

Exergy Analysis of a Double-Effect Solar Absorption Refrigeration System in Ngaoundere

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Abstract

Solar energy is replacing more and more traditional sources of energy because of the fact that it's also fighting about global warming. This study is based on exergy analysis of a double-effect series flow absorption refrigeration system powered by solar energy in Ngaoundere. The simulation is done on the basis of a half hourly analysis for the first time, from 6.30 AM to 6.30 PM, using water-lithium bromide as working pair. The main parameters for the performance of an absorption cycle, which are the COP and the ECOP, have been analyzed and the results show that this two parameters increase while increasing the temperature of the main generator. The exergy loss of each component of the system and the total exergy loss of the system have been analyzed and their effectiveness calculated, using the first and second law of thermodynamics. The highest exergy loss occurs in the main generator GI and in the absorber, making these components more important in an absorption cycle. This analysis is based on a mathematical model using FORTRAN language. The results obtained may be useful for the optimization of solar absorption refrigeration systems.

Keywords

Double Effect, Refrigeration, Half Hourly, Exergy Loss

1. Introduction

Refrigeration is one of the active cooling methods studied here to enable nu-

merous villages from remote areas and why not towns, to benefit from welfare offered by nature; areas where electricity is in short supply are also concerned. Refresh air or refrigeration is becoming more and more important in the daily activities nowadays. The refrigerators are useful for making ice, for storing vaccines and food in areas where electricity is unavailable or high in cost that is the case in Africa in general and Cameroon in particular. Passive cooling systems, that is: good insulation, double glazing or use of thermal mass and ventilation are no longer sufficient. In inter-tropical regions, with the sun available all the yearlong, solar energy has a considerable interest because of the promising advantages. The First solar refrigeration system was introduced by Ferdinand Carré in 1895 [1]. Double-effect absorption refrigeration cycle was introduced for the first time in 1956 for developing the system performance within a heat source at high temperature [2].

In order to improve the performance of a solar absorption refrigeration system, number optimization studies based on the energy analysis have been done. Energy analysis takes into account the first law of thermodynamics which deals with the conversion of energy; this cannot show where the irreversibility occurs in a system that has many sources of energy [3].

The exergy analysis based on the second law of thermodynamics, is the only way to detect irreversibility in different components of the system that is the only rational basis for evaluating the system performance [4].

Many researchers have carry out studies on exergy analysis of double-effect refrigeration systems. K. Talukdar *et al.* [5] performed an exergy analysis to show the performance variation of both the topping PC and the bottoming ARS with changing HPG temperature from 120°C to 150°C. S.A.M Said *et al.* [6] brought two modifications to the traditional solar absorption refrigeration system by recovering waste heat from dephlegmator and utilization of a refrigerant storage unit. The COP increases 18% because of these modifications. The exergy variation of a solar assisted double-effect absorption refrigeration system, was studied in detail and the parameters of the cycle analyzed by Ravikumar T.S *et al.* [7] on the basis of first and second law effectiveness and the results indicated that a better quality of the evaporator, has more effect on the system performance than the better quality of other components. The study of Nahla Bouazis *et al.* [8] reveals that, the performance of a novel configuration is better than that of the two stage conventional configuration and it allows a lower temperature, about 60°C - 120°C instead of 100°C - 160°C for conventional cycle. T. Avanesian *et al.* [9] analysed and compared different water-cooled LiBr-H₂O absorption systems under different operating and climatic conditions, and the effect of considering the chemical exergy of the LiBr-H₂O solution on the second law analysis of such a system is probed. Milad Ashouri *et al.* [10] examined the performance of a double pressure organic Rankine cycle couple with a solar collector, via a thermal tank during a day. Results showed that the system is capable of generating stable power during the day with a solar fraction of 100%. Kai Zhang

et al. [11] discussed in detail the exergy loss for each component, in order to identify the inefficient components and then to indicate the way to improve the performance of a double-effect evaporating temperature chiller. Rabah Gomri [12] investigated the potential of application of a single effect and multiple effect absorption cooling systems for the production of chilled water. The simulation results were used to study the influence of the various operating parameters on the performance of single, double and triple effect absorption cooling cycles. Omer Kaynakli *et al.* [13] investigated the equation temperature of the overall system and its components by conducting a parametric study on heat capacity and exergy destruction of the HPG, coefficient of performance (COP) of the system and mass flow rate of heat sources. R. Maryami *et al.* [14] done a comparative study between the five classes of LiBr/water absorption refrigeration systems to investigate the influence of various operating parameters on the coefficient of performance. Zeyu Li *et al.* [15] give the yearly performance analysis of a solar air cooled double effect LiBr/H₂O absorption cooling system in subtropical city, including total efficiency, cooling capacity per area of collector and solar fraction. Ahmed Hamed *et al.* [16] optimized the coefficient of performance of a double-effect absorption system and predicted the coefficient of performance and exergetic efficiency of the system with two equations developed. Rabah Gomri *et al.* [17] simulated a double-effect vapor absorption cooler system to calculate exergy losses, the coefficient of performance and the number of exergy of each component. A. Allouhi *et al.* [1] presented a review of the available Technologies to provide cooling from solar energy for both thermal and photovoltaic ways. A market study is also carried out to evidence the potentiel of these solar cooling technologies with overview of the various solar cooling installations in Europe, Egypt and China was also presented. S.W Sharshir *et al.* [18] highlighted design methods to allow the researchers to optimize the solar still further development and proposed some recommendations.

In this study, in order to analyse the system with a great accuracy, we decided to look more closely by the simulation of a half hour analysis, from 6:30 AM to 6:30 PM of a double effect series flow absorption refrigeration system, using water-lithium bromide solution as working fluid with an Evacuated tube solar collector type. The data, which are solar insolation and environmental temperature are taken from M. Tenkeng *et al.* [19] and were collected on the 15th of January 2014, in Ngaoundere, Cameroon. The main focus of this study is concentrated on the exergy analysis of each component of the system, precisely the exergy loss [20]. The coefficient of performance (COP) and the exergetic coefficient of performance (ECOP) of the system are also investigated.

2. Description of the Model and Assumptions

2.1. Description of a Series Flow Solar Absorption Cycle

Double-effect consists of components similar as in single-effect absorption system with an additional generator, see **Figure 1**. The weak solution at state 1 is

pumped from the absorber to the first-effect solar assisted generator (GI) through two heat exchangers (HEX I and HEX II). The LiBr-H₂O solution in the first-effect generator is heated at high temperature. This high temperature heat comes from a solar collector to supplies to the first-effect generator to boil out the water vapour from the solution. The vapour refrigerant generated is condensed at high pressure in the second-effect generator. Thus, the primary vapour, from the first-effect generator goes to the secondary condenser incorporated in the second-effect generator (GII). The heat of condensation from this condenser is utilized by the second-effect generator. The strong solution, so called because it's poor in water, which is released by the first-effect generator at state 5 flows to the second-effect generator, through HEX II where the solution is cooled to some extent by exchanging heat with the weak solution. The heat rejected is used to produce addition refrigerant vapour from the solution coming from the first-effect generator. At the outlet of the second-effect generator vapour is produced, which together with the condensed water vapour from the secondary condenser, flows into the main condenser through a throttle valve where the heat of condensation is rejected to the atmosphere. Thus, the overall amount of liquid refrigerant leaving the main condenser is the sum of refrigerant coming from the first and second-effect generators. The refrigerant in a liquid phase from this condenser, on expansion, flows to the evaporator through a throttle valve (REV) where it is evaporated at low pressure. After extracting heat from the medium to be cooled, the refrigerant evaporates and then passes to the absorber where it gets absorbed by the strong solution coming from the second-effect generator, through HEX I and a throttle valve (SEV). The resulting weak solution in the absorber is then pumped to the first-effect generator and the cycle completes.

Figure 2 shows the P-T-X diagram of our chiller. Ideally, this system has three pressure levels and five operating temperature levels which are: T_s ; T_g ; T_c ; T_{gl} and T_{gll} . Low pressure prevails in the evaporator and absorber ($P_1 = P_a = P_e$ LPL), high pressure in the first-effect generator (GI) ($P_3 = P_{GI}$ HPL) and intermediate pressure in the condenser and the second-effect generator (GII) ($P_2 = P_c = P_{GII}$ IPL).

2.2. Assumptions

For the simulation of the cycle, following assumptions were made [21] [22] [23]:

- 1) The ambient temperature is assumed to be 298.15 K;
- 2) The refrigerant completely evaporates in the evaporator;
- 3) The thermal and pressure losses can be ignored;
- 4) The solution is saturated after is passes through the absorber and the generator;
- 5) The refrigerant vapor is completely condensed in the condenser;
- 6) For the temperature range considered, the absorbent is non-volatile;
- 7) The refrigerant leaves the condenser and evaporator at saturated state;

- 8) Changes occur only at pump and pressure-reducing valves for the pressure;
- 9) Because thermodynamic work required change in specific volume liquid, the solution pump work is neglected;
- 10) In the valves, the expansion process is isenthalpic.

3. Thermodynamic Analysis

3.1. Exergy Analysis of the System

The exergy of fluid stream can be defined as [24]:

$$\varphi = (h - h_0) - T_0 (s - s_0) \quad (1)$$

where φ is the exergy of the fluid at temperature T , h and s are respectively enthalpy and entropy of the fluid, h_0 and s_0 are respectively the enthalpy and entropy of the fluid at environmental temperature T_0 (298.15 K).

The specific exergy content of a mixture with m components is defined by A. Sözen *et al.* 2007 [25] as:

$$\varphi = \sum_{n=1}^m x_n h_n - T_0 \sum_{n=1}^m x_n s_n - \sum_{n=1}^m x_n \mu_{n0} \quad (2)$$

where T_0 is the reference temperature (25°C), x_n is mass fraction and μ_{n0} the chemical potential of the n^{th} component of the mixture at T_0 and P_0 .

The exergy loss or the availability loss in each component is given by [21]:

$$\Delta\varphi = \sum m_i \varphi_i - \sum m_o \varphi_o - \left[\sum Q \left(1 - \frac{T_0}{T} \right)_i - \sum Q \left(1 - \frac{T_0}{T} \right)_{ref} \right] + \sum W \quad (3)$$

where $\Delta\varphi$ is the lost exergy or irreversibility that occurs in the process. The first two terms of the right hand side are the exergy of the inlet and outlet streams of the control volume. The third and fourth terms are the exergy associated with the heat transferred from the source maintained at a temperature T . The last term is the exergy of mechanical work added to the control volume. The term is negligible for absorption systems as the solution pump has very low power requirements.

The total exergy loss of absorption system is the sum of exergy loss in each component and is written as:

$$\Delta\varphi = \sum_{n=1}^m \Delta\varphi_n \quad (4)$$

De Vos (1992) [26] established equations for calculating the exergy loss to the collector per unit area as function of the exergy emitted from the sun minus the albedo of the earth and the radiation emitted from the solar collector. The exergy analysis of the collector can be obtained with Pridasa was *et al.* 2004 [27]:

Exergy (radiation) input:

$$\varphi = f \sigma T_{sun}^4 + (1 - f) \sigma T_p^4 - \sigma T_{sc}^4 \quad (5)$$

where sun is for sun, p for planet and sc for solar collector, f is the sunlight dilution factor equal to 2.16×10^{-5} on earth. The solar radiation that reaches the

solar collector is transformed into heat. This heat is partly absorbed by thermal fluid and the surrounding equipment and partly lost to the environment. The available solar radiation is transformed into available heat for the process, however, the second law of thermodynamics hinders the transformation of all heat into exergy [26] [27]. The exergy of the solar heat input to the solar collector is given by the relationship:

Exergy (heat) input:

$$\varphi_{s,h} = Q_{ava} \left(1 - \frac{T_{ref}}{T_{sc}} \right) \quad (6)$$

Exergy loss during the transformation process for each component is as follows:

$$I_{sc,r} = \varphi_s - \varphi_{s,h} \quad (7)$$

The useful exergy gained by the solar collector is:

$$\varphi_{su} = Q_u \left(1 - \frac{T_{ref}}{T_{sc}} \right) \quad (8)$$

where Q_u is the useful steady state energy gain to the solar collector. The exergy loss from the input of solar collector to the working fluid can be calculated using the equation:

$$\Delta\varphi_{sc} = \varphi_s - \varphi_{su} - \Delta\varphi_{en} - \Delta\varphi_{sc,r} \quad (9)$$

where $\Delta\varphi_{sc,r}$ is the exergy lost during the transformation from solar energy radiation to heat on the solar collector and written as [26]:

$$\Delta\varphi_{sc,r} = \varphi_s - \varphi_{s,h} \quad (10)$$

And $\Delta\varphi_{en}$ is the average loss caused by the wind effect on the solar collector. It's impossible to calculate a priori, since the speed, the direction and the sense of the wind are unknown [27]. Moreover, as the plate temperature is a function of $\Delta\varphi_{en}$, $\Delta\varphi_{en}$ cannot be determined either. For these reasons, it will be assume for simplicity $\Delta\varphi_{en} = 0$ and therefore:

$$\Delta\varphi_{sc} = \varphi_s - \varphi_{su} - \Delta\varphi_{sc,r} \quad (11)$$

The exergy efficiency for cooling is the ratio of the chilled water exergy at the evaporator to the exergy of the heat source at the generator:

$$\varphi_{cooling} = \frac{m_{22}(\varphi_{22} - \varphi_{23})}{m_{18}(\varphi_{18} - \varphi_{19})} \quad (12)$$

For the thermodynamic analysis, we have equations adapted from O. Kaynakli *et al.* (2007) [28]:

The mass balance:

$$\dot{m}_w = \dot{m}_s + \dot{m}_{H_2O} \quad (13)$$

$$\dot{m}_w X_w = \dot{m}_s X_s \quad (14)$$

where X is the LiBr concentration, w (weak solution) and s (strong solution).

The flow rate of the strong and weak solutions can be determined from equation below:

$$\dot{m}_s = \frac{X_w}{X_s - X_w} \dot{m}_{H_2O} \quad (15)$$

$$\dot{m}_w = \frac{X_s}{X_s - X_w} \dot{m}_{H_2O} \quad (16)$$

The double circulation ratios have been defined in the cycle as:

$$f_1 = \frac{X_w}{X_{s1} - X_w} = \frac{X_1}{X_5 - X_1} \quad \text{and} \quad f_2 = \frac{X_{s1}}{X_{s2} - X_{s1}} = \frac{X_5}{X_{14} - X_5} \quad (17)$$

Using circulation ratios, the heat capacity of each component is:

For the generators

$$Q_{GI} = h_{10} + f_1 h_5 - (f_1 + 1) h_{17} \quad (18)$$

$$Q_{GII} = (f_2 (f_1 + 1)) h_{13} - f_2 h_9 + f_1 (f_2 / (f_1 + 1)) h_{14} \quad (19)$$

For the condenser

$$Q_C = (f_2 / (f_1 + 1)) h_{13} - (f_2 / (f_1 + 1) + 1) h_{15} \quad (20)$$

For the evaporator

$$Q_E = ((f_2 / (f_1 + 1)) + 1) (h_{17} - h_{16}) \quad (21)$$

For the absorber

$$Q_A = (f_2 / (f_1 + 1)) h_{17} + ((f_2 f_1) / (f_1 + 1)) h_7 - (f_2 + 1) h_1 \quad (22)$$

For the heat exchangers

$$Q_{HEXI} = (f_2 + 1) (h_3 - h_2) \quad (23)$$

$$Q_{HEXII} = (f_2 + 1) (h_4 - h_3) \quad (24)$$

The pump work can be expressed as:

$$W_p = v_{ws} (P_{GII} - P_E) (f_2 + 1) / \eta_p \quad (25)$$

The coefficient of performance for the system is given by:

For the exergy destruction, the following equations are determined as:

For the generators

$$I_{GI} = \dot{m}_{10} (\varphi_{18} (f_2 + 1) - \varphi_5 f_2 - \varphi_{10}) + \dot{m}_{18} (\varphi_{18} - \varphi_{19}) \quad (26)$$

$$I_{GII} = \dot{m}_{10} (\varphi_9 f_2 - \varphi_{13} (f_2 / (f_1 + 1)) - \varphi_{14} (f_1 (f_2 / (f_1 + 1)))) + \dot{m}_{10} (\varphi_{10} - \varphi_{11}) \quad (27)$$

For the condenser

$$I_C = \dot{m}_{10} (\varphi_{13} (f_2 / (f_1 + 1)) + \varphi_{12} - \varphi_{15} (f_2 / (f_1 + 1))) + \dot{m}_{20} (\varphi_{20} - \varphi_{21}) \quad (28)$$

For the evaporator

$$I_E = \dot{m}_{10} ((f_2 / (f_1 + 1)) + 1) (\varphi_{16} - \varphi_{17}) + \dot{m}_{22} (\varphi_{22} - \varphi_{23}) \quad (29)$$

For the absorber

$$I_A = \dot{m}_{10} \left((\varphi_{17} (f_2 / (f_1 + 1)) + 1) + \varphi_7 ((f_2 f_1) / (f_1 + 1)) - \varphi_1 (f_2 + 1) \right) + \dot{m}_{24} (\varphi_{24} - \varphi_{25}) \quad (30)$$

For the throttle valves

$$I_{REV} = \dot{m}_{10} \left((f_2 / (f_1 + 1)) + 1 \right) (\varphi_{15} - \varphi_{16}) \quad (31)$$

$$I_{SEV} = \dot{m}_{10} f_2 (\varphi_6 - \varphi_7) \quad (32)$$

For the heat exchangers

$$I_{HEXI} = \dot{m}_{10} \left((f_1 + 1) \right) (\varphi_2 - \varphi_3) + (f_1 f_2) / (f_2 + 1) (\varphi_6 - \varphi_{14}) \quad (33)$$

$$I_{HEXII} = \dot{m}_{10} \left(f_2 (\varphi_5 - \varphi_8) + (f_2 + 1) (\varphi_3 - \varphi_4) \right) \quad (34)$$

For the pump

$$I_{PUMP} = \dot{m}_{10} (f_1 + 1) (\varphi_1 - \varphi_2) \quad (35)$$

The total exergy destruction for all the system is:

$$I_S = I_{GI} + I_{GII} + I_C + I_A + I_E + I_{REV} + I_{SEV} + I_{HEXI} + I_{HEXII} + I_{PUMP} \quad (36)$$

A. Sözen *et al.* 2007 [27] defined the exergetic coefficient of performance on which depends the effectiveness of an ARS:

$$ECOP = \frac{\dot{q}_e \left(1 - \frac{T_0}{T_e} \right)}{\dot{q}_g \left(1 - \frac{T_0}{T_g} \right) + \dot{W}_{pe}} = \frac{m_{17} (\varphi_{17} - \varphi_{16})}{m_4 \varphi_4 + m_{10} \varphi_{10} - m_5 \varphi_5} \quad (37)$$

And the coefficient of performance of the system for cooling purpose is:

$$COP = \frac{\dot{q}_e}{\dot{q}_g} = \frac{m_{17} (h_{17} - h_{16})}{m_4 h_4 + m_{10} h_{10} - m_5 h_5} \quad (38)$$

3.2. Input Parameters for the Simulation

To simulate the above equations, a simulation program written in FORTRAN was developed. Properties of the LiBr-H₂O solution and typical overall heat transfer coefficient were adopted from Gur Mittelman 2007 [29]. Also, the solar insolation and the ambient temperature, the solution mass flow rate and the solution concentration are given in **Table 1**.

The cooling load of the system is $Q_E = 10$ kw and the evaporator temperature is $T_E = 6^\circ\text{C}$, the reference temperature is $T_0 = 25^\circ\text{C}$. The temperature of the cooling water in the absorber and the condenser is 3°C . The solar collector is a flat plate collector with 60° angle of incidence and 100 m^2 of area. Properties of the LiBr-H₂O solution and typical overall heat transfer coefficient were adopted from Gur Mittelman [29]. For the heat transfer area or total conductance, we adopted the values from M. Tenkeng *et al.* [19]: (UA)_a = 1.800 kW/K, (UA)_e = 2.250 kW/K, (UA)_g = 1.000 kW/K, (UA)_c = 1.200 kW/K.

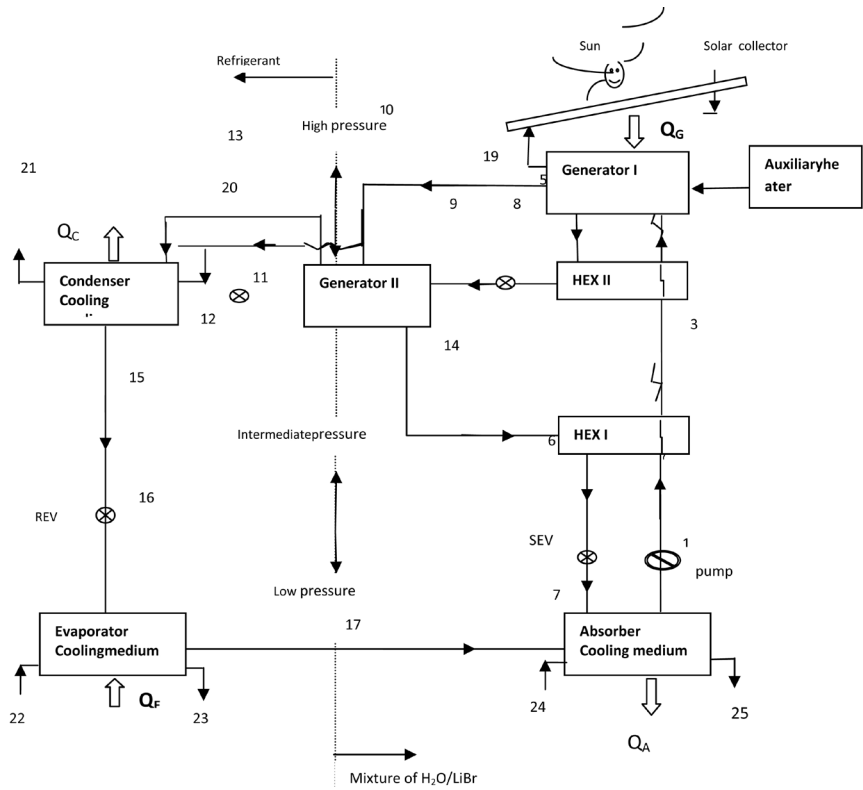


Figure 1. Double effect series flow solar absorption refrigeration chiller.

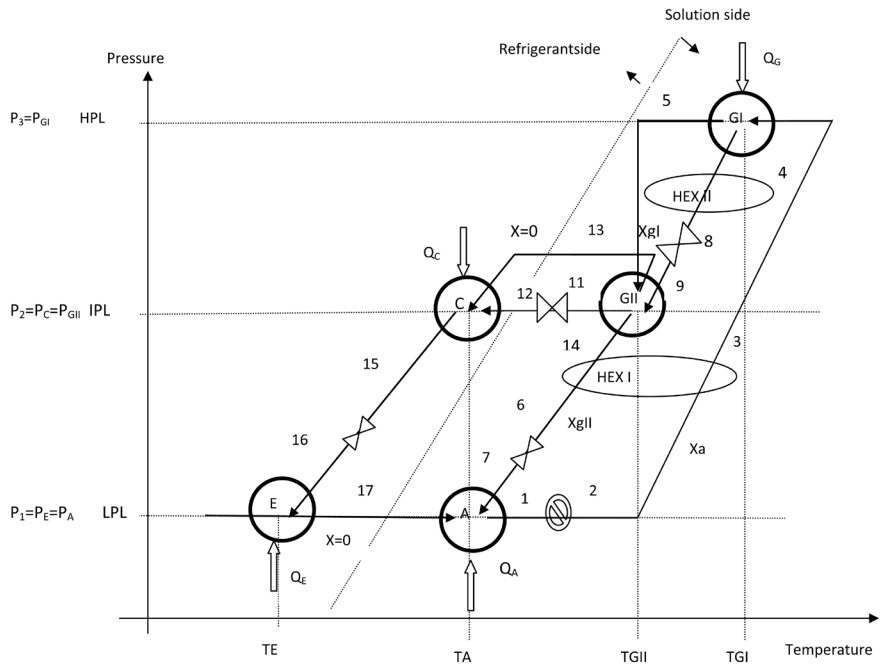


Figure 2. P-T-X diagram of double effect series flow absorption refrigeration cycle.

4. Results and Discussion

To have a better understanding of the results and their interpretations, thermodynamics properties of each point of the entire system are given in **Tables 1-4**.

Table 1. Ambient temperature and Solar insolation in Ngaoundere on the 15th of January 2014 [19].

Temperature	Solar insolation (W/m ²)	Hour
25.4125	29.126	6.30
26.875	153.846	7.00
28.184	218.446	7.30
28.375	326.923	8.00
29.737	422.330	8.30
29.875	528.846	9.00
31.097	635.922	9.30
31.3	750	10.00
31.75	815.534	10.30
32.3125	865.385	11.00
32.6875	907.766	11.30
33.0625	932.6923	12.00
33.25	946.602	12.30
33.4375	951.923	13.00
33.625	936.893	13.30
33.4375	875	14.00
33.25	830.097	14.30
32.875	721.154	15.00
32.6875	616.504	15.30
32.50	509.615	16.00
32.3125	393.204	16.30
32.125	298.077	17.00
31.75	174.175	17.30
31.5625	134.615	18.00
31.0	53.398	18.30

Table 2. The four level temperature at 12.30 PM.

Component	Temperature (°C)
Evaporator T_E	6
Condenser T_C	35
Absorber T_A	35
Second effect generator T_{GII}	81.45
First effect generator T_{GI}	137.10

Table 1 gives the solar insolation and the ambient temperature collected in the Meteorological Station of Ngaoundere on the 15th of January 2014 by M. Tenkeng

Table 3. Thermodynamic properties of the cycle at 12.30 PM.

i	$T(i)$ °C	$h(i)$ (kJ/kg)	$s(i)$ (KJ/kg·K)	$m(i)$ (Kg/s)	$X(i)$ (%) LiBr	$\Psi(i)$ (kJ/kg)
1	35.00	81.45	0.4625	0.1836	52.0	48.271
2	35.00	81.36	0.2630	0.1836	52.0	48.271
3	64.00	138.47	0.3869	0.1836	52.0	51.463
4	115.20	135.41	0.5012	0.1836	52.0	77.120
5	137.10	240.23	0.5124	0.1591	55.0	111.324
6	46.50	132.14	0.2658	0.1578	60.0	88.251
7	46.50	132.14	0.2658	0.1578	60.0	88.251
8	80.35	191.42	0.4857	0.1591	55.0	84.120
9	80.35	191.42	0.4857	0.1591	55.0	84.120
10	137.10	2712.58	6.8636	0.0245	00.0	19.102
11	89.65	365.95	1.2011	0.0245	00.0	0.965
12	35.00	365.95	1.2412	0.0245	00.0	0.183
13	81.45	2638.41	8.5834	0.0133	00.0	3.220
14	81.45	192.17	0.4658	0.1458	60.0	101.241
15	35.00	145.25	0.5124	0.0378	00.0	0.122
16	6.00	145.25	0.5326	0.0378	00.0	-0.598
17	6.00	2612.4	8.9236	0.0378	00.0	-16.330
18	147.00	167.22	5.8768	0.5500	-	98.210
19	141.00	163.45	5.6789	0.5500	-	85.118
20	25.00	103.79	0.3586	0.2500	-	-7.140
21	31.15	124.28	0.3978	0.2500	-	-4.222
22	16.00	60.26	0.1879	0.4000	-	24.145
23	9.00	39.89	0.1145	0.4000	-	30.811
24	25.00	103.79	0.3586	0.2800	-	1.398
25	31.15	124.28	0.4127	0.2800	-	4.976

Table 4. The effectiveness for each component at 12.30 PM.

Component	Effectiveness
Generator I	0.37
Generator II	0.33
Condenser	0.68
Evaporator	0.74
Absorber	0.78

et al. [19]. **Table 2** presents the temperature of the components of the system at 12.30 PM. **Table 3** shows the thermodynamic properties of each of the points within the cycle, enthalpy, entropy and exergy at 12:30 PM can be slightly compared to that obtained by Omer Kaynakli *et al.* [13] for the different points of the system. The values of the exergy for the two studies are similar. **Table 4** shows the effectiveness of the two generators, the absorber, the evaporator and the

condenser. It is noted that the first effect generator GI is a component with a low effectiveness. This indicates that the generator needs a particular attention and must be optimized. **Figure 1** shows the variation of insolation during the day chosen for our study. This curve has the same behavior with that of the ambient temperature shown in **Figure 3**. The variation of the exergy loss of the whole system with the temperature of the first effect generator GI is presented in **Figure 4**. While the temperature of GI increases, the exergy loss decreases. The lowest exergy loss is obtained for a generator temperature of 150°C, the maximum temperature for the second effect generator powered by solar energy [13]. **Figure 5** shows the variation of the ambient temperature with the maximum value at 12.30 PM. The temperature of the solar collector has the same behavior. **Figure 6** and **Figure 7** show the variation of the COP and ECOP respectively with the temperature of the first effect generator GI. The COP and ECOP depend on the generator load Q_{ge} and the evaporator load Q_{ev} . The COP decreases almost nonlinearly while the temperature of GI increases. This is a general

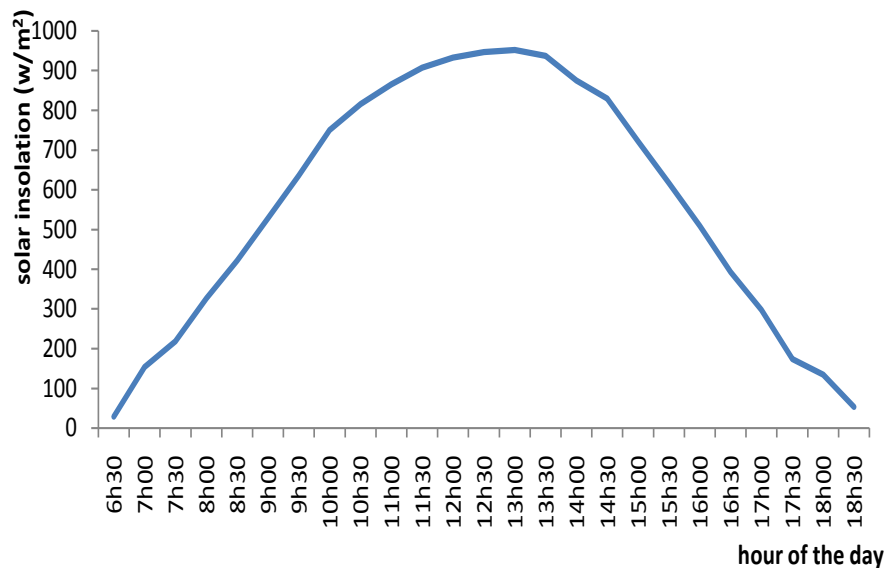


Figure 3. Variation of solar insolation during the day.

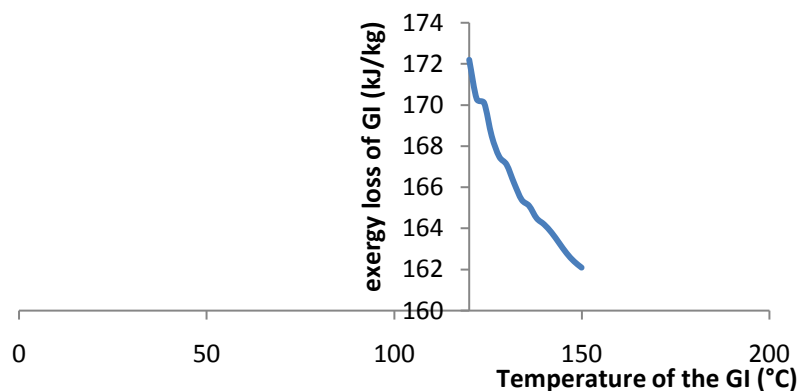


Figure 4. Variation of the exergy loss of GI with the temperature of GI.

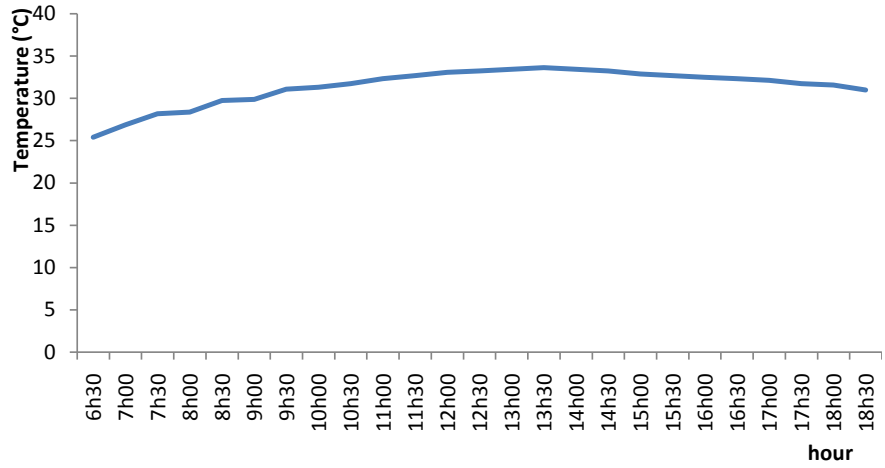


Figure 5. Variation of the ambient temperature during the day.

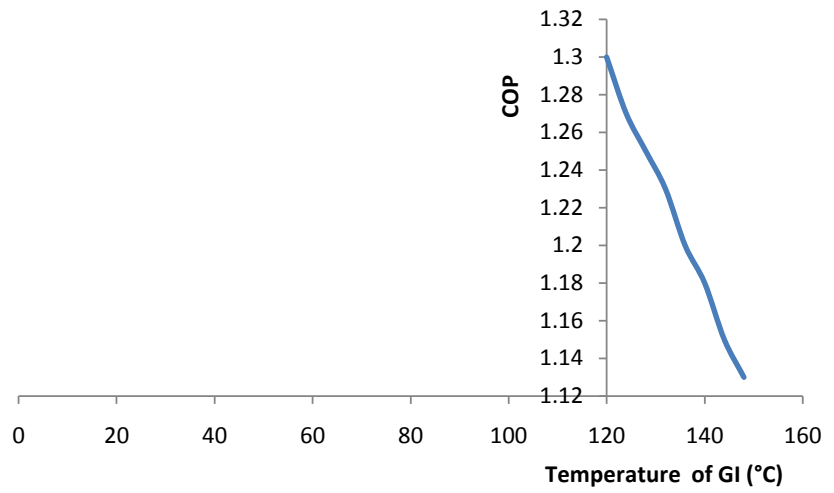


Figure 6. Variation of the COP with the temperature of GI.

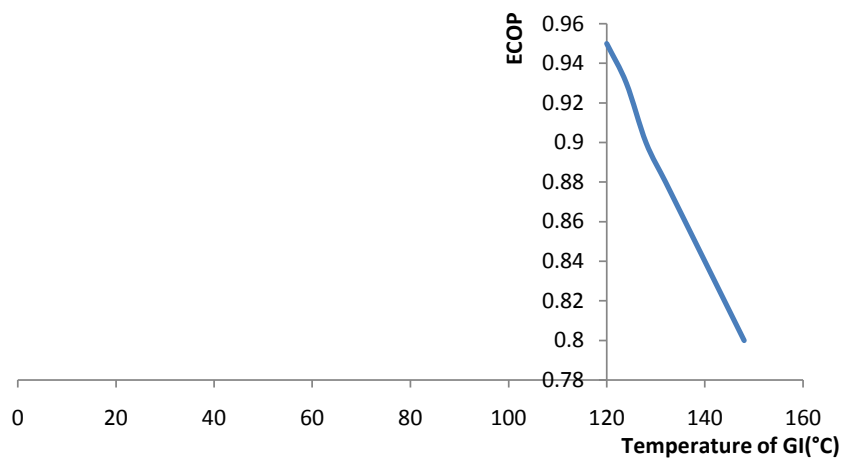


Figure 7. Variation of ECOP with the GI temperature.

behavior of Absorption Refrigeration Systems between the COP and the temperature of the generator receiving the main energy (GI). The ECOP is a very

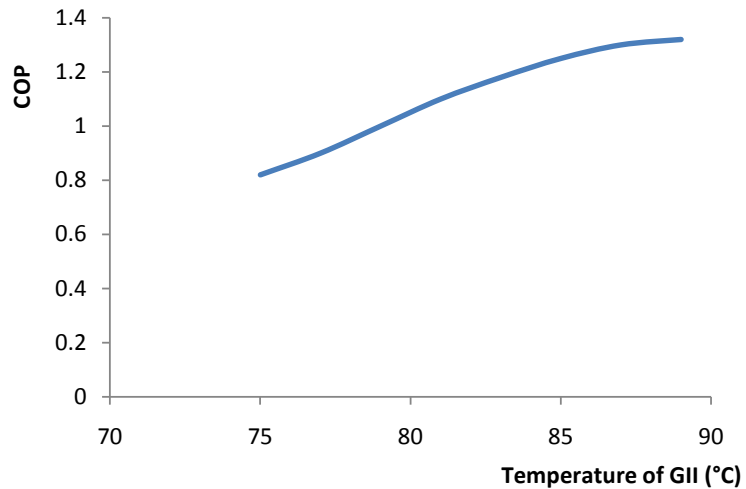


Figure 8. Variation of the COP with the temperature of GII.

important tool for the evaluation of the performance of a system while applying the second law of thermodynamics. The ECOP decrease when the temperature of the first effect generator increase with a minimum value of ECOP which is 0.8, result in accordance with the litterature. **Figure 8** shows the variation of the COP with the temperature of GII, the COP increases when the temperature increases. This result is in adequation with the need of the second generator in the double effect series flow absorption refrigeration system. So, the main impact of the second generator is to increase the value of the COP of the system.

5. Conclusions

By applying the first and the second laws of thermodynamics, the exergy analysis of a double-effect series flow solar absorption refrigeration system, with the most commun working pair, that is LiBr-H₂O, has been performed under weather conditions of Ngaoundere, in Cameroon, of the 15th January 2014. The study was made throughout the middle of the dry season. The exergy loss of the absorption cooling device and half hourly exergy destruction values of components were determined. The COP and the ECOP of the system were also investigated. The main conclusions obtained from the present study are as follows:

- 1) The maximum exergy destruction occuring in the solar energy assistance system is in the generator and the evaporator.
- 2) The exergy loss in the generator indicated that the generator is the component that needs to optimise.
- 3) The COP and ECOP variation with the temperature of the generators brings us to the conclusion that the second generator has a great contribution to the double effect refrigeration system. The system can therefore be implemented in Cameroon and in countries with high solar availability.

Our study can serve as preliminary investigation to solar water with cooled double effect absorption refrigeration system in tropical cities, and it is necessary to provide a foundation for further researches.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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Nomenclature

COP: coefficient of performance
ECOP: exergy coefficient of performance
f: circulation ratio
h: enthalpy (kJ/kg)
s: entropy (kJ/kgK)
m: mass flow rate (kg/s)
Q: heat flow rate (kW)
T: temperature (°C or K)
P: pressure (kPa)
ψ: exergy (kJ/kg)
x: mass fraction of lithium bromide (%)

Subscripts

i: component i
ref: reference
g: ge: generator
e: ev: evaporator
a: ab: absorber
c: co: condenser
hex: heat exchanger
ev: expansion valve
pump: p: pump
p: planet
ws: w: weak solution
sc: solar collector
s: solution or system
sc, r: solar collector radiation
h: heat
p: planet
HEX I: heat exchanger
HEX II: heat exchanger
SEV: solution expansion valve
S1: weak solution at the inlet of GI
S2: weak solution at the inlet of GII
W: work
REV: refrigerant expansion valve
u: useful
ava: available.
sun: sun