Journal of Power and Energy Engineering, 2017, 5, 1-18
http://www.scirp.org/journal/jpee
ISSN Online: 2327-5901
ISSN Print: 2327-588X

Exergy Analysis of a Solar Absorption Refrigeration System in Ngaoundere

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How to cite this paper: Tenkeng, M., Wouagfack, P.A.N., Lissouck, D. and Tchinda, R. (2017) Exergy Analysis of a Solar Absorption Refrigeration System in Ngaoundere. *Journal of Power and Energy Engineering*, **5**, 1-18.
https://doi.org/10.4236/jpee.2017.510001

Received: August 24, 2017
Accepted: October 22, 2017
Published: October 25, 2017

Abstract

In this study, the first and second laws of thermodynamics are used to analyze the performance of a single-stage absorption refrigeration system powered by solar energy. The working pair used in this study is LiBr-H_2O where water (H_2O) is the refrigerant and the lithium bromide (LiBr) is the absorbent. A mathematical model based on exergy analysis is applied to analyse the system performance. Temperature, enthalpy, entropy, mass flow rate and exergy loss of each component including evacuated tube solar collector are evaluated. Furthermore, the overall coefficient of performance ($COP_{cooling}$) and the overall exergetic coefficient of performance ($ECOP_{cooling}$) of the solar absorption system (absorption system coupled to an evacuated tube solar collector) for cooling purpose are calculated from the thermodynamic properties of the working fluids under weather conditions of Ngaoundere city, Cameroon. The calculations were done on the basis of a half hourly analysis from 6:30 AM to 6:30 PM. The results were compared and they show that the exergy destruction highly occurs in the generator and the solar collector. The simulation results can be used for the thermodynamics optimization of solar absorption refrigeration systems.

Keywords

Refrigeration, Absorption, Exergy Analysis, Solar Collector

1. Introduction

The need of refresh air or refrigeration is becoming more and more important in the daily activities nowadays. The refrigerators are useful for making ice and 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. Refrigeration which is one of the active cooling methods is studied here to enable numerous villages from remote areas to benefit from welfare offered by nature; areas where electricity is in short supply are also concerned. In Africa, people have the opportunity to contribute to the protection of the environment by fighting against global warming. This can be done through the use of renewable energy sources such as solar, wind, geothermal, wasted heat and so on. Most of African countries are exposed to the sun all over the year, even during rainy seasons. Solar energy can therefore be a great opportunity for people living in this continent. Among the solar refrigeration technologies, the solar thermal with single-effect absorption system with the mature technology come into view to be the best option [1].

In order to improve the performance of a solar absorption refrigeration system, many 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 [2]. The exergy analysis based on the second law of thermodynamics, is the only way to detect irreversibility in different components of the system [3].

In Ravikumar *et al.* [4] study, exergy analysis of a double-effect solar assisted absorption system is carried out and influence of generator I, generator II temperatures on exergy values is shown. In A. A. Hasan *et al.* [5] paper, a second law efficiency is defined relative to a reversible cycle and maximized in order to find the optimum operating conditions of the cycle. The cycle performance is investigated over a heat source temperature range of 330 K - 470 K. Ghaddar *et al.* [6] have carried out research into solar absorption system performance in Beirut. Ezzine *et al.* [7] conducted a study on solar systems assisted with the double effect absorption refrigeration. The irreversibility of each component in the chiller was quantified and the potential of each component to contribute to the overall system's energy efficiency was determined. A. Fellah *et al.* [8] studied the performance of a Driven Solar Absorption Refrigeration System and submitted the cycle, under different operating and design conditions, to analyze the optimum conditions for which the maximum refrigeration effect can be achieved. M. Talbi and B. Agnew [9] performed the exergy analysis of an absorption refrigerator using lithium bromide and water as the working fluids. Recently C. Onan *et al.* [10] carried out studies on the hourly exergy destruction for each component in solar assisted absorption cooling system. Heng-Yi Li *et al.* [11] evaluated exergy losses in each component of a New Small Concentrating Solar Power Plant in China and they have seen that the system could run at full capacity all day long. Jieting Wei *et al.* [12] experimented in 2013 operating characteristics of a Solar-Assisted Heating System in Changchun and they concluded that it was feasible and also have a certain effect. G.C. Tubreoumya, *et al.* [13] developed a mathematical model representing the evolution of heat and mass transfer at each component of a solar adsorption refrigerator in Burkina Faso. Their work focused on the aim at the Solar Assisted Absorption Refrigeration Systems (SAARS) and various environment conditions, exergy loss of each component was calculated separately. Also calculations have been done twice with the dead state temperature equal to 25˚C as environmental temperature with a more realistic approach.

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 single stage absorption refrigeration system, using water-lithium bromide solution as working fluid with an Evacuated tube solar collector type. On the 15th of January 2014, we went to Ngaoundere, Cameroon, to collect data, which are solar insolation and environmental temperature. The main focus of this study is concentrated on the exergy analysis of each component of the system, precisely the exergy loss [14]. The coefficient of performance (*COP*) and the exergetic coefficient of performance (*ECOP*) of the system are also investigated.

2. Description of the Model

2.1. Solar Absorption Refrigeration System Modeling

The model of absorption system studied here is a single-effect or single-stage absorption refrigeration machine as shown in **Figure 1**. The system includes heat exchangers, a pump, valves and piping. Absorption systems are basically used to avoid the compression work. The working pair used here is LiBr-H_2O, where LiBr is the absorbent and H_2O is the refrigerant. This working pair offers good thermodynamic performance and is environmentally benign [15]. LiBr-H_2O absorption systems are the more suitable for solar application [16]. The environmental temperature is 25˚C. For the night, there is an auxiliary heater. The type of collector used is an evacuated one with selective surface and a total area of 10 m^2. The work of the pump is neglected; there is no spill over at the evaporator. Solar insolation values and ambient temperatures for Ngaoundere city, Cameroonare are shown in **Table 1**. These values are taken from Ngaoundere Meteorological Service. Ambient temperature during cool season is maximum at 33.62˚C, and minimum at 25.41˚C. The angle of incidence of the collector is taken as 60˚C. Cool water, Chilled water and hot water flow rates are taken as 0.00474, 0.00474 and 0.05691 kg/s respectively in the absorber, the condenser and the evaporator.

When the refrigerant vapour is coming from the evaporator, it's absorbed in a liquid strong solution. This liquid weak solution is pumped to higher pressure, where the refrigerant is boiled out of the solution by the additional heat, which is collected in solar system collectors and/or an auxiliary heating boiler. The refrigerant vapour is condensed by rejecting heat in the condenser and the pressure of saturated liquid refrigerant is reduced through an expansion valve. Heat transfer from the cooling space causes vaporization of the refrigerant at low pressure, and then flows to the absorber. The liquid strong solution returns to the absorber through a throttling valve whose purpose is to provide a pressure drop to

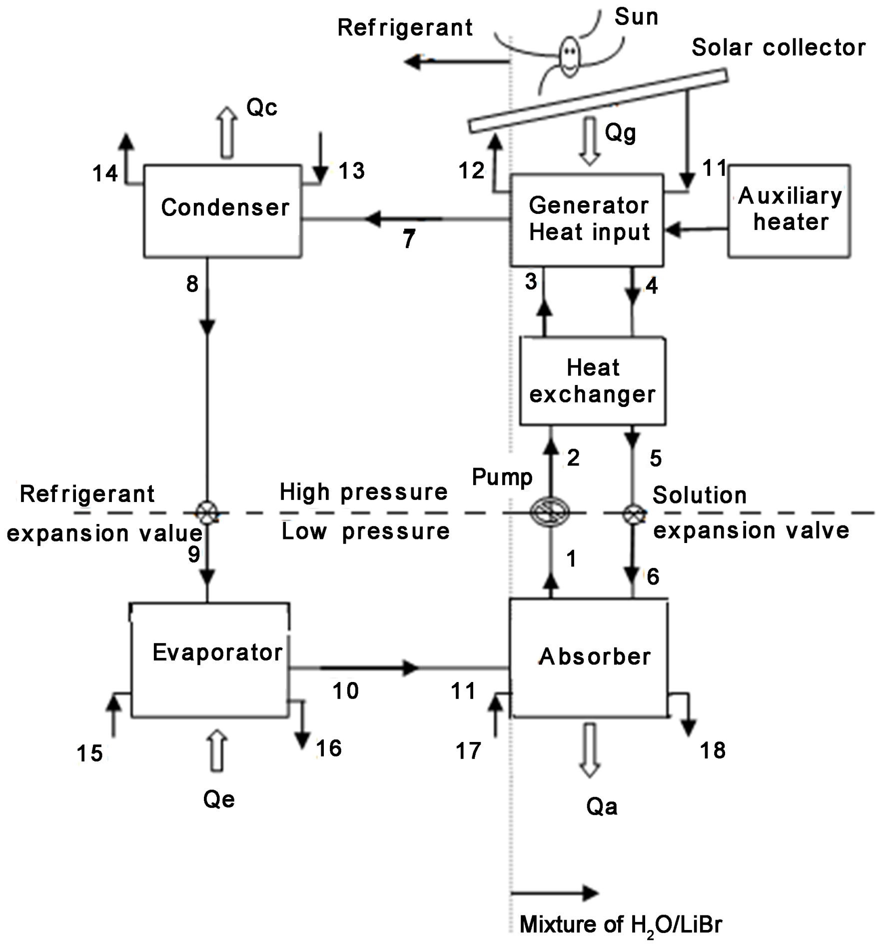

Figure 1. Single effect solar absorption refrigeration system.

maintain the difference between the generator and absorber. Heat recovery between the weak and strong solution is achieved through a solution heat exchanger and a constant flow rate of weak solution is maintained by a pump with negligible energy consumption [17].

Absorption is the process of attracting and holding moisture by substances called desiccants. Desiccants are sorbent materials that have an ability to attract and hold other gases or liquids and have a particular affinity for the refrigerant. During absorption the desiccant undergoes a chemical change as it takes in the moisture [18].

The basic idea of an absorption system is to avoid the compression work; this is done by using a suitable working pair [19]. The absorption process consist of transfer of material from one phase to another, interpenetrates the second phase to form solution [20].

2.2. Refrigerant

One of the most important elements of any refrigeration system is the refrigerant, since the working pair conditions and compatibility with the environment

Table 1. Solar insolation and ambient temperature for Ngaoundere, on the 15th of January 2014.

Time	Solar insolation ($kJ \cdot m^{-2}$)	Temperature (˚C)
6:30	29.126	25.413
7:00	153.846	26.875
7:30	218.446	28.184
8:00	326.923	28.375
8:30	422.330	29.737
9:00	528.846	29.875
9:30	635.922	31.097
10:00	750.000	31.300
10:30	815.534	31.750
11:00	865.385	32.313
11:30	907.766	32.687
12:00	932.692	33.062
12:30	946.602	33.250
13:00	951.923	33.437
13:30	936.693	33.625
14:00	875.000	33.437
14:30	830.097	33.250
15:00	721.154	32.875
15:30	616.504	32.687
16:00	509.615	32.500
16:30	393.204	32.312
17:00	298.077	32.125
17:30	174.175	31.750
18:00	134.615	31.562
18:30	53.398	31.000

principally depend on it. A refrigerant needs two requirements: high latent heat per unit volume and good thermal stability. The thermodynamic properties for $LiBr/H_2O$ are pressure, temperature, concentration, enthalpy, entropy and density. These properties are interdependent and are necessary for computer simulation of absorption refrigeration systems. They are given by [9] and [21] [22] [23] [24].

3. Thermodynamic Analysis

The exergy of fluid stream can be defined by Arzu Şencan *et al.* [25]:

$$\varphi = (h - h_0) - T_0 (s - s_0) \tag{1}$$

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.* [26] as:

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

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 Arzu Şencan *et al.*:

$$\Delta\varphi=\sum m_i\varphi_i-\sum m_0\varphi_0-\left[\sum Q\left(1-\frac{T_0}{T}\right)_I-\sum Q\left(1-\frac{T_0}{T}\right)_{ref}\right]+\sum W \tag{3}$$

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:

$$\Delta\varphi=\sum_{n=1}^{m}\Delta\varphi_n \tag{4}$$

De Vos (1992) [27] 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 Pridasawas *et al.* 2004 [28]:

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

where subscripts *sun, p* and *sc* stand for *sun*, planet and solar collector respectively, 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. The exergy of the solar heat input to the solar collector is given by the relationship:

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

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

$$I_{sc,r} = \varphi_s - \varphi_{s,h} \tag{7}$$

The useful exergy gained by the solar collector is:

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

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:

$$\Delta\varphi_{sc} = \varphi_s - \varphi_{su} - \Delta\varphi_{en} - \Delta\varphi_{sc,r} \tag{9}$$

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:

$$\Delta\varphi_{sc,r} = \varphi_s - \varphi_{s,h} \tag{10}$$

$\Delta\varphi_{en}$ 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 [29]. 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 assumed for simplicity $\Delta\varphi_{en} = 0$ and therefore:

$$\Delta\varphi_{sc} = \varphi_s - \varphi_{su} - \Delta\varphi_{sc,r} \tag{11}$$

The exergy loss for the solar assisted absorption refrigeration system is given by:

$$\Delta\varphi_{system} = \Delta\varphi_T + \Delta\varphi_{SC} \tag{12}$$

The following are the performance equations for each of the components considering the second law of thermodynamics (exergy balance).

Generator

Exergy available: $$\varphi_{gen} = Q_g \left(1 - \frac{T_{ref}}{T_g}\right) \tag{13}$$

Exergy loss: $$I_{gen} = T_{ref}\left(m_g\left(s_3 - s_1\right) + m_{sc}\left(s_{g-sc-out} - s_{g-sc-in}\right)\right) \tag{14}$$

Ejector

Exergy loss: $$I_j = T_{ref}\left[\left(m_e + m_g\right)S_6 - m_g S_3 - m_e S_9\right] \tag{15}$$

Condenser

Exergy loss: $$I_c = T_{ref}\left(\left(m_g + m_c\right)\left(S_7 - S_6\right) + \frac{Q_c}{T_{ref}}\right) \tag{16}$$

Pump

Exergy loss: $$I_p = W_{pump} + m_g\left[\left(h_1 - h_7\right) - T_{ref}\left(S_1 - S_7\right)\right] \tag{17}$$

Expansion device

Exergy loss: $$I_{\exp} = m_e\left[T_{ref}\left(S_8 - S_7\right)\right] \tag{18}$$

Evaporator

Exergy delivered: $\varphi_e = Q_e\left(1-\frac{T_{ref}}{T_{room}}\right)$ (19)

Exergy loss: $I_e = T_{ref}\left[m_e\left(S_9 - S_8\right) - \frac{Q_e}{T_{room}}\right]$ (20)

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:

$$\varphi_{cooling} = \frac{m_{17}\left(\varphi_{17}-\varphi_{18}\right)}{m_{11}\left(\varphi_{11}-\varphi_{12}\right)} \tag{21}$$

The exergetic efficiency for heating is the ratio of the combined supply of hot water exergy at the absorber and condenser to exergy of heat source at the generator:

$$\varphi_{heating} = \frac{m_{15}\left(\varphi_{16}-\varphi_{15}\right)+m_{13}\left(\varphi_{14}-\varphi_{13}\right)}{m_{11}\left(\varphi_{11}-\varphi_{12}\right)} \tag{22}$$

The second law efficiency of the absorption system is measured by the exergetic efficiency. A. Sözen *et al.* 2007 defined the exergetic coefficient of performance of an ARS:

$$ECOP_{cooling} = \frac{\dot{q}_e\left(1-\frac{T_0}{T_e}\right)}{\dot{q}_g\left(1-\frac{T_0}{T_g}\right)+\dot{W}_{pe}} = \frac{m_{10}\left(\varphi_{10}-\varphi_9\right)}{m_7\varphi_7 + m_3\varphi_3 - m_4\varphi_4} \tag{23}$$

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

$$COP_{cooling} = \frac{\dot{q}_e}{\dot{q}_g} = \frac{m_{10}\left(h_{10}-h_9\right)}{m_7h_7 + m_3h_3 - m_4h_4} \tag{24}$$

For the mass balance, the different equations are given by O. Kaynakli *et al.* [30]:

$$\dot{m}_w = \dot{m}_s + \dot{m}_{H_2O} \qquad \text{(Total mass balance)} \tag{25}$$

$$\dot{m}_w X_w = \dot{m}_s X_s \qquad \text{(LiBr mass balance)} \tag{26}$$

where x is the LiBr concentration and subscripts w and s are for weak solution and strong solution respectively.

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

$$\dot{m}_s = \frac{X_w}{X_s - X_w}\dot{m}_{H_2O} \tag{27}$$

$$\dot{m}_w = \frac{X_s}{X_s - X_w}\dot{m}_{H_2O} \tag{28}$$

Properties of the LiBr-H_2O solution and typical overall heat transfer coefficient were adopted from Gur Mittelman [31]. For the heat transfer area or total conductance, we adopted the values: $(UA)_a$ = 1.800 kW/K, $(UA)_e$ = 2.250 kW/K,

$(UA)_g = 1.000$ kW/K, $(UA)_c = 1.200$ kW/K.

4. Results and Discussion

During the first 30 min the system chiller needs to start up and at 6:30 AM when the sun rises the water temperature inside the collector start to increase; that's why an abnormal behaviour of some curves is observed before 7:00 AM. During this period, the solution temperature varies rapidly from the ambient temperature to the boiling temperature. The heat supplied to the generator during this period is small and partially used for running the machine until the different components reach the process temperature. Similar observation is made after 5:30 PM. This is because the auxiliary heater needs to start below a temperature not reached at those moments.

Table 2 shows the thermodynamic state of each of the points within the cycle, enthalpy, entropy and exergy can be slightly compared to that obtained by Talbi *et al.* [9] for the different points of the system. The values of the two studies are similar. The Table 3 shows the effectiveness of the generator, the absorber, the evaporator and the condenser. It is noted that the generator is a component with a low effectiveness. This indicates that the generator needs a particular attention and must be optimized.

Figure 2 shows the variation of insolation during the day chosen for our study. This curve has the same behaviour with that of the temperature of the

Table 2. Operating conditions for the cycle with heat exchangers at 12:30 pm.

i	$T(i)$ °C	$h(i)$ (kJ/kg)	$s(i)$ (KJ/kg)	$m(i)$ (Kg/s)	$x(i)$ (%)LiBr	$\varphi(i)$ (kJ/kg)
1	35	282.24	2.24	0.1836	60	151.18
2	218	282.24	2.24	0.1836	60	151.18
3	223	322.24	2.24	0.1836	60	191.16
4	219.17	1200.33	2.24	0.1707	55	1135.72
5	86	1237.00	2.24	0.1707	55	1172.39
6	73	1237.00	2.24	0.1707	55	1172.39
7	219.17	3429.00	6.11	0.0129	-	1138.61
8	36.47	2620.18	8.33	0.0129	-	−341.00
9	38.47	2620	8.33	0.0129	-	−341.00
10	34.39	267319	8.37	0.0129	-	−328.74
11	129	2716.6	6.99	0.55	-	166.2
12	71	2494.6	9.23	0.55	-	−741.14
13	35	146.76	0.50	0.25	-	−7.27
14	36	146.76	0.50	0.25	-	−7.27
15	35	146.76	0.50	0.4	-	−7.27
16	36	150.53	0.52	0.4	-	−7.24
17	35	146.76	0.50	0.28	-	−7.27
18	36	282.24	0.51	0.28	-	−7.25

Table 3. The effectiveness for each component.

Component	Effectiveness
Generator	0.35
Condenser	0.68
Evaporator	0.74
Absorber	0.78

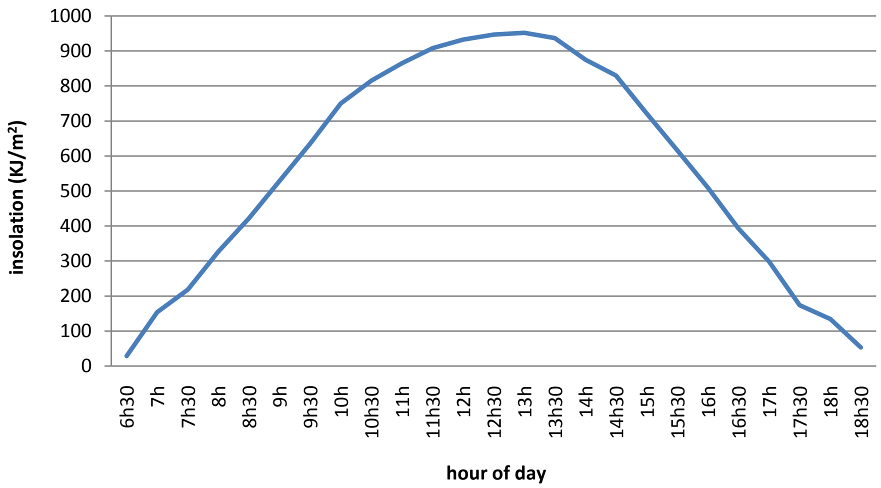

Figure 2. Variation of solar insolation during the day.

generator shown in **Figure 3**. The similar variation is observed in **Figure 4**, **Figure 5** with the generator load, the solar collector temperature and the exergy gained by the fluid in the solar collector respectively. These curves reached their maximum at 12:30 PM when the ambient temperature is also maximum. **Figure 6** shows the variation of the *COP* and *ECOP* of cooling during the day. The *COP* and *ECOP* depend on the generator load Q_{ge} and the evaporator load Q_{ev}. The evaporator load remains constant while the generator load increases. That's while instead of having *COP* and *ECOP* which with the temperature of the generator, we can observe a decreasing. **Figure 7** shows the variation of *COP* and *ECOP* of cooling with the temperature of the generator. **Figure 8** represents the variation of the exergy of the solar collector during the day which follows the variation of ambient temperature. **Figure 9** presents the variation of the exergy gained by the fluid in the solar collector during the day. One can observe that the exergy gained also by this important component of the system has the same evolution with the insolation. **Figure 10** shows the exergy loss of condenser. The energy remove from the refrigerant at this level makes the exergy loss decreasing. The variation of the exergy loss of absorber along the day is shown in **Figure 11**. This component is very complex with the fact that it's where the weak solution and the refrigerant are mixed to form the strong solution. The exergy loss is negative because the disorder (entropy) is higher than the order (enthalpy). **Figure 12** shows the exergy loss of generator that presents a diminution around

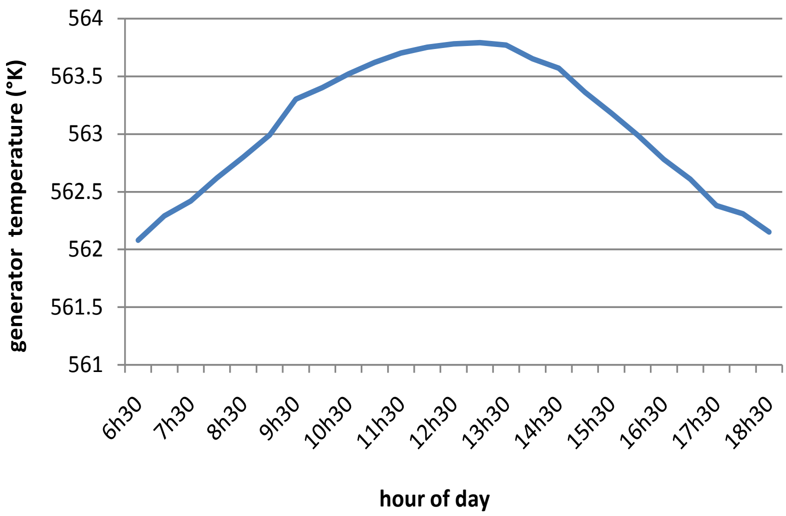

Figure 3. Variation of the generator temperature during the day.

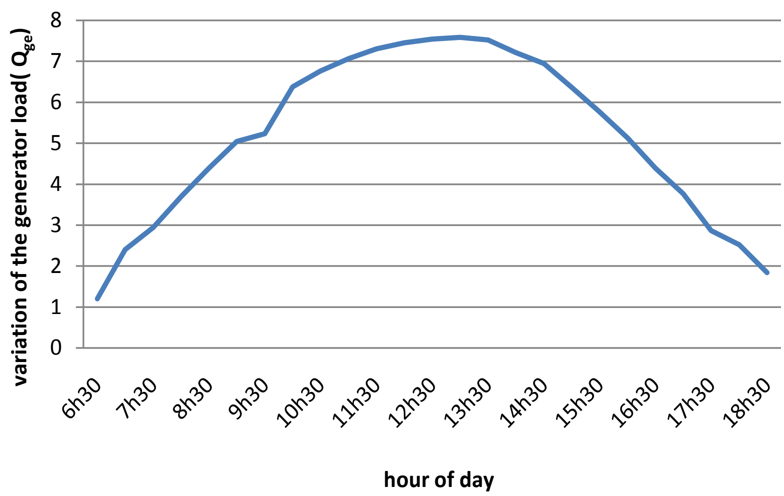

Figure 4. Variation of the generator load Q_{ge} during the day.

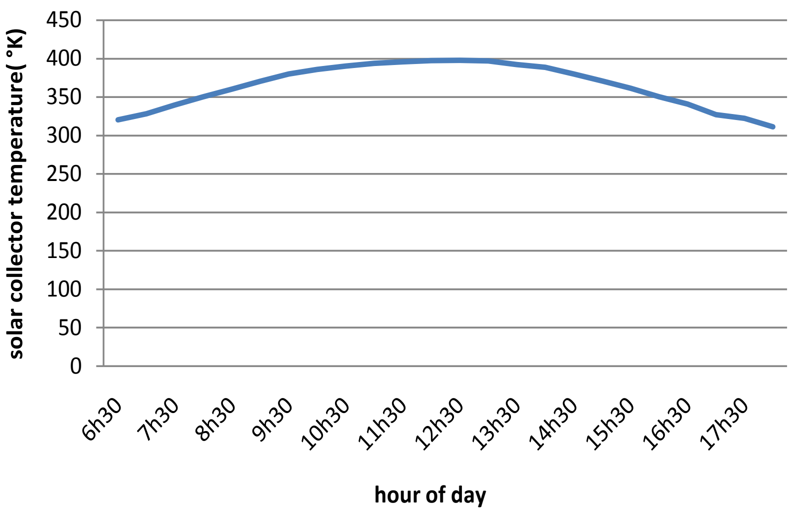

Figure 5. Variation of the temperature of the solar collector.

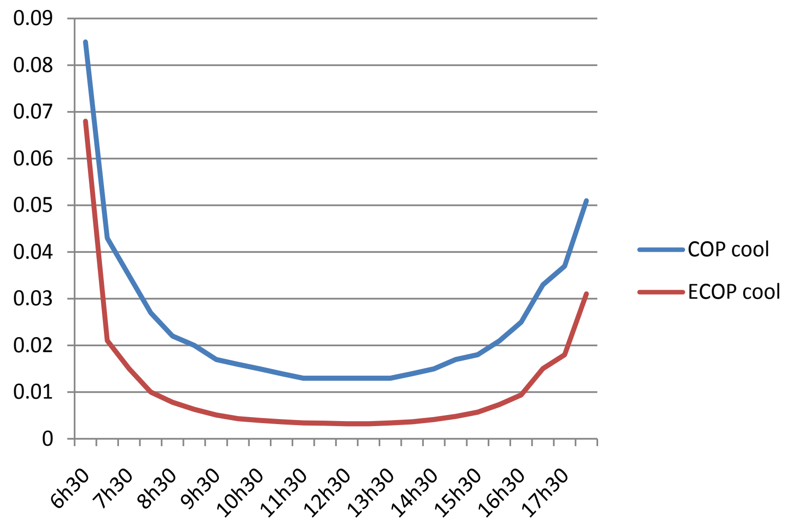

Figure 6. Variation of the *COP* and *ECOP* of cooling during the day.

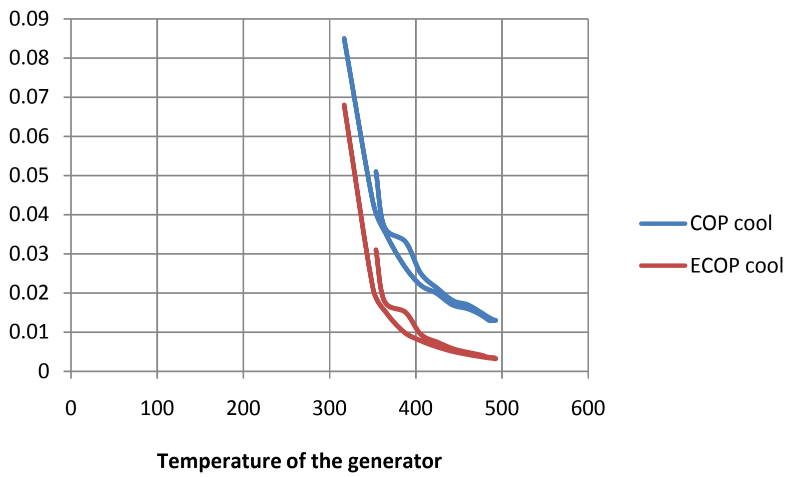

Figure 7. Variation of *COP* and *ECOP* with temperature of generator.

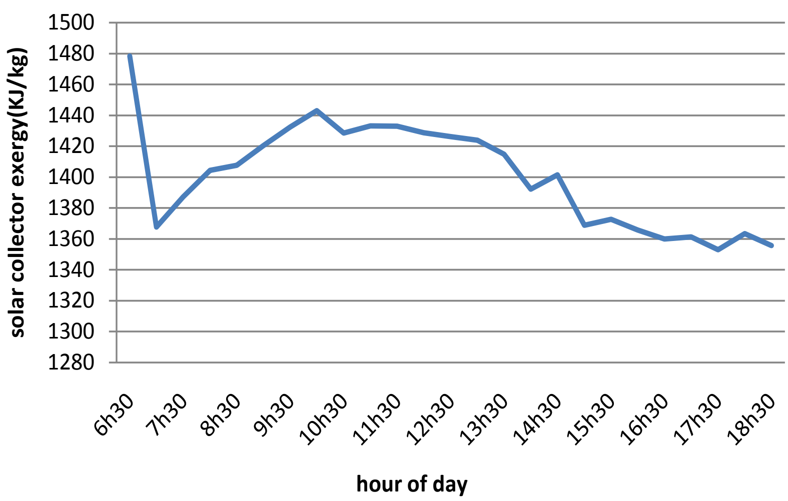

Figure 8. Variation of the exergy gain by the solar collector.

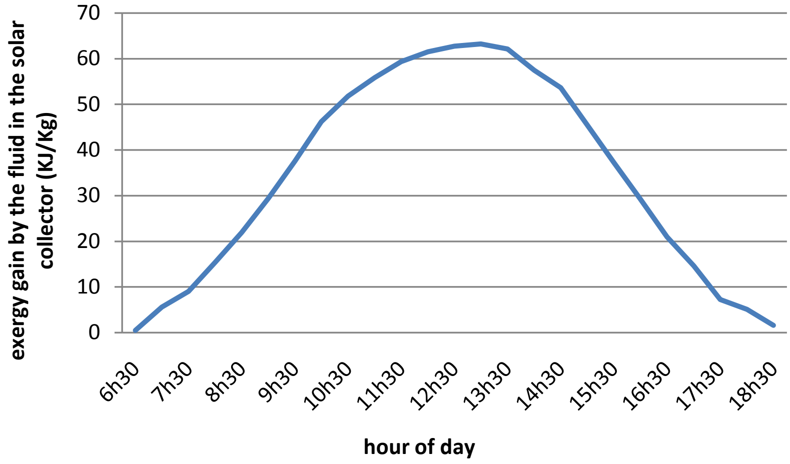

Figure 9. Variation of the exergy gain by the fluid in the solar collector.

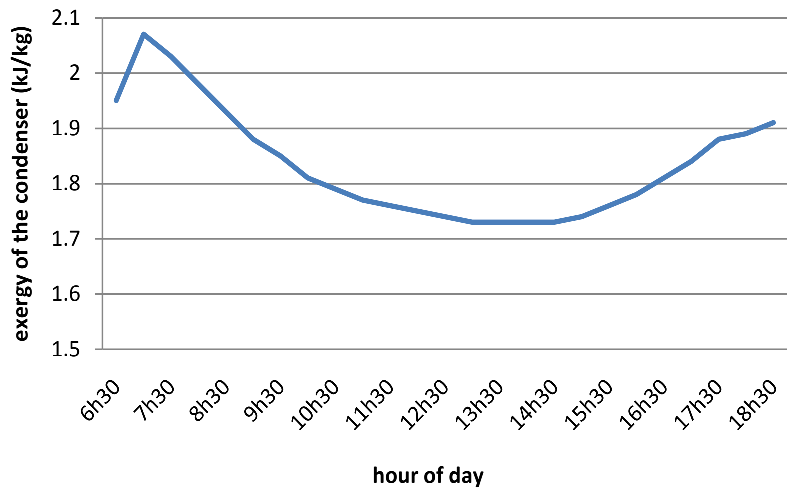

Figure 10. Variation of the exergy loss of the condenser.

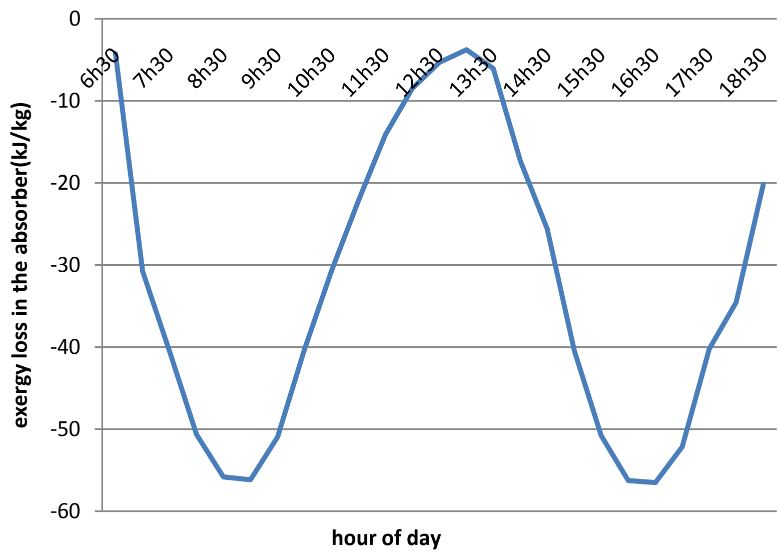

Figure 11. Variation of the exergy loss of absorber during the day.

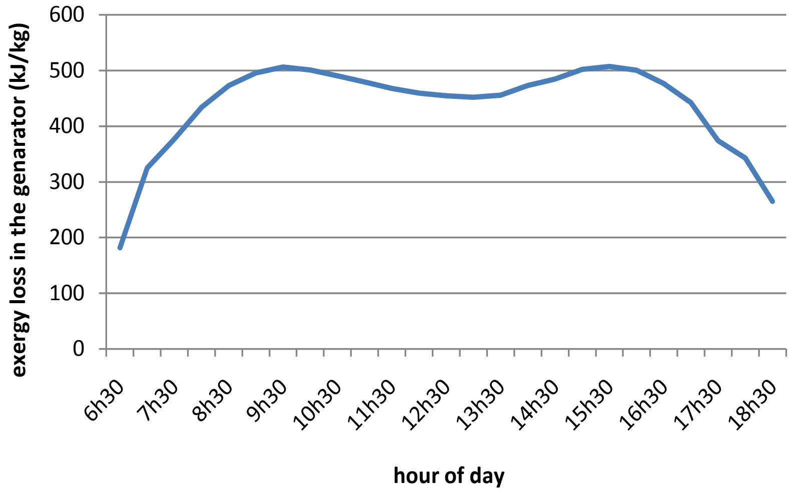

Figure 12. Variation of the exergy loss in the generator during the day.

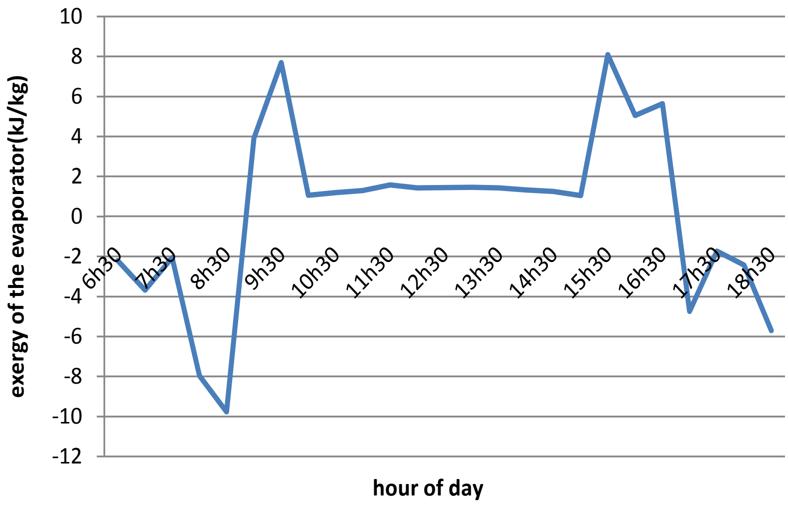

Figure 13. Variation of the exergy loss of evaporator.

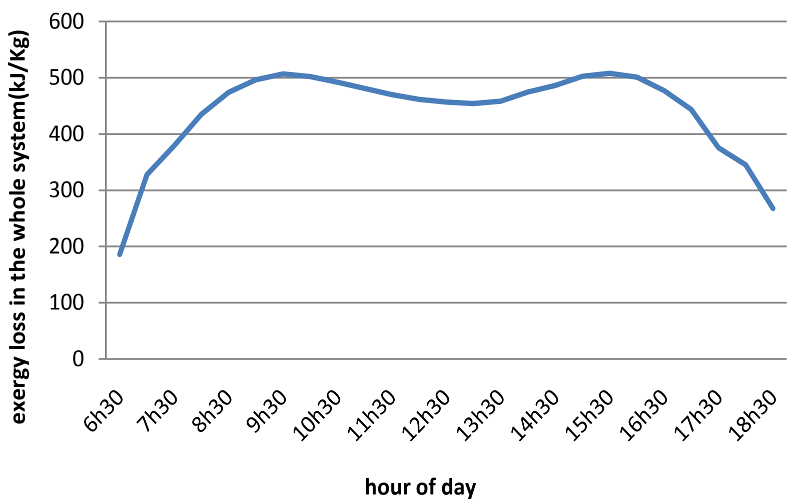

Figure 14. Evolution of the exergy loss of the whole system.

12:30 PM. This behaviour is normal because the insolation is high and the energy needed to run the system during this period is available. **Figure 13** presents the exergy loss of evaporator which is constant during the permanent period that is from 10 AM to 4 PM. Transitory periods which are before 10 AM and after 4 PM show the necessity of an auxiliary heater. **Figure 14** illustrates the variation of exergy loss of the whole system during the day. It's the same evolution with the exergy loss of the generator. This demonstrates the importance of the generator for absorption refrigeration systems.

5. Conclusions

By applying the first and the second laws of thermodynamics, the exergy analysis of a single-effect solar absorption refrigeration which working pair is LiBr-H_2O has been performed under weather conditions of Ngaoundere city, 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 solar collector of the absorption cooling device.

2) The exergy losses in the generator and in the absorber alternate; this is coherent because the generator absorbs energy while the absorber liberates energy.

3) The half hourly study has enabled us to have more accurate analysis of the exergy of the system. By doing this, we have investigated a more accurate study of the system.

The system can be implemented in Cameroon and in countries with high solar availability.

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Nomenclature

COP: coefficient of performance
ECOP: exergetic coefficient of performance
h: enthalpy
s: entropy
m: mass flow rate
Q: heat flow rate
T: temperature
P: pressure
φ: exergy
x: mass fraction of lithium bromide
Dpsi: variation of exergy of the system

Subscript

ref: reference
g: *ge*: generator
e: evaporator
a: absorber
c: condenser
j: ejector
exp: expansion valve
pump: pe: pump
sc: solar collector
ava: available