# EXERGOECONOMIC ANALYSIS OF A SINGLE PRESSURE ABSORPTION REFRIGERATION CYCLE

**Wilson Luciano de Souza, wilson@ufs.br**
Universidade Federal de Sergipe (UFS), Av. Marechal Rondon, s/n, Jardim Rosa Elze, 49100-000, São Cristovão-SE, Brasil.

**Carlos Antonio Cabral dos Santos, cabral@les.ufpb.br**
Universidade Federal da Paraíba (UFPB)/Laboratório de Energia Solar (LES), 58051-970, João Pessoa-PB, Brasil.

**Ednildo Andrade Torres, ednildo@ufba.br**
Universidade Federal da Bahia (UFBA)/Escola Politécnica/DEQ/LEN, 40210-630, Salvador-BA, Brasil.

**Jacques Cesar dos Santos, jcquesles@yahoo.com.br**
Universidade Federal da Paraíba (UFPB)/Laboratório de Energia Solar (LES), 58051-970, João Pessoa-PB, Brasil.

**Evânia Celeste da Costa Moreira, evaniaceleste@ig.com.br**
Universidade Federal da Paraíba (UFPB)/Laboratório de Energia Solar (LES), 58051-970, João Pessoa-PB, Brasil.

***Abstract.*** *This article presents a thermodynamic modelling of a single pressure absorption refrigeration system, starting from the concepts of the 1st and 2nd Law of the Thermodynamics and from the cost exergetic and monetary theory, that it allowed to evaluate the energetic, exergetic and exergoeconomic performance of this system through parametric simulation. The cycle in study is the diffusion absorption one, operating with three working fluids: ammonia as refrigerant, water as absorbent, and helium as auxiliary gas. The circulation of the working fluids is driven by a bubble-pump and the pressure equalization throughout the cycle is provided by the auxiliary gas. The computational code was developed at the EES (Engineering Equation Solver) Software for the energetic and exergoeconomic analysis having been supplied as main entrance parameters the concentrations of the solution ammonia-water in the input to the generator and in the input to the absorber of the system, condensation temperature and heat supplied to the steam generator. Through the code, were evaluate COP, the irreversibilities and the degree of perfection of each component.*

***Keywords****: Absorption-diffusion refrigeration ammonia-water-helium, thermodynamic model and performance.*

## 1. INTRODUCTION

The diffusion absorption refrigeration cycle, subject of this study, was invented by Platen and Munters and patented in 1928 (*apud* Srikhirin and Aphornratana, 2002). It is operated by thermal energy, that can be powered by kerosene or liquid petroleum gas and do not require electrical and mechanical energy. As it consists of no moving parts and not presents noise and vibration during the operation, it is recommended to application in hotel rooms, offices, camping and as recreative vehicle refrigerator. The working fluids are ammonia-water: ammonia as a refrigerant, water as an absorbent. It uses hydrogen or helium as an auxiliary gas. The system is of single pressure and the circulation of the working fluids is driven by a bubble-pump and the pressure equalization throughout the cycle is provided by the auxiliary gas.

Beyond these advantages, the system exhibits good reliability, durability and minimum maintenance costs. It operates without chlorofluorcarbons.

Recently, the system has been studied by every researchers to improve the current cycle performance. Chen *et al*. (1996) modified the original system whit the inclusion of an heat exchanger at the generator. The working fluids are ammonia-water-hydrogen. The system with the new generator demonstrated a significant improvement in the cooling COP of as much as 50% compared to the original system.

Srikhirin and Aphornratana (2002) studied the cycle with the working fluids ammonia-water-helium. They fabricated an experimental unity based on the Platen and Munters cycle. The results of their developed mathematical model was compared with the experimental results, and showed that the system performance is strongly dependent upon the bubble-pump characteristics and the absorber and evaporator mass transfer performance.

Zohar *et al*.(2005) developed a thermodynamic model for an ammonia-water diffusion absorption refrigerator, manufactures by Eletrolux Sweden, whose inert gas was hydrogen or helium. The performance of the cycle was investigated by computer simulation and their results showed that the best performance was obtained for a concentration range of the rich solution of 0.2-0.3 ammonia mass fraction and the recommended concentration of weak solution was 0.1. The results showed that the system operating with helium as auxiliary gas presented the coefficient of performance up to 40% higher than a system working whit hydrogen.

Souza (2007) did an extensive review of technologies for cooling by absorption and thermodynamic models developed for analyzing performance, optimization and provide identification of components that generate higher costs and irreversibility in cycles of absorption of single pressure, represented by cycles of diffusion - Einstein and the absorption cycle.

Because of low efficiency and high temperature limit of the cooling cycles of the pressure has only limited applications where features such as mobility, simplicity, portability, robustness, quiet operation and low cost are important. Improved efficiency in open other potential commercial applications.

## 2. CYCLE DESCRIPTION

The description of the operation of the refrigeration cycle for absorption-diffusion can be found in the literature (Chen *et al.*, 1996, Srikhirin *et al.*, 2002, Zohar *et al.*, 2005a, Souza, 2007). The diffusion absorption refrigeration cycle studied is showed in the Figure 1.

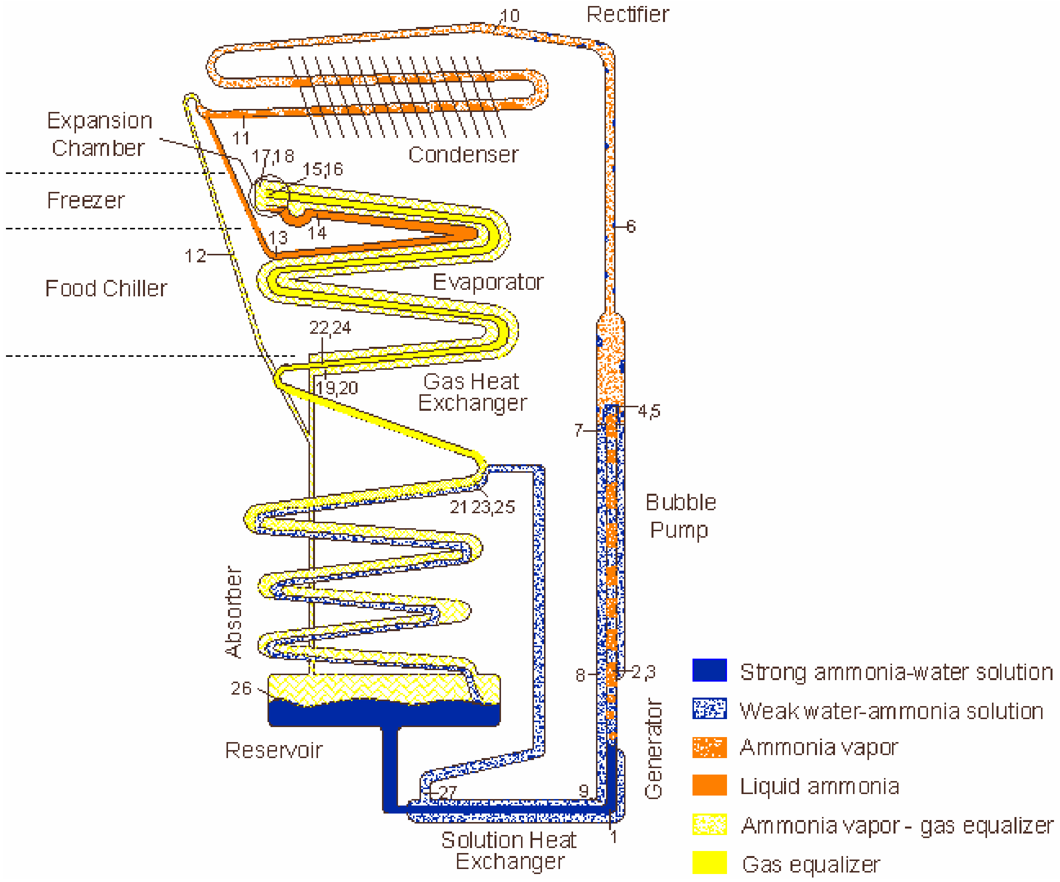


Figure 1. Schematic diagram of the diffusion absorption refrigerator cycle.

In the generator of this system, vapor of ammonia is separated from the rich solution and the vapor bubbles then rises inside the bubble pump. Weak solution and the ammonia-water vapor exit the bubble pump. In the rectifier, the ammonia vapor is purified and flows to the condenser, and the water vapor condenses to joint the weak solution, that flows to the absorver through solution heat exchanger. The condenser is cooled by air and the ammonia liquid flows to the evaporator. Uncondensed ammonia flows to the reservoir through the gas bypass. At the evaporator entrance the sub-cooled liquid ammonia meets helium arriving from the absorber after having passed through the gas heat exchanger and its partial pressure drops. In the absorber, ammonia absorption takes place and turning into rich solution, which flows into the reservoir and continues to the generator. The auxiliary gas is not absorbed and returns to the evaporator.

## 3. EXERGOECOMONIC MODEL

To developed the exergoeconomic model, the mass, concentration and energy conservation laws (First and Second Laws of the Thermodynamics) (Bejan *et al.*,1996), and the theory of exergetic and monetary cost (Valero *et al.*, 1986 and LOZANO *et al.*, 1989), was applied in each component of the cycle, where the established control volume for each one of them includes the external reservoir. Every element of the system was analyzed separately. The subscripts of the various properties are relations to the locations indicated in Figure 1 and Figure 2.

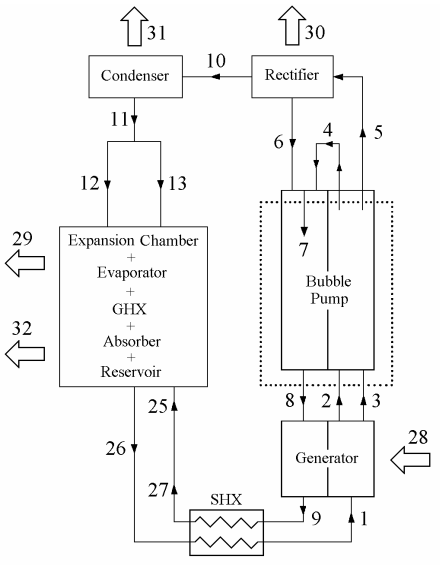


Figure 2. Flowchart of diffusion-absorption cycle for termoeconomic analysis.

Table 1 shows the input variables for computer simulation of the proposed model, which served as the basis of input data in the parametric study platform developed in EES (Engineering Equation Solver).

Table 1. Input parameters for computer simulation of the refrigeration system through absorption-diffusion.

| **Parameters** / **Papers** | **Chen *et al.* (1996)** | **Srikhirin *et al.* (2002)** | **Zohar *et al.* (2005a e 2005b)** | **Adopted in this study** |
|---|---|---|---|---|
| Rich concentration of the solution | 0.29 | 0.35 | 0.30 | 0.30 |
| Poor concentration of the solution | 0.15 | 0.10 | 0.10 | 0.10 |
| Generator Temperature (ºC) | 180 | 180 | 200 | 195 |
| Bubble pump temperature (ºC) | – | – | 195 | 190 |
| Temperature of ammonia in the rectifier (ºC) | – | 75 a 80 | ≈ 70 | 78 |
| Absorber temperature (ºC) | – | – | 58 | 58 |
| Temperature gas equalizer with residual ammonia SHX entering (ºC) | – | – | 40 | 40 |
| Temperature, evaporation of ammonia in the freezer (ºC) | – | –30 a –18 | –35 | –34 |
| Temperature, evaporation of ammonia in the food chiller (ºC) | – | –5 | –5 | –5 |
| Total system pressure (bar)[1] | 25.5 | ≈ 25 | 25 | 25 |
| Partial pressure of ammonia in the freezer (bar) | 1 | 1 | 1 | 1 |
| Partial pressure of auxiliary gas in the freezer (bar) | – | 24 | – | 24 |
| Partial pressure of ammonia in the food chiller (bar) | 3 | 3 | 3 | 3 |
| Power supplied to the generator (W) | 175 a 370 | 1000 a 2500 | – | 3000 |

This model was simulated in platform ESS, which allowed the use of termophysical properties intrinsic to the software for both the ammonia-water pair as for the helium / hydrogen. In thermodynamic modeling, the following assumptions were made:

- All components operate on a steady state;
- The control volume each component involves only the working fluid entry and exit;

---

[1] The total pressure of the system governing the temperature at which the capacitor operates. For pure ammonia, the temperature of the first drop of condensate is determined by the load of auxiliary gas. If the system is filled to 25 bar, and the assumption is made that the rectifier all the water was removed, then the temperature of the first drop of condensate correspond to the saturation temperature of pure ammonia at 25 bar (58 ° C).

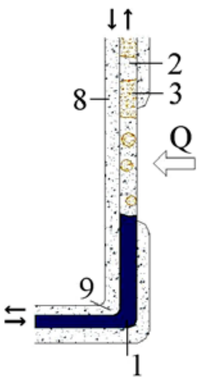
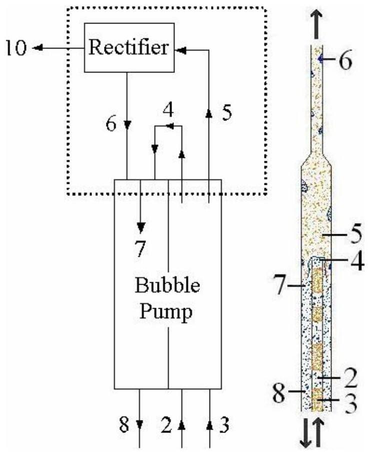
- Changes in kinetic and potential energy are considered negligible;
- disregards the load losses in pipes;
- There is no completion of work;
- Since the air is cooled, the temperature of the condenser is equal to the heat exchanger solution;
- In the output of the generator, solution and vapor bubbles of ammonia-water temperatures are equal;
- All processes are adiabatic, except those involving the generator, rectifier, condenser, evaporator and absorber;
- The properties of the gas equalizer were determined considering it to behave as ideal gas.

### 3.1. Control Volumes of the Cycle Absorption-Diffusion

The definition of controls volumes for the cycles of absorption-diffusion has been outlined so that all components were involved, allowing circumvent the problems by applying the balance of mass, species and energy, lack of data and measurements in the literature for some points of the cycles.

The control volumes were chosen: a steam generator, pump bubble, rectifier, condenser, solution heat exchanger, evaporator, expansion chamber, absorber and heat exchanger for gas, Tab (2).

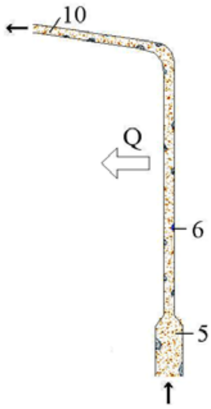
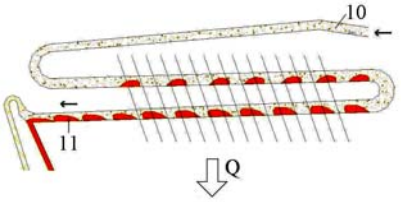
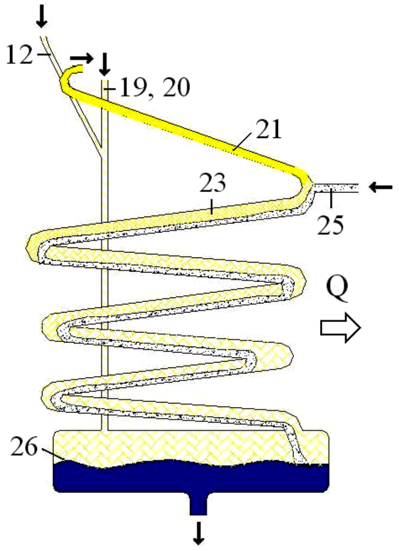
Table 2. Control Volumes and equating proposed for the cooling system of absorption-diffusion.

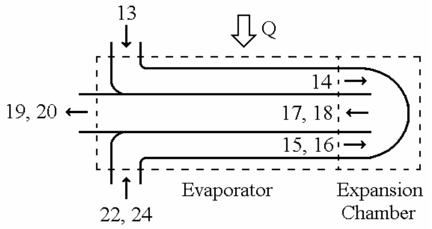
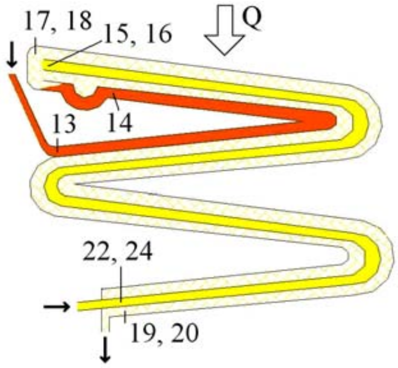
| | **Equations**[2] | **Control Volumes** |
|---|---|---|
| Generator | 1. $\dot{m}_2 + \dot{m}_3 + \dot{m}_9 = \dot{m}_1 + \dot{m}_8$<br>2. $\dot{m}_2 x_2 + \dot{m}_3 y_3 + \dot{m}_9 x_9 = \dot{m}_1 x_1 + \dot{m}_8 x_8$<br>3. $\dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{m}_9 h_9 = \dot{m}_1 h_1 + \dot{m}_8 h_8 + \dot{Q}_{ger}$<br>4. $\dot{m}_2 s_2 + \dot{m}_3 s_3 + \dot{m}_9 s_9 = \dot{m}_1 s_1 + \dot{m}_8 s_8 + \dfrac{\dot{Q}_{ger}}{T_{2,3}} + \dot{S}_{ger,ger}$<br>5. $\eta_{ger} = \left(\dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{m}_9 h_9 - (\dot{m}_1 h_1 + \dot{m}_8 h_8)\right) \big/ \dot{Q}_{ger}$<br>6. $\varepsilon_{ger} = \left(\dot{m}_2 ex_2 + \dot{m}_3 ex_3 + \dot{m}_9 ex_9 - (\dot{m}_1 ex_1 + \dot{m}_8 ex_8)\right) \Big/ \left(1 - \dfrac{T_0}{T_2}\right)\dot{Q}_{ger}$<br>7. $\xi_{ger} = \left(\dot{m}_2 ex_2 + \dot{m}_3 ex_3 + \dot{m}_9 ex_9\right) \Big/ \left(\dot{m}_1 ex_1 + \dot{m}_8 ex_8 + \left(1 - \dfrac{T_0}{T_2}\right)\dot{Q}_{ger}\right)$<br>8. $Ex_2^* + Ex_3^* + Ex_9^* = Ex_1^* + Ex_8^* + Ex_{28}^*$<br>9. $C_2^* + C_3^* + C_9^* = C_1^* + C_8^* + C_{28}^* + Z_{gerador}$[3] |  |
| Bubble Pump | 1. $\dot{m}_4 + \dot{m}_5 + \dot{m}_8 = \dot{m}_2 + \dot{m}_3 + \dot{m}_7$ e $\dot{m}_7 = \dot{m}_4 + \dot{m}_6$<br>2. $\dot{m}_4 x_4 + \dot{m}_5 y_5 + \dot{m}_8 x_8 = \dot{m}_2 x_2 + \dot{m}_3 y_3 + \dot{m}_7 x_7$<br>3. $\dot{m}_4 h_4 + \dot{m}_5 h_5 + \dot{m}_8 h_8 = \dot{m}_2 h_2 + \dot{m}_3 h_3 + \dot{m}_7 h_7$<br>4. $\dot{m}_4 s_4 + \dot{m}_5 s_5 + \dot{m}_8 s_8 = \dot{m}_2 s_2 + \dot{m}_3 s_3 + \dot{m}_7 s_7 + \dot{S}_{ger,bomba}$<br>5. $\eta_{bomba} = \dfrac{\dot{m}_4 h_4 + \dot{m}_5 h_5 - \dot{m}_7 h_7}{\dot{m}_2 h_2 + \dot{m}_3 h_3 - \dot{m}_8 h_8}$<br>6. $\varepsilon_{bomba} = \dfrac{\dot{m}_4 ex_4 + \dot{m}_5 ex_5 - \dot{m}_7 ex_7}{\dot{m}_2 ex_2 + \dot{m}_3 ex_3 - \dot{m}_8 ex_8}$<br>7. $\xi_{bomba} = \dfrac{\dot{m}_4 ex_4 + \dot{m}_5 ex_5 + \dot{m}_8 ex_8}{\dot{m}_2 ex_2 + \dot{m}_3 ex_3 + \dot{m}_7 ex_7}$<br>8. $Ex_4^* + Ex_5^* + Ex_8^* = Ex_2^* + Ex_3^* + Ex_7^*$<br>9. $C_4^* + C_5^* + C_8^* = C_2^* + C_3^* + C_7^* + Z_{bomba}$ |  |

---

[2] Where the numbers 1, 2, 3, 4, 5, 6, 7, 8 and 9 respectively mean: balance of mass, species, energy, entropy generation, efficiency of 1st and 2nd Law Thermodynamics, degree of perfection, exergetic and monetary cost.

[3] The cost of money off subsystems i, $Z_i$, of the cooling unit (Valero *et al*., 1989): $Z_i = \dfrac{(A/P)}{t_{op}} \cdot \dot{r}_i \cdot F$

| | **Equations** | **Control Volumes** |
|---|---|---|
| Rectifier | 1. $\dot{m}_6 + \dot{m}_{10} = \dot{m}_5$<br>2. $\dot{m}_6 x_6 + \dot{m}_{10} y_{10} = \dot{m}_5 y_5$<br>3. $\dot{m}_6 h_6 + \dot{m}_{10} h_{10} = \dot{m}_5 h_5 + \dot{Q}_{ret}$<br>4. $\dot{m}_6 s_6 + \dot{m}_{10} s_{10} = \dot{m}_5 s_5 + \frac{\dot{Q}_{ret}}{T_{10}} + \dot{S}_{ger,ret}$<br>5. $\eta_{ret} = \frac{\dot{Q}_{ret}}{\dot{m}_6 h_6 + \dot{m}_{10} h_{10} - \dot{m}_5 h_5}$<br>6. $\varepsilon_{ret} = \left(1 - \frac{T_0}{T_{10}}\right)\dot{Q}_{ret} \Big/ \left(\dot{m}_6 ex_6 + \dot{m}_{10} ex_{10} - \dot{m}_5 ex_5\right)$<br>7. $\xi_{ret} = \left(\dot{m}_6 ex_6 + \dot{m}_{10} ex_{10} + \left(1 - \frac{T_0}{T_{10}}\right)\dot{Q}_{ret}\right) \Big/ \dot{m}_5 ex_5$<br>8. $Ex_6^* + Ex_{10}^* + Ex_{30}^* = Ex_5^*$<br>9. $C_6^* + C_{10}^* + C_{30}^* = C_5^* + Z_{retificador}$ |  |
| Condenser | 1. $\dot{m}_{11} = \dot{m}_{10}$<br>2. $\dot{m}_{11} x_{11} = \dot{m}_{10} y_{10}$<br>3. $\dot{m}_{11} h_{11} = \dot{m}_{10} h_{10} + \dot{Q}_{cond}$<br>4. $\dot{m}_{11} s_{11} = \dot{m}_{10} s_{10} + \frac{\dot{Q}_{cond}}{T_{11}} + \dot{S}_{ger,cond}$<br>5. $\eta_{cond} = \dot{Q}_{cond} / \left(\dot{m}_{11} h_{11} - \dot{m}_{10} h_{10}\right)$<br>6. $\varepsilon_{cond} = \left(1 - \frac{T_0}{T_{11}}\right)\dot{Q}_{cond} \Big/ \left(\dot{m}_{11} ex_{11} - \dot{m}_{10} ex_{10}\right)$<br>7. $\xi_{cond} = \left(\dot{m}_{11} ex_{11} + \left(1 - \frac{T_0}{T_{11}}\right)\dot{Q}_{cond}\right) \Big/ \dot{m}_{10} ex_{10}$<br>8. $Ex_{11}^* + Ex_{31}^* = Ex_{10}^*$<br>9. $C_{11}^* + C_{31}^* = C_{10}^* + Z_{condensador}$ |  |
| Absorber + Reservoir | 1. $\dot{m}_{H,21} + \dot{m}_{23} + \dot{m}_{26} = \dot{m}_{12} + \dot{m}_{19} + \dot{m}_{H,20} + \dot{m}_{25}$<br>$\dot{m}_{19} = \dot{m}_{13} + \dot{m}_{23}$ e $\dot{m}_{12} = \dot{m}_{10} \cdot x_{q,11}$<br>2. $\dot{m}_{H,21} + \dot{m}_{23} y_{10} + \dot{m}_{26} x_{26} = \dot{m}_{12} y_{10} + \dot{m}_{19} y_{10} + \dot{m}_{H,20} + \dot{m}_{25} x_{25}$<br>3. $\dot{m}_{H,21} h_{H,21} + \dot{m}_{23} h_{23} + \dot{m}_{26} h_{26} = \dot{m}_{12} h_{12} + \dot{m}_{19} h_{19} + \dot{m}_{H,20} h_{H,20} +$ $+ \dot{m}_{25} h_{25} + \dot{Q}_{abs}$<br>4. $\dot{m}_{H,21} s_{H,21} + \dot{m}_{23} s_{23} + \dot{m}_{26} s_{26} = \dot{m}_{12} s_{12} + \dot{m}_{19} s_{19} + \dot{m}_{H,20} s_{H,20} +$ $+ \dot{m}_{25} s_{25} + \frac{\dot{Q}_{abs}}{T_{26}} + \dot{S}_{ger,abs}$<br>5. $\eta_{abs} = \dot{Q}_{abs} \Big/ \left(\begin{array}{l} \dot{m}_{H,21} h_{H,21} + \dot{m}_{23} h_{23} + \dot{m}_{26} h_{26} - \\ - (\dot{m}_{12} h_{12} + \dot{m}_{19} h_{19} + \dot{m}_{H,20} h_{H,20} + \dot{m}_{25} h_{25}) \end{array}\right)$<br>6. $\varepsilon_{abs} = \frac{\left(1 - \frac{T_0}{T_{26}}\right)\dot{Q}_{abs}}{\left(\begin{array}{l} \dot{m}_{H,21} ex_{H,21} + \dot{m}_{23} ex_{23} + \dot{m}_{26} ex_{26} - \\ - (\dot{m}_{12} ex_{12} + \dot{m}_{19} ex_{19} + \dot{m}_{H,20} ex_{H,20} + \dot{m}_{25} ex_{25}) \end{array}\right)}$<br>7. $\xi_{abs} = \frac{\dot{m}_{H,21} ex_{H,21} + \dot{m}_{23} ex_{23} + \dot{m}_{26} ex_{26} + \left(1 - \frac{T_0}{T_{26}}\right)\dot{Q}_{abs}}{\dot{m}_{12} ex_{12} + \dot{m}_{19} ex_{19} + \dot{m}_{H,20} ex_{H,20} + \dot{m}_{25} ex_{25}}$ |  |

| | **Equations** | **Control Volumes** |
|---|---|---|
| Solution Heat Exchanger | 1. $\dot{m}_1+\dot{m}_{27}=\dot{m}_9+\dot{m}_{26}$<br>2. $\dot{m}_1x_1+\dot{m}_{27}x_{27}=\dot{m}_9x_9+\dot{m}_{26}x_{26}$<br>3. $\dot{m}_1h_1+\dot{m}_{27}h_{27}=\dot{m}_9h_9+\dot{m}_{26}h_{26}$<br>4. $\dot{m}_1s_1+\dot{m}_{27}s_{27}=\dot{m}_9s_9+\dot{m}_{26}s_{26}+\dot{S}_{ger,TCS}$<br>5. $\eta_{TCS}=(\dot{m}_1h_1-\dot{m}_{26}h_{26})/\dot{m}_9h_9-\dot{m}_{27}h_{27}$<br>6. $\varepsilon_{TCS}=(\dot{m}_1ex_1-\dot{m}_{26}ex_{26})/(\dot{m}_9ex_9-\dot{m}_{27}ex_{27})$<br>7. $\xi_{TCS}=(\dot{m}_1ex_1+\dot{m}_{27}ex_{27})/(\dot{m}_9ex_9+\dot{m}_{26}ex_{26})$<br>8. $Ex_1^*+Ex_{27}^*=Ex_9^*+Ex_{26}^*$<br>9. $C_1^*+C_{27}^*=C_9^*+C_{26}^*+Z_{TCS}$ |  |
| Expansion Chamber | 1. $\dot{m}_{H,17}+\dot{m}_{18}=\dot{m}_{14}+\dot{m}_{15}+\dot{m}_{H,16}$<br>$\dot{m}_{18}=\dot{m}_{10}\cdot(1-x_{q,11})+\dot{m}_{23}=\dot{m}_{19}$ e $\dot{m}_{14}=\dot{m}_{10}\cdot(1-x_{q,11})=\dot{m}_{13}$<br>2. $\dot{m}_{H,17}+\dot{m}_{18}y_{10}=\dot{m}_{14}x_{11}+\dot{m}_{15}y_{10}+\dot{m}_{H,16}$<br>3. $\dot{m}_{H,17}h_{H,17}+\dot{m}_{18}h_{18}=\dot{m}_{14}h_{14}+\dot{m}_{15}h_{15}+\dot{m}_{H,16}h_{H,16}$<br>4. $\dot{m}_{H,17}s_{H,17}+\dot{m}_{18}s_{18}=\dot{m}_{14}s_{14}+\dot{m}_{15}s_{15}+\dot{m}_{H,16}s_{H,16}+\dot{S}_{ger,CE}$<br>5. $\eta_{CE}=(\dot{m}_{14}h_{14}+\dot{m}_{15}h_{15}-\dot{m}_{17}h_{17})/(\dot{m}_{18}h_{18}-\dot{m}_{16}h_{16})$<br>6. $\varepsilon_{CE}=(\dot{m}_{14}ex_{14}+\dot{m}_{15}ex_{15}-\dot{m}_{17}ex_{17})/(\dot{m}_{18}ex_{18}-\dot{m}_{16}ex_{16})$<br>7. $\xi_{CE}=(\dot{m}_{17}ex_{17}+\dot{m}_{18}ex_{18})/(\dot{m}_{14}ex_{14}+\dot{m}_{15}ex_{15}+\dot{m}_{16}ex_{16})$ |  |
| Evaporator + Gas Heat Exchanger | 1. $\dot{m}_{19}+\dot{m}_{H,20}=\dot{m}_{13}+\dot{m}_{H,22}+\dot{m}_{24}$<br>2. $\dot{m}_{19}y_{19}+\dot{m}_{H,20}=\dot{m}_{13}x_{13}+\dot{m}_{H,22}+\dot{m}_{24}y_{24}$<br>3. $\dot{m}_{19}h_{19}+\dot{m}_{H,20}h_{H,20}=\dot{m}_{13}h_{13}+\dot{m}_{H,22}h_{H,22}+\dot{m}_{24}h_{24}+\dot{Q}_{evap}$<br>4. $\dot{m}_{19}s_{19}+\dot{m}_{H,20}s_{H,20}=\dot{m}_{13}s_{13}+\dot{m}_{H,22}s_{H,22}+\dot{m}_{24}s_{24}+\frac{\dot{Q}_{evap}}{T_{18}}+$<br>$+\dot{S}_{ger,evap}$<br>5. $\eta_{evap}=\dot{Q}_{evap}/(\dot{m}_{19}h_{19}+\dot{m}_{H,20}h_{H,20}-(\dot{m}_{13}h_{13}+\dot{m}_{22}h_{22}+\dot{m}_{24}h_{24}))$<br>6. $\varepsilon_{evap}=\dfrac{\dot{Q}_{evap}\left(1-\frac{T_0}{T_{18}}\right)}{\dot{m}_{19}ex_{19}+\dot{m}_{H,20}ex_{H,20}-(\dot{m}_{13}ex_{13}+\dot{m}_{22}ex_{22}+\dot{m}_{24}ex_{24})}$<br>7. $\xi_{evap}=\dfrac{\dot{m}_{19}ex_{19}+\dot{m}_{H,20}ex_{H,20}}{\dot{m}_{13}ex_{13}+\dot{m}_{22}ex_{22}+\dot{m}_{24}ex_{24}+\dot{Q}_{evap}\left(1-\frac{T_0}{T_{18}}\right)}$<br>8. $Ex_{26}^*+Ex_{32}^*=Ex_{12}^*+Ex_{13}^*+Ex_{25}^*+Ex_{29}^*$<br>9. $C_{26}^*+C_{32}^*=C_{12}^*+C_{13}^*+C_{25}^*+C_{29}^*+Z_{evaporador,CE,absorvedor,reservatório}$ |  |

Sections 28 to 32 of Figure 2, are the exergy of heat flows that cross the areas of volumes of the control system, which respectively are: generator, evaporator, rectifier, condenser and absorber, which are quantified by the following equations:

$\dot{E}x_{28}=\left(1-\frac{T_0}{T_2}\right)\dot{Q}_{gerador}$ , $\dot{E}x_{29}=\left(1-\frac{T_0}{T_{18}}\right)\dot{Q}_{evaporador}$ , $\dot{E}x_{30}=\left(1-\frac{T_0}{T_{10}}\right)\dot{Q}_{retificador}$ , $\dot{E}x_{31}=\left(1-\frac{T_0}{T_{11}}\right)\dot{Q}_{condensador}$ e

$\dot{E}x_{32}=\left(1-\frac{T_0}{T_{26}}\right)\dot{Q}_{absorvedor}$ .

The symbols, in the Table (1), *x* e *y* representing respectively mixing in the liquid phase and gas. The sub-index "H" represents the gas pressure equalizer (which may be hydrogen or helium). This balance $\dot{m}_{23}$ is the vapor mass fraction of residual ammonia recirculating, and $x_{q,11}$ is the quality of the fluid at point 11.

Note that the mass flow of the fluid pressure equalizer is the same in the points: and that the flow of ammonia vapor in the residual are the same points:

$\dot{m}_{H,16} = \dot{m}_{H,17} = \dot{m}_{H,20} = \dot{m}_{H,21} = \dot{m}_{H,22}$ and that the flow of ammonia vapor residual are the same in points: $\dot{m}_{15} = \dot{m}_{23} = \dot{m}_{24}$

The irreversibility of control volumes for the set, can be calculated by the Gouy-Stodola theorem, Eq (1).

$$\dot{I}_{vc} = T_0 \cdot \dot{S}_{ger.vc} \geq 0 \tag{1}$$

The performance of the refrigeration system is analyzed through the COP (coefficient of performance) and is defined as the ratio between the heat removed by the evaporator and the heat supplied to the generator, Eq (2).

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{ger}} \tag{2}$$

The input data for the termoeconomic and exergoeconomic analysis ar: the cost of the machine for cooling investment, budgeted at R$ 3,500.00 and for return on investment of 10 years at an interest rate of 5%, these values, applied in determining the factor recovery of capital as described above.

The system of cooling by absorption, which is a compact unit, their total value is apportioned between its internal components. For such a factor is assigned to each subsystem to the refrigeration unit for absorption. The program developed allows these factors are modified and so it can simulate other situations. For the case study, given the factors are in Table (2), which were based on the work of Santos (2005a) and Marques (2005).

Table 2. Factor of the investment division of the refrigeration system Einstein proposed.

| **Subsystem** | **Factor (%)** |
|---|---|
| Generator | 20 |
| Bubble pump | 5 |
| Rectifier | 5 |
| Solution Heat Exchanger | 10 |
| Pre-cooler + Evaporator + Condenser/absorber | 60 |
| **Total** | 100 |

## 4. RESULTS AND DISCUSSION

The results of the analysis of absorption refrigeration cycle distribution by computer code, are shown in Figs. (3) and (4).

The Figure (3) shows the efficiencies of the first and second law of thermodynamics and the Degree of Thermodynamics Perfection for each component.

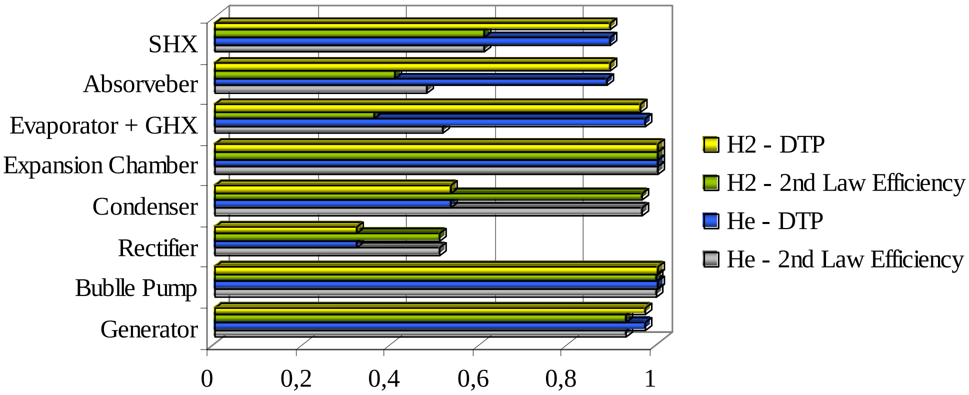


Figure 3. Performance by the 2nd Law Thermodynamics and the Degree of Perfection.

The addition of energy analysis is expressed by the evaluation exergetic, where the principle of exergy allows the identification of the irreversibility of the system and the detection of the loss more pronounced. These results enable the investment to improve the system is directed to points where the response will be more expressive.

The Figure (4) shows the irreversibility of each component of the refrigeration system before the total irreversibility of the equipment. This is the essence of the analysis by the Second Law of Thermodynamics, it tells where the greatest losses occurring in the system, exergetics destruction, thus indicating in which volume control, efforts should be concentrated for improvement.

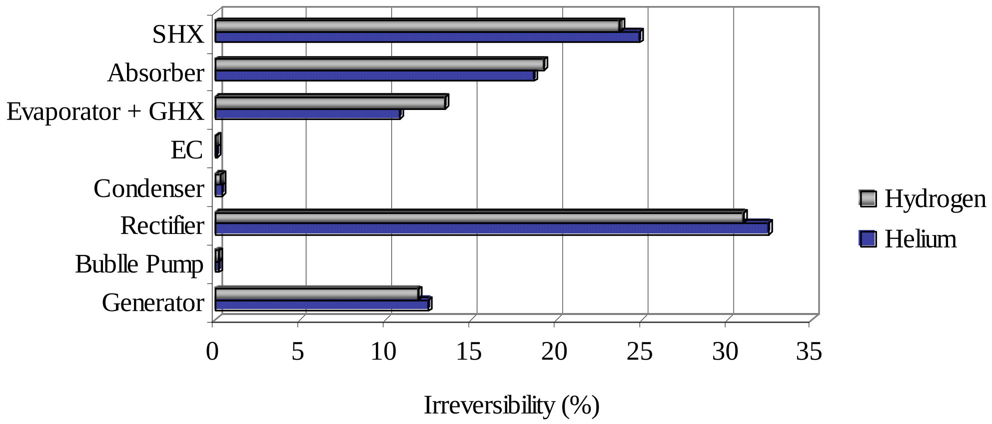


Figure 4. Irreversibility  each component on the total system cooling. Temperature of the generator set at 195 º C.

The rectifier, Fig (4), is responsible for the largest portion of the irreversibility generated in the cycle for the temperature rise of 195 º C, due to non recovery of heat rejected by the equipment. Have the generator as part of a share of irreversibility of the system. This is due to several factors: i) is the generator which have the highest temperatures of the system ii) the process of desorption, in other world, the separation of the vapor of ammonia solution is an intrinsically irreversible process to involve different chemical species.

The equipment responsible for the absorption of the vapor condenser and heat exchangers of solution and gas (with the evaporator) stand out as equipment that generate high rates of irreversibility. This was expected, not only by the level of temperature and, by the nature of the chemical reaction that occurs in volumes of control.

The results of irreversibility obtained in this work indicate that the equipment in the process of desorption (generator and rectifier) and absorption require investments aimed at improving this type of technology to generate cold.

For the monetary balance, the total cost in the output of the unit must be equal to the total cost of entry plus the cost of investment. The Table (3) evidence that balance sheet.

Table 3. Monetary costs to the refrigeration unit.

| **Input** | | | **Output** | |
|---|---|---|---|---|
| Flow | ($10^3$ R$/year) | | Flow | ($10^3$ R$/year) |
| 28 | 0.34 | Refrigeration Unit | 29 | 0.26 |
| | | | 30 | 0.12 |
| | | | 31 | 0 |
| Investment | 0.04 | | 32 | 0 |
| **Total** | 0.38 | | **Total** | 0.38 |

Other scenarios can be obtained for the analysis of investment costs involved in system and depreciated over time. The Table (4) shows the results of the cost for an interest rate of 8, 10 and 12% and for a period of operation of 10, 15 and 20 years. Observe that the cost of the product does not suffer a significant variation.

Table 4. Scenarios monetary system cooling.

| | Time Operation (years) | 10 | | | 15 | | | 20 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | Interest Rate (%) | 8 | 10 | 12 | 8 | 10 | 12 | 8 | 10 | 12 |
| Investment | Refrigerator (R$/year) | 52.16 | 56.96 | 61.94 | 27.26 | 30.68 | 34.26 | 17.82 | 20.56 | 23.43 |
| Product | Cold – Flow (29) (R$/year) | 267.93 | 272.36 | 276.96 | 244.95 | 248.11 | 251.41 | 236.24 | 238.76 | 241.42 |

## 5. CONCLUSION

Increased system performance can be achieved with the optimization of the rectifier, the heat exchanger of the solution, the absorber and the evaporator, since these components showed lower efficiencies rational.

The components most vulnerable to changes in temperature of the generator, producing higher irreversibility are the rectifier and the heat exchanger of the solution.

Have the results of the exergetic analysis was possible the exergoeconomic analysis, which provides results for exergetics and monetary costs flows of the unit. To do this, put up the input to the refrigeration unit and investment in equipment. Maintenance costs for equipment were not covered in this study. The analysis of depreciation of capital employed in this system over its useful life was developed for a net interest rate of 5% and for a time of use for 10 years. Therefore, it is possible to spread the investment over time and records it in the analysis. Thus, for the refrigeration system through absorption-diffusion there is a depreciation of R$ 45.33 per year.

This study means a development of own technology for this kind of system that can assist to the needs of economy of energy and for application in hotel rooms, offices, camping and as recreative vehicle refrigerator.

## 6. ACKNOWLEDGEMENTS

We present our thanks to CNPq for financial support through scholarships, as well as FINEP, Petrobras and RECOGÁS by financial support for the project.

## 7. REFERENCES

ANEEL, Agência Nacional de Energia Elétrica, 2005, Atlas de energia elétrica do Brasil. 2ª ed. Brasília.

Bejan A., Tsatsaronis G., Moran M., 1996, “Thermal Design and Optimization”, 1st Ed. New York, Wiley & Sons.

Chen, J., Kim, K. J., Herold, K. E., 1996, “Performance Enhancement of a Diffusion-Absorption Refrigerator”, International Journal of Refrigeration – Elsevier, v. 19, n.3, pp. 208-218.

ELETROBRÁS, 2005, Programa nacional de universalização do acesso e uso da energia elétrica: manual de operacionalização. Versão 2.0, ELETROBRÁS, MME.

Herold, Keith E., Radermacher, Reinhard, Klein, Sanford A., 1996, Absorption Chillers and Heat Pumps. Florida, New York, USA, CRC Press.

IBGE, 2006, PNAD – Pesquisa nacional por amostra de domicílios: Síntese de indicadores 2005. IBGE – Instituto Brasileiro de Geografia e Estatística, Rio de Janeiro, RJ.

Jornal Gazeta Mercantil, 22/02/2002, “Caderno Agronegócios”, www.gazeta.com.br, São Paulo.

Koyfman, A., Jelinek, M., Levy, A., Borde, I., 2003, “An Experimental Investigation of Bubble Pump Performance for Diffusion Absorption Refrigeration System with Organic Working Fluids”, Applied Thermal Engineering – Pergamon, v. 23, pp. 1881-1894.

Lozano, M. A., Valero, A., Guallar, J., 1989, Teoria del Custe Exergético. Universidad Zaragoza, Spain.

Marques, A. S., 2005, Análise Termoeconômica de um Sistema de Refrigeração Automotivo por Absorção. Dissertação de M.Sc, CT/UFPB, João Pessoa, PB, Brasil.

MME, Ministério de Minas e Energia, 2006, Balanço Energético Nacional. Brasília.

Santos, C. M. S., 2005a, Análise Exergoeconomica de uma Unidade de Cogeração a Gás Natural com Refrigeração por Absorção. Dissertação de M.Sc., CT/UFPB, João Pessoa, PB, Brasil.

Souza, Wilson L. de, 2007, Analise Exergoêconomica dos Ciclos de Refrigeração por Absorção de Única Pressão. Tese de doutorado, CT/UFPB, João Pessoa, PB, Brasil.

Srikhirin, Pongsid, Aphornratana, Satha, 2002, “Investigation of a Diffusion Absorption Refrigerator”, Applied Thermal Engineering – Pergamon, v. 22, pp. 1181-1193.

Srikhirin, Pongsid, Aphornratana, Satha, Chungpaibulpatana, Supachart, 2001, “A review of Absorption Refrigeration Technologies”, Renewable and Sustainable Energy Reviews – Pergamon, v. 5, pp. 343-372.

Valero, A., Lozano, M. A., Muñoz, M., 1986, “A General Theory of Exergy Saving: Part I. On the Exergetic Cost, Part II. On the Thermoeconomic. Part III. Energy Saving and Thermoeconomics”. In: Computer-Aided Engineering Energy Systems, vol. 3, Second Law Analysis and Modelling, R. A. Gaggioli, ed. ASME, New York.

Zohar, A., Jelinek, M., Levy, A., Borde, I., 2005, “Numerical Investigation of a Diffusion Absorption Refrigeration Cycle”, International Journal of Refrigeration – Elsevier, v. 28, pp. 515-525.

Zohar, A., Jelinek, M., Levy, A., Borde, I., 2005, “The Influence of Diffusion Absorption Refrigeration Cycle Configuration on the Performance”, International Journal of Refrigeration – Elsevier, Article in Press.

## 8. RESPONSIBILITY NOTICE

:

The authors are the only responsible for the printed material included in this paper.