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NUMERICAL INVESTIGATION OF THE DIFFUSION ABSORPTION CYCLE TO PRODUCE COLD USING SOLAR ENERGY AS SOURCE

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Abstract. *This study aims to present a numerical investigation of the cold production by a diffusion absorption cycle, using NH_3-H_2O as the working fluids and H_2 as pressure equalizer, aiming cold production using solar energy as the source. A complex thermodynamic analysis was done in order to calculate the mass flow, rate heats. To assure the analysis was elaborate obeying all the thermodynamic laws, entropy, enthalpy, and exergy was also calculated. So to conclude the thermodynamic analysis was calculated the inlet, outlet, irreversibilities and the efficiency of all the components in the system. After the first analysis, some parameters were varied in order to achieve the maximum coefficient of performance and the highest refrigeration rate. The parameters include the poor partial pressure, the system pressure, and the evaporator temperature. So it was found the best work cycle condition, and then this state was analyzed in order to see the impact of the evaporator temperature variation. So it was possible to detect some important values and inputted data to get the project solved. The software Engineering Equation Solver was used, which showed the most significant values for this project, the input heat Q_{GEN} = 692 W and the output heat Q_{EV} = 227 W, and also the system coefficient of performance, COP = 0,33.*

Keywords: *Absorption, diffusion, solar, hydrogen, ammonia.*

1. INTRODUCTION

The access to electricity is considered primordial for minimal living conditions. Recently, Steve McCarney, et al., (2013) analyzing data presented by International Energy Agency (IEA), presented that approximately of 1.3 billion people didn't have access to electricity in 2010, more than 85% of this number represents people who live in rural area. In countries where electric energy consumption is less than one ton equivalent of petrol per year the numbers of child mortality is too high and the life expectancy is too low (Martins, et al., 2003). The lack of right condition to store the food is one of the main problems for those people who don't have access to electricity, the work published by Metcalf, et al., (2009) presents a system that produces ice using solar energy as source and its production could be over than 90 kilos of ice daily, in places like city of Dakar in Senegal, where it would be amazing having this system because fishing is the sole or main source of economic activity they practice as a way to survive, and a system like this could help keep the product fresh for a longer period of time.

In order to attend changes in market's demand and achieve the sustainable and clean energy production, several studies have been developed, according to Praene, et al., 2010, the government of the Reunion Island has a project which the main goal is to have a total electricity independence settled until 2030 using only renewable energy. In a recent work (Li, et al., 2013) showed the use of a simulation of absorption cycle based on Water, lithium, and bromide (H_2O-LiBr) and the results showed that could be possible to spare more than 32% of electricity consumed by air conditioning systems in China.

Analyzing the recent studies of refrigeration in the production of cold or ice, some of them has shown interesting results. Izquierdo, et al., 2014 studied the idea of an air conditioning system using absorption refrigeration system to produce 4.5 to 7 kilowatts (kW) of energy. During the study, they concluded that this system could cover more than 65% of the demand from Madrid University in Spain.

Hamed, et al., (2011) tried to improve a solar absorption system in a transient model and got to the conclusion of the advantages of this system, such as the possibility of using heat from any source, it is noise free and do not cause any damage to the ozone layer.

The first mentions about diffusion-absorption refrigeration (DAR) cycle were around of 1920 by two Swedish (Hildbrand, et al., 2004). The huge difference between this cycle and a compression cycle is that the diffusion-absorption cycle works without any mechanical work input (Hildbrand, et al., 2004). The cycle utilizes ammonia as the refrigerant, water as absorbent and an inert gas that is in charge to equalize the pressure of the system (Zohar, et al., 2004). The fact that this equipment does not have any move parts is another advantage, since it is noise free.

This numerical investigation has the purpose to elaborate a thermic analysis of a diffusion absorption refrigeration system using ammonia, water, and hydrogen (NH_3- H_2O – H_2) to produce cold or ice and using solar energy as the source; an inert gas (hydrogen) is used to equalize internal pressure of the system, as consequence the inert gas exclude the necessity to use any mechanical or electrical energy, also ammonia was used as refrigerant and water as absorbent.

2. METHODOLOGY

This study proposes a thermodynamic analysis of a DAR system using solar energy as source. Huge amounts of studies about this cycle have been developed in the last years, with the main point of increase COP and refrigeration rate, at the same time, decrease the costs of production in large scale. According to Hildbrand, et al., 2004 this system has never come up from the shadows because of the inherent irreversibilities, bigger than those found in normal compression cycles, which are increased by the high resistance of mass transfer in function of hydrogen presence. The irreversibilities on this cycle are derived from the interaction between of the three fluid present on the system, the interaction between the fluids and the pipes, and all the heats exchange involved on this DAR system.

However, the changes in the market demands and the laws about the environment has been making necessary to improve those technologies that use clean energy as the DAR system (ZEYU, et al., 2013). So, to developed this study was chosen the software Engineering Equation Solver (EES) that has an extensive library and allows to calculate per mathematic interactions the thermodynamic properties that are necessary to determine the heat rates ($\dot{Q}$), enthalpy (h), entropy (s), exergy (e) and others peculiarities of the cycle obeying all the thermodynamic laws.

To create a better understanding it’s necessary to know the function of all equipment of the cycle. So Fig. (1) presents all the outfit used in this system, the generator is responsible for input the heat from an external source on the system. The bubble pump is one pipe that pass inside the generator and is where start the separation of ammonia and water because of the heat inputted on the system. The rectifier is where the rest of the water vapor is removed by one process equal of a usual heat exchanger. On the condenser, the ammonia interacts with the environment through the pipes and exchange heat decreasing your temperature. In all those outfits previously presented flows only ammonia and water.

On the other hand, for all the equipment presented below, through them flows water, ammonia, and hydrogen. On the expansion chamber, occur the pressure drop because of the interaction of the three fluids, residual ammonia, hydrogen, and pure ammonia. The evaporator works similar to one common refrigerator, inside has one fluid (water, air) which will be cooled, because this fluid will exchange heat with the mixture of ammonia and hydrogen that presents a lower temperature, producing ice or air refrigeration. On the absorber is where ammonia vapor is absorbed in the weak solution of ammonia and water. The reservoir is the responsible for store all the fluid.

The solution heat exchanger and the gas heat exchanger are responsible for decrease some possible irreversibilities, but they are not indispensable for the cycle, because the cycle can works without them but probably the coefficient of performance would be lower.

Figure 1 presents a schematic example of a diffusion-absorption cycle. In the beginning of the cycle at point 1 is where the heat (source power) goes into the cycle and occur the separation of ammonia vapor and rich solution (high concentration of ammonia), then ammonia vapor pass through the bubble pump and at (2 and 2v) ammonia-water and weak solution (low concentration of ammonia) leaves.

So water vapor condenses, Fig. (1) point 4, join the weak solution and goes to the absorber passing through solution heat exchanger, the pure ammonia ,after the rectifier (water separator), flows to the condenser where it exchanges heat with the surroundings and flows to the evaporator, uncondensed ammonia goes to the absorber by the gas bypass.

Ammonia sub-cooled, Fig. (1) point 7, crosses hydrogen and residual ammonia coming from the absorber passing through evaporator and gas heat exchanger and pressure fall. Then the solution, ammonia-hydrogen goes in parallel passing the shell side of the evaporator toward the reservoir exiting at point 12.

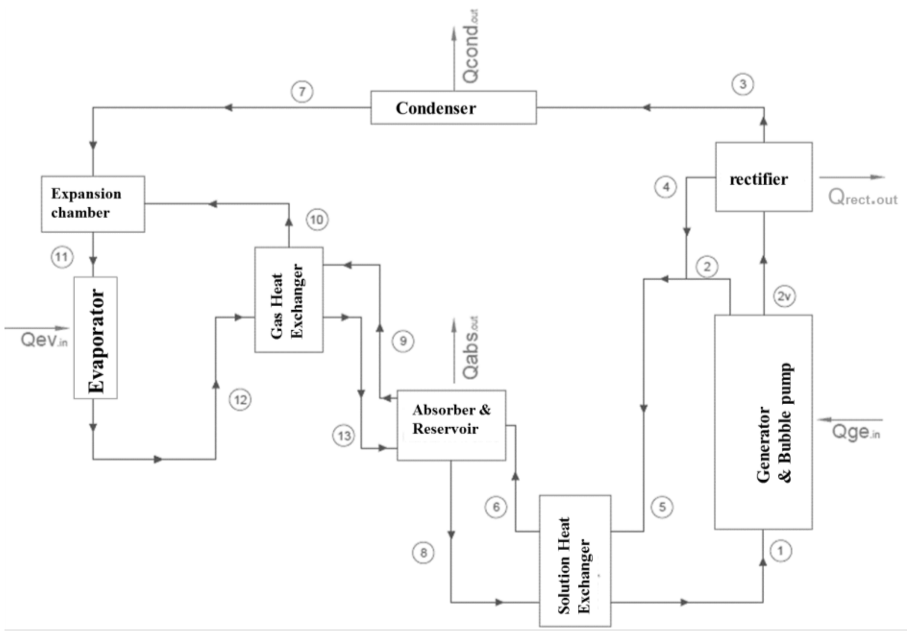

Figure 1. Hardware of diffusion-absorption refrigerator hardware schematic

A huge number of variations on the diffusion-absorption cycle have been manufactured, sometimes changing the type of the flow inside the evaporator, the position of the gas heat exchanger, the inert gas and some other small things (Zohar, et al., 2004). All these modifications are to increase the coefficient of performance (COP) of the DAR system, which represents the ratio of cooling capacity Q_{evap} and Q_{gen}. Where Q_{evap} represents the amount of cold that can be produced and Q_{gen} represents the amount of heat that was introduced in the system on the generator (Fig.1), and by these two values, it is possible to determine how well the cycle works.

2.1 Thermodynamic analysis

The mass and energy balance equation for all the elements of the system are presented below. The numbering of the different properties are related to the location indicated in Fig. (1). In order to elaborate the right thermodynamic analysis the use different equations was necessary, in order to achieve the main points of this study. To calculate all the required properties different references were used like Moran and Shapiro, 2006, Almén, 1985. Thus, the following equations below:

$$\sum_i \dot{m}_{in} = \sum_i \dot{m}_{ex} \tag{1}$$

Where ṁ represents the mass flow rate, the index in and ex represent inlet and exit of the component that is in analysis and the index i represents the number of the state. Equation (1) was used to calculate the mass flow rate in all the components of the system. Some modifications were added at Eq. (1) to make possible to calculate x_s (ammonia concentration on the strong solution), x_w (ammonia concentration on the weak solution) and y (concentration of ammonia vapor) Eq. (2).

$$\dot{m}_{in} * x_{in} = \dot{m}_{out} * x_{out} \tag{2}$$

To calculate the heat exchanges on the generator, rectifier, condenser, evaporator, gas and solution heat exchanger, and absorber was used the Eq. (3):

$$\dot{Q}_{equipment} = \sum_i \dot{m}_{in} * h_{in} - \sum_i \dot{m}_{ex} * h_{ex} \tag{3}$$

Where $\dot{Q}$ equipment is the heat that is being exchanged and h_{in} and h_{ex} are the enthalpy on the inlet and on the exit of the equipment. In some cases, the heat on the system is positive and some is negative, when it is negative is said that the heat was transferred from the system, on the other hand, when it is positive we say that the heat was transferred to the system (Moran and Shapiro, 2006). However, to analyze the mass flow rate at the evaporator in function of hydrogen, ammonia vapor, and gas presence was necessary the use of Eq. (4) and Eq. (5)

$$Z_{rich} = \frac{(M_{NH3} * P_{prich})}{M_{H2} * (P_{sys} - P_{prich})} \tag{4}$$

$$Z_{poor} = \frac{(M_{NH3} * P_{ppoor})}{(M_{H2} * (P_{sys} - P_{ppoor})} \tag{5}$$

Where Z_{rich} represents the ammonia charge in the hydrogen on the rich solution, Z_{poor} is ammonia charge in the hydrogen on the poor solution. M_{NH3} and M_{H2} represents molar masses of ammonia and hydrogen, and they have a default value presented in all basic chemistry books. Finally, P_{sys}, P_{ppoor}, and P_{prich} they match respectively to system pressure, poor partial pressure and rich partial pressure. So, it was possible to calculate the quantity of hydrogen mass that exits expansion chamber on point 12 (Fig. 1), all these data were presented by Almén (1985), thus by Eq. (6) was calculated the charge of hydrogen on state 12:

$$\dot{m}_{12H2} = \frac{\dot{m}_8}{Z_{rica} - Z_{poor}} \tag{6}$$

Where $\dot{m}_{12H2}$ represent the charge of hydrogen on state 12 as the same way of $\dot{m}_8$ represents the mass flow on state 8. To calculate the value of enthalpy, entropy and exergy of states 10 to 14 was necessary to use different equations, which EES doesn't have on its literature, this equation was collect at a reputed book of Thermodynamic Engineering Moran and Shapiro, 2006. The equations presented below show h for the specific enthalpy and s for the specific entropy Eq. (7) and Eq. (8), respectively:

$$h\ [i]=y[i]*\ h_{NH3[i]}+(1\text{-}y[i])*\ h_{H_2\ [i]} \tag{7}$$

$$s\ [i]=y[i]*\ s_{NH3[i]}+(1\text{-}y[i])*\ s_{H_2\ [i]} \tag{8}$$

In order to determine the specific flow exergy of the states 1 to 8, it was used Eq. (9) presented below, Moran and Shapiro (2006):

$$ex_f\ [i]=h[i]\text{-}\ h[0]\text{-}\ T_0*(s[i]\text{-}s[0]) \tag{9}$$

Where e_f represents the specific flow exergy, T_0 is the temperature of dead state, all the characteristics related to the [0] are correlated with the dead state. To calculate specific flow exergy of the states 10 to 14 were used Eq. (10) and Eq. (11):

$$h_{[0-1014]} \ = y_{[0]} * h_{[0-NH3]} + (1 - y_{[0]}) * s_{[0-H2]} \tag{10}$$

$$e_{f[i]} = h_{[i]} - h_{[0-1014]} - T_{[0]} * (s_{[i]} - s_{[0-NH3]}) \tag{11}$$

Chemical exergy is, by definition, the maximum theoretical work that could be developed by the combined system (Moran and Shapiro (2006). So, to calculate the specific chemical exergy of the states 1 to 8 were used Eq. (12) and Eq. (13):

$$ex_{qH2O}\ [i]=(1\text{-}x[i])*(\frac{3120}{M_{H2O}}) \tag{12}$$

$$ex_{qNH3}\ [i]=(x[i])*(\frac{341250}{M_{NH3}}) \tag{13}$$

So eq_{H2O} and eq_{NH3} represent, respectively, specific chemical exergy of water and ammonia, the values 3120 kJ mol^{-1} and 341250 kJ mol^{-1} represent the standard chemical exergy of water and ammonia. The same way to calculate specific chemical exergy of the states 9 to 14 were necessary to use different equations, Eq. (14) and Eq. (15):

$$ex_{qH2}\ [i]=(1\text{-}y[i])*(\frac{238490}{M_{H2}}) \tag{14}$$

$$ex_{qNH3}\ [i]=(y[i])*(\frac{341250}{M_{NH3}}) \tag{15}$$

So e_{qH2} represents specific chemical exergy of hydrogen, 238490 kJ mol^{-1} is the standard chemical exergy of hydrogen. In this way, the total exergy is the sum of specific flow exergy and specific chemical exergy. To guarantee the right exergetic analysis, it was necessary to determine the irreversibilities of the system using the inlet exergy (input) and the outlet exergy (product and tailings). So, the following equations were used to calculate the input and the outlet.

To all the equations below $\dot{F}_1$ represents the inlet exergy $\dot{P}_1$ represents outlet exergy, $\dot{m}$ the mass flow according to the index following, ex represents the exergy on the state of analysis. Some of the information on the equations were presented before, and some of them are presented in Fig. (1). The inlet exergy equations below are from bubble pump, rectifier, solution heat exchanger, condenser, expansion chamber, evaporator, gas heat exchanger, and absorber and reservoir respectively:

$$\dot{F}_1 = \dot{m}_1 * (ex_1 - ex_2) + \left[1 - \frac{T_0}{T_{bumble}}\right] * \dot{Q}_{gen} \tag{16}$$

$$\dot{F}_2 = \dot{m}_3 * ex_3 - (\dot{m}_4 * ex_4 + \dot{m}_5 * ex_5) \tag{17}$$

$$\dot{F}_3 = \dot{m}_6 * ex_6 - \dot{m}_7 * ex_7 \tag{18}$$

$$\dot{F}_4 = \dot{m}_4 * ex_4 - \dot{m}_8 * ex_8 \tag{19}$$

$$\dot{F}_5 = \dot{m}_{11} * (ex_{11} - ex_{12}) \tag{20}$$

$$\dot{F}_6 = \left[1 - \frac{T_0}{T_{ev}}\right] * \dot{Q}_{ev} \tag{21}$$

$$\dot{F}_7 = \dot{m}_{14} * ex_{14} - \dot{m}_{13} * ex_{13} \tag{22}$$

$$\dot{F}_8 = \dot{m}_9 * (ex_{14} - ex_9) \tag{23}$$

As the same way that was presented before, the following outlet exergy equations are from bubble pump, rectifier, solution heat exchanger, condenser, expansion chamber, evaporator, gas heat exchanger, and absorber and reservoir respectively:

$$\dot{P}_1 = \dot{m}_3 * (ex_3 - ex_1) \tag{24}$$

$$\dot{P}_2 = \left[1 - \frac{T_0}{T_{ret}}\right] * \dot{Q}_{rect} \tag{25}$$

$$\dot{P}_3 = \dot{m}_1 * ex_1 - \dot{m}_9 * ex_9 \tag{26}$$

$$\dot{P}_4 = \left[1 - \frac{T_0}{T_{cond}}\right] * \dot{Q}_{cond} \tag{27}$$

$$\dot{P}_5 = \dot{m}_8 * (ex_{12} - ex_8) \tag{28}$$

$$\dot{P}_6 = \dot{m}_{13} * ex_{13} - \dot{m}_{12} * ex_{12} \tag{29}$$

$$\dot{P}_7 = \dot{m}_{10} * ex_{10} - \dot{m}_{11} * ex_{11} \tag{30}$$

$$\dot{P}_8 = \dot{m}_7 * (ex_{10} - ex_7) + (1 - (\frac{T_0}{T_{abs}})) * \dot{Q}_{abs} \tag{31}$$

After determining the data of inlet and outlet, it is possible to calculate the irreversibilities and the efficiency of all equipment, then, the equations used were Eq. (32) and Eq. (33), as showed below:

$$I = \dot{F} - \dot{P} \tag{32}$$

$$\eta = \frac{\dot{P}}{\dot{F}} \tag{33}$$

Where I represents the irreversibilities and η represents the efficiency. All the values that were calculated by the equations mentioned before are presented in Tab. 2 on section 3.

So, when the heat exchange on the evaporator and on the generator were determined, it was possible to calculate the coefficient of performance. The COP is defined as the ratio between the heats removed by the evaporator (thus creating the refrigeration effect) to that supplied at the generator, so using Eq. (34):

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \tag{34}$$

Where $\dot{Q}_{gen}$ is the amount of heat provided from an external source, in this study a solar collector, and the $\dot{Q}_{evap}$ is the refrigeration rate.

All the values mentioned before related to the index 0 represents one reference state chosen to be used in order to calculate all characteristics necessary. The value of T0 chosen was 300 K and the value chosen of pressure was 1 bar, some of the equations presented before needed of this value to calculate the final results.

2.2 Simulation of cycle using Engineering Equation Solver (EES)

All the essential equations to calculate all the characteristics of the cycle was written on the code line inside the EES, the equations mentioned before were included on this code line. However some specific data were necessary, as they are present in Tab. 1 along with the parameters that were varied in order to achieve the maximum coefficient of performance and the highest refrigeration rate:

Table 1. Input parameters for the analyses.

Parameters	**Value**	**Variable Range**
Ammonia mass flow	2.8 $*10^{-4}$	-
System pressure	- Bar	15 to 23
ammonia concentration on strong solution	0.3	-
ammonia concentration on weak solution	0.1	-
Concentration of ammonia vapor[Y3]	0.99	-
Temperature 3	105 ºC	
Poor partial pressure	- Bar	1.5 to 3.5
Rich partial pressure	4.3 bar	-
Temperature 1	60 ºC	-
Temperature 2	105 ºC	-
Out of rectifier only ammonia [y4]	1	-
Evaporator temperature	-	-22 to -12

More than inputted data, it was crucial to insert at EES some hypotheses to guarantee the cycle working as it was planned:

- The ammonia vapor on the inlet of the condenser is pure;
- The condenser temperature is determined by the pressure of the system;
- The ammonia vapor and the weak solution left the generator at the same temperature;
- Between point 1 and 2 Fig. 1 there is a delta of 5 ºC (Levy, et al., 2005);
- There is a delta of 15 ºC between the inlets and outlets of the solution heat exchanger;
- The temperatures T4 and T5 are the same;
- The heat exchangers are adiabatic, which means that they are isolated of the extern environment;
- Hydrostatic pressure is negligible (Pereira and Osaki , 2103);
- Charge losses are considered negligible;
- All the properties of the gases were calculated considering the law of the Ideal Gas Model;
- On the exit of the absorber the ammonia partial and system pressure are equal (Yousuf, et al., 2014);
- Ammonia concentration was established as strong at states 1.5 and 9;
- Ammonia concentration was established as weak at states 6 and 7;

During the research, in the process of the analysis, some parameters were identified, which affects more on the COP and refrigeration rate of the cycle. During the analysis, using software EES, some variation were applied on these parameters in order to achieve the maximum COP and refrigeration rate.

The parameters were system pressure, the gas poor partial pressure and the inlet evaporator temperature. The values chosen to be variable were based in some studies that were developed by some authors. The poor partial pressure was studied by Chen, et al., 1996 and he found the value of 0.3 — where you achieve the maximum coefficient of performance. Almén, 1985 affirms that the ideal pressure system should be around of 23 bar and that the range of evaporator temperature should be -22 to -5.

Solar collectors

It's possible to define solar collector as a special type of heat exchanger that has the ability to transform the radiation of the sun into heat (Duffie and Beckman, 2013). But, of course, the solar collectors are really different from usual heat exchangers. During the elaboration of a study (Duffie and Beckman, 2013) were divide the solar collectors into two distinct models, the Flat-Plate Collectors and the Concentrating Collectors.

The flat-plate collectors are used where the temperatures of the equipment are lower than 100 ºC, this type is used a lot to heat water, heat indoors environment and in some heating industrial process. On the other hand, the Concentrating Collectors can achieve temperatures higher than Flat-plate Collectors, which are 400 ºC using thermal oil, however, the costs of the Concentrating Collectors are higher, too. The main geometry difference between these two collectors mentioned before is that the first one has a flat surface, where the pipes are attached, and the second one the area of absorption are composed by the pipe, where it circulates the fluid, and the parabolic reflector surface.

As the aim of this study is to analyze the cycle from the thermodynamic point of view, was chosen as the source for the system a concentrating collector, which can provide the necessary energy on the inlet of the system (generator).

3. RESULTS AND DISCUSSIONS

After all the simulation on the EES, all the expected results were calculated. Table 2 and Tab. 3 presented below show the most important results.

Table 2. Results got from Engineering Equation Solver

Component	F (kJ kg $^{-1}$)	P(kJ kg $^{-1}$)	I (kJ kg $^{-1}$)	Efficiency	Heat (kW)
Bubble pump	5.0454	3.9559	1.0895	0.7841	0.6919
Rectifier	0.0097	0.0083	0.0013	0.8602	0.0527
Solution heat exchanger	0.0241	0.0089	0.0152	0.3677	-
Condenser	0.0241	0.0241	0.0000	1.0000	0.2868
Expansion chamber	8.0709	7.9492	0.1218	0.9849	-
Evaporator	0.0314	0.0314	0.0000	1.0000	0.2268
Gas heat exchanger	0.1594	0.0048	0.1546	0.0300	-
Absorber and reservoir	53.3195	50.1120	3.2074	0.9398	0.4529

Following all the thermodynamic laws, the inlet, outlet, irreversibilities and the efficiency of the components of the cycle were calculated and, as expected, the irreversibilities were equal or upper than zero and all the values of inlet were bigger than values of the outlet. So at this point, it's possible to say that from the thermodynamic point of view the cycle it's viable.

Some inputted data were varied and some the results are presented in Fig.3 and Fig. 4 below. In the Fig. 3 some parameters were used to simulate, they are poor and rich partial pressure of 1.5 and 4.3 bar, combined with a system pressure variation between 15 and 23 bar. On the other hand, in the Fig. 4 the value of poor and rich partial pressure were 3.5 and 4.3 bar, and, in the same way, with a system pressure variation between 15 and 23 bar.

Figure 3 (a) and Fig. 4 (a) present the variation of the system pressure and the impact of this variation on the coefficient of performance, which was a linear progress. Thus, during this analysis, it was concluded that the higher the system pressure used, the higher the COP, however the increase of system pressure has a limit, and the limit is the value of the 25 bar, as mentioned by Almén, 1985 — when you use a system pressure higher than 25 bar the irreversibilities on this system is so considerable that makes the system inefficient.

Figure 3 (b) and Fig. 4 (b) present the alteration of the heat generator, condenser, and evaporator, when varied the system pressure. When the value used was in a range of 23 to 25 bar, a small drop on the numbers was noticed, which may have been caused by the system configuration chosen in thermodynamic analysis. When the value is bigger than 25 bar some values grew up and some decreased in a way that make the simulation of the cycle impracticable.

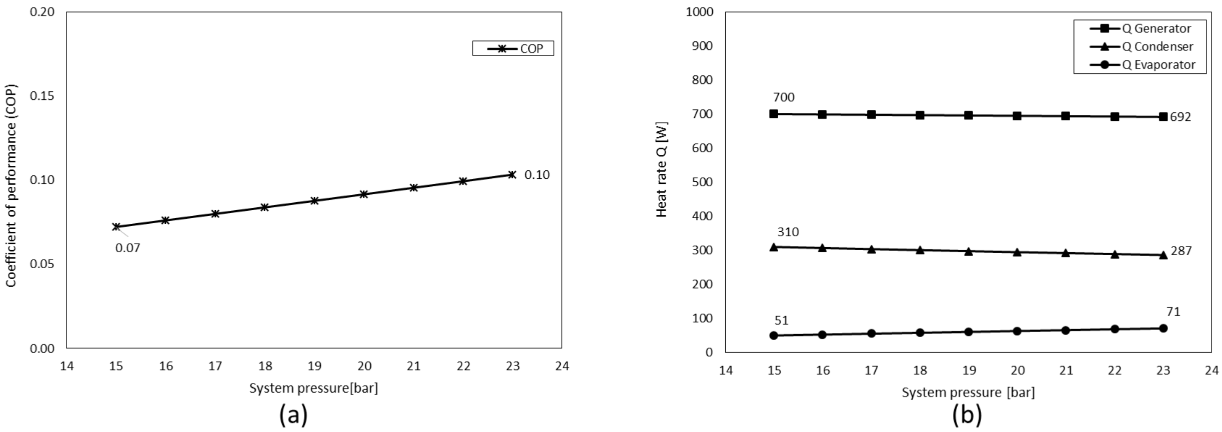

Figure 3. Variation of system pressure using poor partial pressure of 1.5.

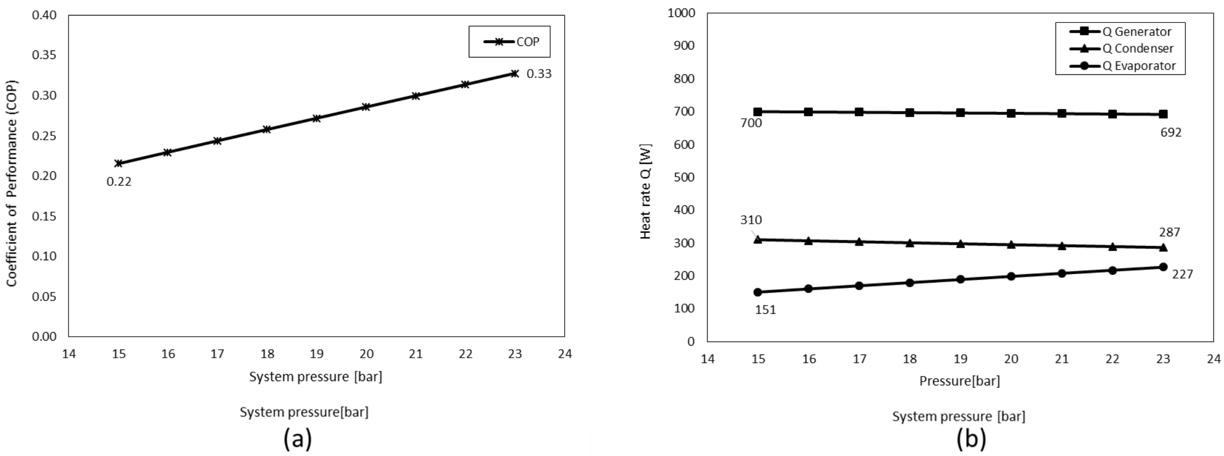

Figure 4. Variation of system pressure using poor partial pressure of 3.5

As seen in the Fig. 4 (a) a bigger coefficient of performance (0.33) was obtained using the conditions mentioned above, poor and rich partial pressure of 3.5 and 4.3 bar. So, these condition was chosen to be better analyzed, first of all, because it was found a smaller value of heat inputted on the generator. Secondly, the same thing happened to the condenser, and thirdly, the refrigeration rate (evaporator heat) was bigger. These conditions can be justified because of the increasing of the poor partial pressure. When you have a small variation range of poor and rich partial pressure the irreversibilities found on the generator, condenser, and evaporator are smaller than when you use a bigger range, as presented in the Fig. 3 a range of 1.5 to 4.3.

So, it is found that next to the pressure of the 23 bar the best conditions were achieved, and since the main point of this thermodynamic study is to achieve the maximum COP and at the same time the best rate refrigeration, this point was chosen to be deeper analyzed, the results obtained with this second analysis is shown in Fig. 5. All the facts mentioned before can justify a possible lower cost manufacturing of this equipment.

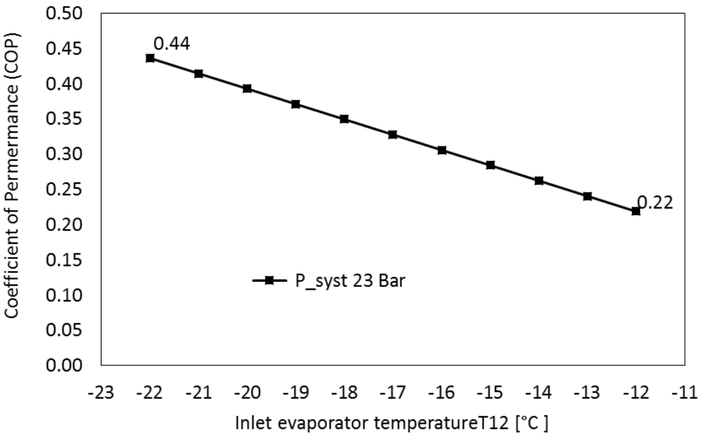

Figure 5. Impact of the inlet evaporator temperature on the coefficient of performance

Analyzing Fig. 5 it’s clear that the lowest temperature on the evaporator inlet promotes a bigger refrigeration rate keeping the same outlet evaporator temperature. This happens because the increasing of the evaporator range of inlet and outlet so, in this way, is possible to produce the biggest quantity of cold.

Another factor that is important to say is that the diffusion absorption cycle, as any other refrigeration cycle, have as the main point to produce cold and the more heat is inputted on the generator, the bigger it’s going to be the refrigeration rate, but this can have bad consequences as the increasing of the irreversibilities.

4. CONCLUSION

By using the thermodynamic model presented previously, the heat exchanges were calculated in all components of the cycle, and by modifying some inputted data, it was possible to increase the results. It was also determined the coefficient of performance, all the possible irreversibilities and other thermodynamic characteristics, ruled by all the thermodynamic laws validating this article.

Well, a future manufacturing of this DAR system to produce cold using as source solar energy it’s totally feasible, when analyzed from the thermodynamic point of view, and more the cycle presented on this article produce an amount satisfactory of cold, which is 227 W.

The research field of diffusion absorption is not too extensive, so during the elaboration of this paper it was evident that this area still needs improvement in order to enable a possible manufacture of this equipment on a large scale, since it uses only renewable energy and its gases do not impact the greenhouse effect at all. So, more studies are still necessary to improve the production and improve the coefficient of performance.

Thus, through the calculated results, it was found as generator heat inputted on the system the value of 691.9 Watts, producing a refrigeration rate of 227 W, and having as consequence a coefficient of performance of 0.33.

A proposition for future work in this area is the study of the influence of the collector’s size and the absorbent fluid on the coefficient of performance.

5. ACKNOWLEDGEMENTS

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