CO2}$ = -6.301 °C)

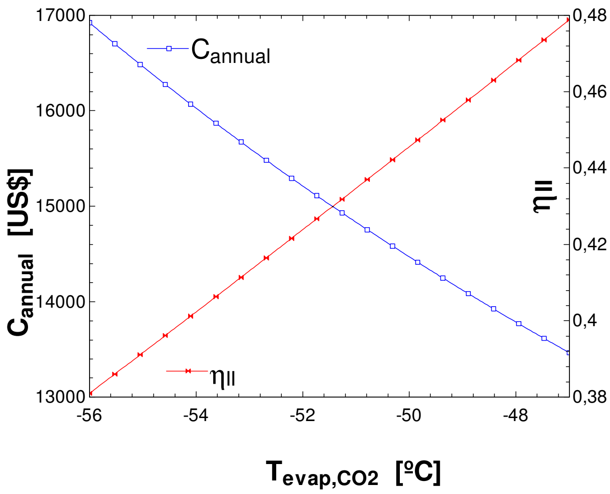

Figure 5. Diagram of C_{annual} and η_{II} versus $T_{evap,CO2}$ ($\Delta T_{cascade}$ = 2 °C, $T_{cond,NH3}$ =40 °C, $T_{cond,CO2}$ = -6.301 °C)

Regarding the behavior shown in Fig. 2, how higher the temperature difference in cascade heat exchanger how higher will be system irreversibilities. This situation, the increase in system irreversibilities, promotes reduction in system efficiency which results an increase in compressor power input to supply a constant thermal load and consequently ocurrs increasing in operational cost. Then, in this case, the increase in exergetic efficiency and the reduction in operational cost occur with the lowest temperature difference in the cascade heat exchanger. A similar behavior can be observed in Figure 4, when it consideres the influence of NH3 condensation temperature in system efficiency and cost. The reasons for this behavior are the same for those related to Fig. 2.

Analyzing Fig. 3 it can be seen that the increase in CO_2 condensation temperature from -11°C to -6.301°C promotes increase in system efficiency parallel to the reduction in operational costs. Above this temperature value occur simultaneously reduction in system efficiency and increase in operational cost. Above this temperature, -6.301°C, value also increases system irreversibilities which causes efficiency loss and consequently increase in operational cost.

Finally, referring to Fig. 5, the reduction in CO_2 evaporation temperature promotes reduction in system irreversibilities due to the reduction in the variation between CO_2 temperature evaporation and the temperature of the refrigerated space. This reduction in system irreversibilities promotes increase in system efficiency and consequently reducing operational cost, a situation like that occurs in Fig. 2 and 4.

6. CONCLUSION

After the development it perceives that for a cascade cooling systems operating with CO_2 and NH_3 as refrigerants, the maximum exergy efficiency occurs parallel to the lower operational cost. It happens due to the reduction in system irreversibilities. Besides that, the increase in system irreversibilities reduces its efficiency e consequently increases its operational cost.

Related to this fact, the increase in temperature difference in the cascade exchanger ($\Delta T_{cascade}$) and the increase in NH_3 condensation temperature($T_{cond,NH3}$) reduce system efficiency and consequently increase its operating costs due to the increase in system irreversibilities. In the other hand, the increase in CO_2 evaporation temperature ($T_{evap,CO2}$) promotes increase in system efficiency and reduces its operating cost due to the reduction in system irreversibilities. This behavior is also true for the CO_2 condensation temperature ($T_{cond,CO2}$).

So, it is recommended that these systems, whenever possible, must operate at its maximum exergy efficiency to promote reduction in the operating costs and thus reduced electricity consumption. As a suggestion for future work, it is recommended an analysis about the impact of exergetic efficiency system in relation to the cost of the heat exchangers and environmental one.

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8. RESPONSIBILITY NOTICE

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