MODELING, OPTIMIZATION, AND SIMULATION OF A COMMERCIAL AMMONIA-WATER ABSORPTION REFRIGERATION CYCLE FOR PRODUCTION OF CHILLED WATER

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Abstract. *This work presents a model for a 5 TR cooling capacity water-ammonia absorption refrigeration cycle directly powered by natural gas burning. Simulations were carried out by solving balances of mass and energy for each component of the cycle with the help of the software EES – Engineering Equation Solver. The introduction of the components in the modeling was gradually, starting from a simplified cycle to a more complete cycle with rectifier and heat exchangers. The study of the configuration of commercial cycles led to the model of an actual cycle. The results obtained in the simulations have shown to be consistent with those found in the literature. Also, it is presented an analysis showing the conditions were maximum COP is expected.*

Key words: *refrigeration by absorption, ammonia, computer simulation, chilled water.*

1. Introduction

An absorption refrigeration system uses a hot source for obtaining the refrigeration effect which ensures a lower consumption of electricity when compared to vapor compression systems and the possibility of using sources of heat, often rejected, such as it occurs in cogeneration systems.

This paper presents a study of an absorption refrigeration system by absorbing water-ammonia, in which ammonia is the coolant and water the absorbent. The study is based on a commercial cycle manufactured by ROBUR with a nominal capacity of 5 TR. Simulations were carried out with the software EES based on a steady state analysis of balances of mass and energy for each component of the cycle. Also, optimum operating conditions were found.

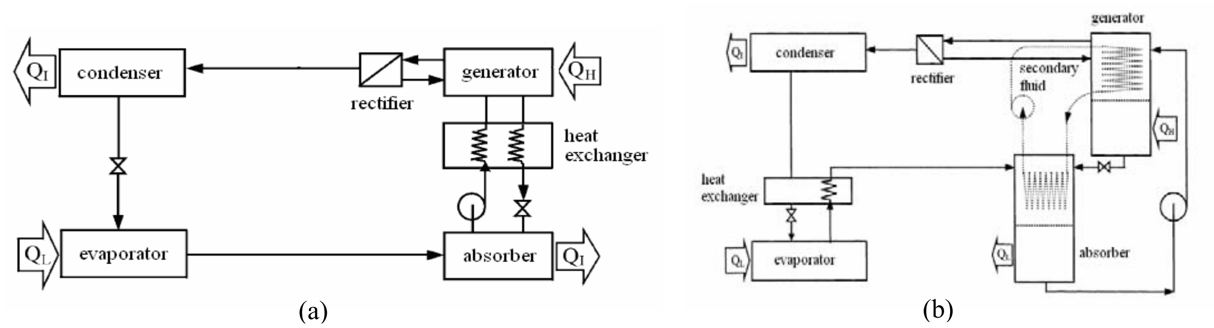
2. Cooling system by absorbing water-ammonia

The cycle can be divided into two parts: high and low pressure. The high pressure is achieved with the help of a small pump that works only with the water-ammonia solution (liquid state) at expenses of a little electrical consumption. The low pressure is obtained by throttling the high pressure liquid to a low pressure evaporator.

The low pressure stream from the evaporator is absorbed by a liquid solution (weak solution) in the absorber, which, from the equilibrium condition, requires that the solution must be cooled down by rejecting heat to the environment as the ammonia vapor is absorbed by the solution. Next, the solution has become rich in ammonia (strong solution) and it is pumped up to a high pressure level. The high pressure strong solution is next driven to a heat exchanger where it is pre-heated and then it reaches the generator, where the solution is heated by an external heating source, which causes the releasing the ammonia vapor. However, the ammonia vapor released from the generator still contains a good deal of water vapor. Thus, the vapor must pass through the rectifier to obtain an ammonia vapor of high purity (high concentration of ammonia). The ammonia vapor is driven to the condenser, and to the expansion valve, and finally to the evaporator to fulfill the thermodynamic cycle. The major components of a regular absorption refrigeration cycle can be seen in Fig 1a.

The raising of the performance of systems for cooling by absorption can be achieved through the use of cycles of *multi-effect*. The term *multi-effect* indicates that these cycles are configured so that the heat rejected in a stage of high temperature is used in a stage of low temperature, allowing the generation of an additional effect of cooling in the stage of low temperature. Meanwhile, the GAX cycles represent an elegant way in achieving high performance through settings that are essentially similar to the settings of simple-effect cycle. The term GAX refers to Generator / Absorber Heat Exchange and the system can be interpreted as two cycles of single-purpose working in parallel. The concept of GAX cycles is to simplify a system of two stages and double-effect, in order to get the same performance. The simplified configuration is shown schematically in Fig. 1b.

The pair absorber-generator can be considered a counter-current heat exchanger. In the absorber, the weak solution from the generator and the low temperature vapor of the evaporator enter in its upper section. The heat produced in the process of absorption should be rejected so that the vapor continues to be absorbed. In the upper section of the absorber, the heat is rejected at high temperature. In the section below, the solution takes up the ammonia vapor while is cooled by the rejection of heat to the environment. In the generator, the strong solution from the absorber enters into its upper section, where the coolant is separated from the solution due the warming caused by the rejection of heat of the absorber. In the lower section of the generator, the solution is separate from an external source of heat. Additionally, it is possible to use a secondary fluid for the transfer of heat between the absorber and generator. As a result, systems absorption of simple effect may have coefficients of performance as high as those that occur in two stages systems and double-effect through the use of GAX technology (Harold, 1996).



(a) (b)

Figure 1. a) *Simple ammonia-water absorption refrigeration system schematics*. (b) *GAX cycles concept.*

3. Study of a commercial cycle

The model originally developed was conceived to solve a simple cycle, consisting of an absorber, a solution pump, a generator, orifice valves, a condenser, and an evaporator. From this model a EES platform program was elaborated that allowed the simulation of the cycle based on a certain data set entry such as the cooling load and condensing and evaporating temperatures, and by setting up the state of the water-ammonia mixture at various parts of the cycle. Over the first simplified model it was gradually being introduced new components, i.e., a rectifier and, finally, a heat exchanger. The simulations confirmed the influence of those components, and their overall influence as well as the improvement of the coefficient of performance of the cycle.

Subsequently, the work was devoted to the study of the configuration of commercial cycles, aiming at to develop a model representative of a real cycle. For this, it was analyzed the absorption refrigeration equipment SERVEL ROBUR, ACF60-00 model, with capacity of 5TR, direct fired and powered by natural gas. Fig. 2 presents a schematics of the cycle of the equipment studied, indicating the path of the solution, coolant and the chilled water. It provides, therefore, valuable information regarding the provision of components and the operation of the cycle.

There are some features in the commercial cycle configuration, as the presence of a tube-in-tube heat exchanger and three expansion valves, resulting in three levels of pressure. In the heat exchanger, the liquid coolant in the outer tube transfers heat to the coolant vapor flowing in the internal tube. The strong solution is pumped to the rectifier before being driven to the pre-absorber, so that it is pre heated before reaching the generator. The weak solution from the section of higher temperature in the generator is sent to the pre-absorber, promoting the pre heating of the strong solution, which is driven to the section of the lower temperature of the generator, featuring the use of GAX technology.

4. Modeling

4.1. Model of the commercial cycle

The model of the commercial cycle was developed from the configuration shown in Fig. 2. The pre-absorber and absorber were represented by a single block.

4.2. Balance of mass and energy

For each component of the cycle were held balances of mass and energy. The balance of mass takes into account the mixture water-ammonia and also the mass of ammonia alone. Indexes from 1 to 15 represent important points of the cycle in Fig. 2 and over the balance equations.

Absorber

The pre-heating of the strong solution was considered in the solution. Balance of mass of the mixture in the absorber, where $\dot{m}$ is the mass flow of the mixture is given by:

$$\dot{m}_6 + \dot{m}_{15} = \dot{m}_1 \tag{1}$$

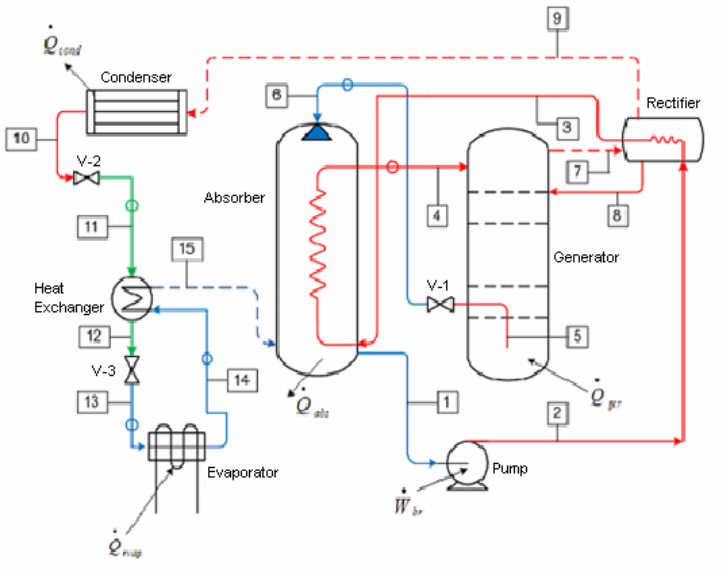


Figure 2. *Schematics of the commercial cycle.*

The mass fraction x is defined as the ratio of the mass of ammonia and the mass of the mixture. The balance of the mass of ammonia is shown in the following equation:

$$x_6 \dot{m}_6 + x_{15} \dot{m}_{15} = x_1 \dot{m}_1 \tag{2}$$

Balance of mass to the mixture in the pre-heated branch: $\dot{m}_3 = \dot{m}_4$ (3)

Balance of mass of ammonia in the pre-heated branch: $x_3 = x_4$ (4)

The energy balance allows the determination of the heat in the absorber, $\dot{Q}_{abs}$:

$$h_6 \dot{m}_6 + h_{15} \dot{m}_{15} + h_3 \dot{m}_3 = \dot{Q}_{abs} + h_1 \dot{m}_1 + h_4 \dot{m}_4 , \tag{5}$$

where, h is the specific enthalpy.

Pump

Balance of mass to the mixture: $\dot{m}_1 = \dot{m}_2$ (6)

Balance of mass of ammonia: $x_1 = x_2$ (7)

The enthalpy in section 2, assuming an isentropic pump (represented by the index s) is given by Eq. (8).

$$h_{2s} = h_1 + v_1 (P_2 - P_1) , \tag{8}$$

where, v is the specific volume and P is the pressure.

The pump power to the isentropic process, $\dot{W}_{ps}$, is calculated from Eq. (9).

$$\dot{W}_{ps} = \dot{m}_1 \cdot (h_{2s} - h_1) , \tag{9}$$

The actual power (represented by the index r) must consider the isentropic efficiency of the pump, η_P:

$$\dot{W}_{pa} = \frac{\dot{W}_{ps}}{\eta_P} , \tag{10}$$

Expansion valves (EV-1, EV-2, EV-3)

Balance of mass to the mixture:

$$\dot{m}_5 = \dot{m}_6 \tag{11}$$

$$\dot{m}_{10} = \dot{m}_{11} \tag{12}$$

$$\dot{m}_{12} = \dot{m}_{13} \tag{13}$$

Balance of mass to ammonia:

$$x_5 = x_6 \tag{14}$$

$$x_{10} = x_{11} \tag{15}$$

$$x_{12} = x_{13} \tag{16}$$

Balance of energy:

$$h_5 = h_6 \tag{17}$$

$$h_{10} = h_{11} \tag{18}$$

$$h_{12} = h_{13} \tag{19}$$

Generator

Balance of mass to the mixture:

$$\dot{m}_4 + \dot{m}_8 = \dot{m}_5 + \dot{m}_7 \tag{20}$$

Balance of mass to ammonia:

$$x_4 \dot{m}_4 + x_8 \dot{m}_8 = x_5 \dot{m}_5 + x_7 \dot{m}_7 \tag{21}$$

The balance of energy allows one to determine the rate of addition of heat to the generator, $\dot{Q}_{gen}$. It is also considered the efficiency of the generator, ε_{ger}, so that the rate of actual heat provided to solution, $\dot{Q}_{gen,actual}$, is calculated by Eq. (16).

$$\dot{Q}_{gen,actual} = \varepsilon_{gen} \cdot \dot{Q}_{gen}, \tag{22}$$

$$h_4 \dot{m}_4 + h_8 \dot{m}_8 + \dot{Q}_{gen,actual} = h_7 \dot{m}_7 + h_5 \dot{m}_5, \tag{23}$$

Rectifier

Balance of mass to the mixture in the rectifier:

$$\dot{m}_7 = \dot{m}_9 + \dot{m}_8 \tag{24}$$

Balance of mass of ammonia in the rectifier:

$$x_7 \dot{m}_7 = x_9 \dot{m}_9 + x_8 \dot{m}_8 \tag{25}$$

Balance of mass to the mixture in the pre-heated branch:

$$\dot{m}_2 = \dot{m}_3 \tag{26}$$

Balance of mass of ammonia in the pre-heated branch:

$$x_2 = x_3 \tag{27}$$

The energy balance must also consider the rate of heat in the rectifier, $\dot{Q}_{rect}$:

$$h_7 \dot{m}_7 = \dot{Q}_{rect} + h_9 \dot{m}_9 + h_8 \dot{m}_8, \tag{28}$$

The rate of heat transfer in the pre-heated branch in the rectifier, $\dot{Q}_{rect,actual}$, must consider the efficiency of the rectifier, ε_{rect}:

$$\varepsilon_{rect} = \frac{\dot{Q}_{rect,actual}}{\dot{Q}_{rect}}, \tag{29}$$

$$\dot{Q}_{rect,actual} = \varepsilon_{rect} \cdot \dot{Q}_{rect} = \dot{m}_3 \cdot \left(h_3 - h_2\right), \tag{30}$$

Condenser

Balance of mass for the mixture: $$\dot{m}_9 = \dot{m}_{10} \tag{31}$$

Balance of mass for ammonia: $$x_9 = x_{10} \tag{32}$$

Balance of energy, which $\dot{Q}_{cond}$ is the rate of heat exchange in the condenser:

$$h_9 \dot{m}_9 = \dot{Q}_{cond} + h_{10} \dot{m}_{10}, \tag{33}$$

Evaporator

Balance of mass for the mixture: $$\dot{m}_{11} = \dot{m}_{12} \tag{34}$$

Balance of mass for ammonia: $$x_{11} = x_{12} \tag{35}$$

The balance of energy considers the rate of heat exchange in the evaporator, $\dot{Q}_{evap}$:

$$h_{11} \dot{m}_{11} + \dot{Q}_{evap} = h_{12} \dot{m}_{12}, \tag{36}$$

Heat exchanger

The enthalpies of sections 12 and 15 are determined from the rate of heat transfer, $\dot{q}_{actual,HE}$, considering the efficiency ε_{HE} of the exchanger and the minimum rate of heat transfer, $\dot{q}_{\min,HE}$.

$$\dot{q}_{actual,HE} = \varepsilon_{HE} \cdot \dot{q}_{\min,HE}, \tag{37}$$

$$h_{12} = h_{11} - \frac{\dot{q}_{actual}}{\dot{m}_{11}}, \tag{38}$$

$$h_{15} = h_{14} + \frac{\dot{q}_{actual}}{\dot{m}_{14}}, \tag{39}$$

Balances of mass for the mixtures in exchanger branches:

$$\dot{m}_{11} = \dot{m}_{12}, \tag{40}$$

$$\dot{m}_{14} = \dot{m}_{15}, \tag{41}$$

Balances of mass for ammonia:

$$x_{11} = x_{12} \tag{42}$$

$$x_{14} = x_{15} \tag{43}$$

Balance of energy:

$$h_{11} \dot{m}_{11} + h_{14} \dot{m}_{14} = h_{12} \dot{m}_{12} + h_{15} \dot{m}_{15} \tag{44}$$

The coefficient of performance (*COP*) of the refrigeration cycle by absorption is determined by Eq. (39).

$$COP = \frac{refrigeration\ capacity}{rate\ of\ \ heat\ addition\ in\ the\ generator} = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \tag{45}$$

It is important to note that the power supplied to the pump was not considered when calculating the coefficient of performance, since it is considerably less than the rate of addition of heat to the generator.

5. Simulations

5.1. Data set entry and operational conditions

From the resolution presented, a program was implemented in EES for determining the states of the points of the thermodynamic cycle shown in Fig. 2, and the mass flow rates, the heat exchange in the various components, the *COP*, among other parameters. Based on a literature review, the following entry data and operational conditions were considered:

Entry data set:

Absorber output temperature of absorber: $T_1 = 40^oC$
Condensing temperature: $T_{10} = 40^oC$
Evaporating temperature: $T_{14} = -10^oC$
Generator output temperature: $T_7 = 87^oC$
Refrigeration load: Qevap = 5TR
Efficiency of the pump: $\eta_P = 0,85$
Other Efficiencies: $\varepsilon_{HE} = \varepsilon_{rect} = \varepsilon_{abs} = 0,95$; $\varepsilon_{gen} = 0,98$

Operational conditions:

Mass vapor qualities: $q_1 = q_5 = q_8 = q_{10} = 0$ (liquid saturated); $q_7 = q_9 = 1$ (saturated vapor)
Vapor quality in the output of evaporator: $q_{14} = 0,915$
Mass fraction at the rectifier output: $x_7 = 0,999634$
Difference in the composition of recirculation of absorber: $x_1 - x_5 = 0,3$

Pressure equatization:

$P_1 = P_6 = P_{13} = P_{14} = P_{15} = P_b$ (low pressure)
$P_2 = P_3 = P_4 = P_5 = P_7 = P_8 = P_9 = P_{10} = P_a$ (high pressure)
$P_{11} = P_{12} = P_m$ (intermediate pressure)

The temperature at the point 7 was defined based on the study of literature search and the temperature in the point 8 was written as a linear combination of temperature in section 7 (generator output) and the temperature in section 9 (rectifier output), as Eq. (38), where α is the weighting factor.

$$T_8 = \alpha \cdot T_7 + (1-\alpha) \cdot T_9 \tag{46}$$

Through the analysis of the examples presented in the literature, the implemented program used $\alpha = 0,5$.

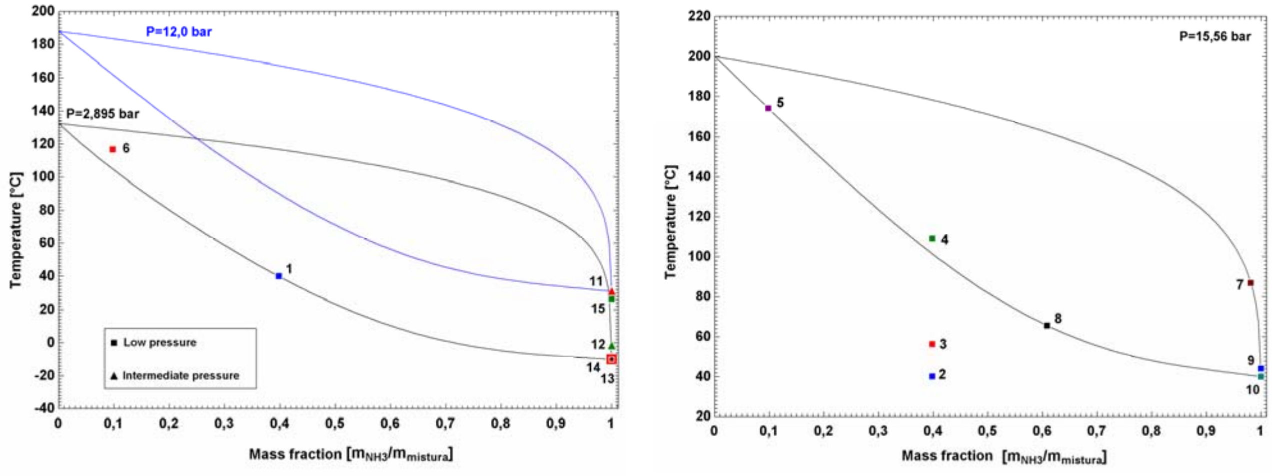
5.2. Results

Table 1 presents the *COP* of the refrigeration system and the rates of heat exchange obtained in the simulation for the components.

Table 1. *Coefficient of performance and rates of heat exchange in the components.*

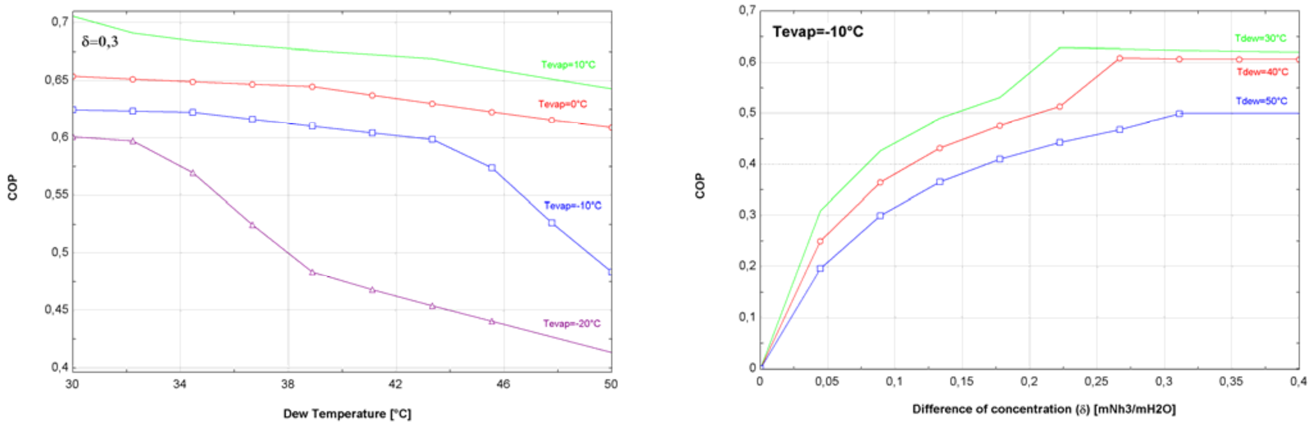
COP	$\dot{Q}_{abs}$ $[kW]$	$\dot{Q}_{gen}$ $[kW]$	$\dot{Q}_{gen,actual}$ $[kW]$	$\dot{Q}_{cond}$ $[kW]$	$\dot{Q}_{rect}$ $[kW]$	$\dot{W}_{pa}$ $[kW]$
0,612	28,76	28,73	28,15	16,88	3,41	0,08

The thermodynamic states can be viewed in diagrams temperature-mass fraction in Figs. 3a and 3b. Fig. 3a presents the diagram for the intermediate and low pressures, while Fig. 3b gives the diagram for the high pressure of the cycle. The indicated points are the ones of the cycle in Fig. 2.



(a) (b)

Figure 3. (a)*Diagram temperature-mass fraction for low and intermediate pressures*. (b) *Diagram for the high pressure*



(a) (b)

Figure 4. (a) *Effect of changes in the evaporation and condensation temperatures*. (b) *Effect of variation of the difference in concentration at the COP*.

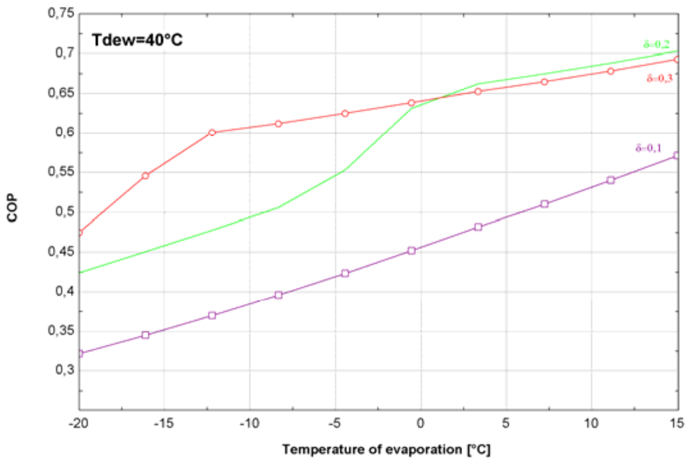


Figure 5. *Effect of the variation of evaporation temperature over the COP*.

The study of absorption refrigeration cycles leads to the discussion of the possibilities for optimization. The improvement of the coefficient of performance of cycles can be achieved through the use of cycles of multiple effects. The use of GAX technology is an interesting alternative, simplifying the systems of two stages and double effect to obtain performance considerably high. Maximizing the *COP* is a goal of any refrigeration engineer, since it really result in a better way of energy usage. However, it is also necessary to consider the costs associated with the construction and operation of the equipment. Moreover, since the equipment available are relatively bulky, the project should consider the development of more compact equipment ones.

Recent works (Ortigosa, 2007) show that when the model of the commercial cycle is established, it is possible to study the effect of changes in certain parameters in the coefficient of performance of the cycle. In particular, it is interesting to examine the influence of the variation in temperatures of evaporation and condensation in high and low pressure so that the state of ammonia in the output of condenser and the evaporator is changed and therefore the input and output heat of the cycle are different, changing thus the value of the *COP*, such changes can be seen in the graphic shown in Fig. 4a. Fig. 4b present as the *COP* varies depending on the difference of concentration (δ) in the absorber recirculation (between 1 and 5 points in Fig. 2) showing that with increasing δ there is an increase in the efficiency of absorption of ammonia in the absorber and the *COP* tends to increase. Finally, Fig. 5 shows that there is a continuous increase of the *COP* as the evaporation temperature approaches the condensation temperature, regardless of the solution concentration, which was an expected result.

6. Conclusions

The methodology for the development of a more complex model by a gradual introduction of new components to the cycle led to the observation of the influence of the rectifier and heat exchangers over refrigeration cycles by absorption. The results confirm that the vapor that leaves the generator have certain quantity of water and that after the passage in the rectifier, the vapor found is practically composed of pure ammonia. The study also highlighted the importance of pre-heating the strong solution before being driven to the generator, as it involves a minor addition of heat in the generator, leading to a rising of the *COP* of the refrigeration system by absorption.

The study of the configuration of the commercial cycle led the work to draw up a more detailed model. It was found that the commercial cycle uses the GAX technology, which is promoted through an exchange of heat between the generator and absorber, which outcomes into a higher coefficient of performance.

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