Parametric study and exergy analysis of solar water-lithium bromide absorption cooling system

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Abstract: This paper presents the energy and exergy analysis of single effect water lithium bromide absorption cooling system driven by the heat supplied by a field of solar thermal collectors with a cooling capacity of 10 kW. The work is devoted to the study, the evaluation and distribution of the destroyed exergy for each component constituting this kind of system. The thermodynamic models have been derived using the first and second laws of thermodynamics. These models are employed in a computer program using the Engineering Equation Solver (EES) software to perform the calculations and a sensitivity analysis to parameters is also presented. The results indicate that the contribution of some components to the overall exergy loss is very important. Exergy analysis illustrates also that the distribution of the destroyed exergy in the system between components depends strongly on the working temperatures.

Keywords: solar refrigeration; absorption; water-Lithium bromide; irreversibility; exergy; destruction.

Reference to this paper should be made as follows: Touaibi, R., Feidt, M., Vasilescu, E.E. and Tahar Abbes, M. (2013) ‘Parametric study and exergetic analysis of solar water-lithium bromide absorption cooling system’, *Int. J. of Exergy*, Vol. x, No. x, pp.xxx–xxx.

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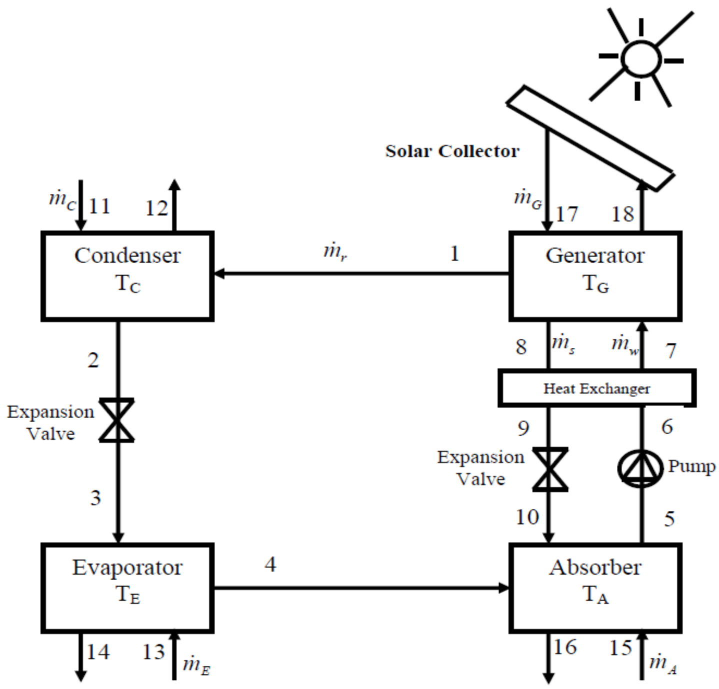
1 Introduction

Solar cooling is an attractive alternative since it has the advantage of removing the majority of harmful effects of traditional refrigeration machines and that the peaks of requirements in cold coincide most of the time with the availability of the solar radiation. The possible use of solar energy as the main heat input for a cooling system has led to several studies of available cooling technologies (Balghouthi et al., 2008). At present, various types of solar-powered systems are available for cooling applications (Xavier, 2006). In Europe, in recent years, more than 50 solar-powered cooling projects in different climatic zones were surveyed and analysed to identify future needs and evaluate the overall prospects of solar cooling (Balaras et al., 2007). The continuous increase in the cost and demand for energy has led to more research and development to utilize available energy resources efficiently by minimizing the waste energy. It is important to note that system performance can be enhanced by reducing the irreversible energy losses. According to the principles of the second law of thermodynamics, an absorption cooling system does not deplete the ozone layer and hence, it poses no danger to the environment (Zubair and Adewusi, 2004). Various researchers presented recently modelling and simulation studies of solar cooling and air-conditioning systems. Several computer models for describing the performance characteristics of absorption chillers are shown in (Joudi and Lafta, 2001; Karamangil et al., 2010; Klomfar and Pátek, 2006; Lucas et al., 2004; Sumathy and Li, 2001). In addition, the use of the solar energy in an absorption heat pump system has been investigated (Assilzadeh et al., 2005; Balghouthi et al., 2005; Eicker and Pietruschka, 2009; Fan et al., 2007; He et al., 2009; Mazloumi et al., 2008).The exergy analysis of absorption cooling systems without solar collector is made by several authors (BenEzzine et al., 2005; Gebreslassie et al., 2010; Izquierdo et al., 2005; Talbi and Agnew, 2000; Ucar and Inalli, 2006). S.C. Kaushik and A. Arora (2009) present the exergy analysis and the effects of generator, absorber and evaporator temperatures on the energy and exergy performances of the single effect water–lithium bromide absorption system without solar collector. The present study applies the first and second law of thermodynamics to study the performance of the solar water-lithium bromide absorption cooling when some parameters are varied. The entropy generation, destroyed exergy of each component of the system and the exergy efficiency of the system are calculated from the thermodynamic properties of the working fluids at various working conditions using the Engineering Equation Solver software (Klein, 2004). The variations of destroyed exergy of all the components and the exergy efficiency are then studied. It is important to note that this study is an extension of the thermodynamic analysis of this type of systems.

2. Solar absorption cooling system Description

Figure 1 shows the schematic diagram of a single effect solar water-lithium bromide absorption cooling system. This system has been the basis of most of study being appropriate for the solar air- conditioning. As shown in Fig. 1 the main components of a solar single effect LiBr/H2O absorption cooling system are the generator, the absorber, the condenser, the evaporator, the pump, the refrigerant expansion valve, the solution expansion valve, the solution heat exchanger and the solar collector.

Figure 1 Scheme of a solar water-lithium bromide absorption cooling



The collector receives energy from sunlight. The solar energy is then transferred to the cooling system. In the absorption system, solar heat energy makes the vapour to escape from the solution in the generator, leaving a strong solution in the generator. The water vapour thus-generated is at high temperature and pressure. It is then passed to the condenser where heat is removed and the vapour cools down to form a liquid. To return the refrigerant, at a low pressure, it expands through an expansion valve. In this state the refrigerant is a two-phase mixture at low pressure. In the evaporator, the refrigerant water is turned into vapour. The vapour then passes to the absorber. The strong solution leaving the generator to the absorber passes through a heat exchanger in order to preheat the weak solution entering the generator. In the absorber, the strong solution absorbs the water vapour leaving the evaporator. The weak solution leaves the absorber and enters into the pump, where increases its pressure up to the generator one then passes through the same heat exchanger to the generator again and the process is repeated. There are many types of solar collectors, which are used in absorption cooling system (Feidt, 1996; Sumathy and Li, 2000).

3 Mathematical models for the components

3.1 Assumptions

The calculations were performed based on the following assumptions.

- All components are assumed to be at steady state and steady flow process.
- The changes in the potential and the kinetic energy of the components are negligible.
- The heat loss and pressure drops in the piping are negligible.
- The expansion valve process is at constant enthalpy.

3.2 Mass balance

The governing equations of mass and type of material conservation for a steady state and steady flow system are (Gomri, 2010; Kizilkan et al., 2007):

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \tag{1}$$

$$\sum \dot{m}_{in} x_{in} - \sum \dot{m}_{out} x_{out} = 0 \tag{2}$$

Where $\dot{m}$ is the mass flow rate and x is mass concentration of LiBr in the solution

3.3 Energy balance

The first law of thermodynamics yields the energy balance of each component of the absorption system as follows:

$$\sum \dot{Q} - \dot{W} = \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out} \tag{3}$$

3.4 Exergy balance

In steady state and neglecting the kinetic energy and the potential energy and according to (Bejan, 2006), the exergy balance equation applied to a fixed control volume is given by:

$$0 = \sum \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_j - \dot{W} + \sum \dot{m}_{in} Ex_{in} - \sum \dot{m}_{out} Ex_{out} - \dot{E}x_D \tag{4}$$

Where $\dot{Q}$ is the heat transfer rate from or to the system; $\dot{W}$ the mechanical power supplied by or to the system. When the kinetic and potential energies are neglected, according to (Bejan, 2006), the physical exergy Ex can be evaluated as,

$$Ex = (h - h_0) - T_0 (s - s_0) \tag{5}$$

The number of transfer units (NTU) method is used to calculate the rate of heat transfer in each heat exchanger of the system as follows:

$$\dot{Q}_j = \varepsilon_j \dot{C}_j (T_{Sji} - T_j) \tag{6}$$

Where, T_{Sji} is the sources or sinks inlet temperatures, $\dot{C}_j = Cp_j . \dot{m}_j$ is the heat capacity rate and ε_j is the heat exchanger effectiveness of each component. Cp_j is a mean value of the specific heat supposed constant in the component j .

Applying energy conservation and exergy balance equation to each component of the cycles, the following equations can be developed to carry out the exergy analysis.

Solar collector

The performance of a solar collector is described by an energy balance that indicates the distribution of incident solar energy into useful energy gain, thermal losses, and optical losses. The energy balance equation of the solar collector can be written as follows (Duffie and Beckman, 2006; Ucar A and Inalli, 2006):

$$I_{total} . A_{rec} = \dot{Q}_u + \dot{Q}_{loss} + \dot{Q}_{stg} \tag{7}$$

Where I_{total} is a total solar radiation incident, A_{rec} is the collector area, $\dot{Q}_{loss}$ is the heat loss from the collector and $\dot{Q}_u$ is the useful heat transferred from the absorber to the fluid, $\dot{Q}_{stg}$ is the heat stored in the collector. The thermal energy lost from the collector to the environment by conduction, convection, and infrared radiation can be represented as the product of a heat transfer coefficient U_l times the difference between the mean absorber plate temperature $\overline{T}_p$ and the ambient temperature T_0 . In steady state ($\dot{Q}_{stg}$ =0) the useful energy output of a collector is the difference between the absorbed solar radiation and the thermal loss and can be written as follows:

$$\dot{Q}_u = A_{rec} \left[(\tau\alpha) I_{total} - U_l \left(\overline{T}_p - T_0 \right) \right] \tag{8}$$

Where $\tau\alpha$ is the transmittance-absorptance product.

The efficiency of solar collectors is the ratio of useful energy obtained in collector to solar radiation incoming to collector. It can be formulated as the following (Duffie and Beckman, 2006; Feidt et al., 2004; Ucar and Inalli, 2006):

$$\eta = \frac{\dot{Q}_u}{A_{rec} I_{total}} \tag{9}$$

The destroyed exergy by the solar collector is written as follows (Saidur et al., 2012):

$$\dot{E}x_{D,rec} = T_0 \left[\dot{m}_G \left(s_{17} - s_{18} \right) - \frac{\dot{Q}_u}{\overline{T}_p} \right] \tag{10}$$

Where $\overline{T}_p$ is the average temperature of the absorber plate.

Generator

$$\dot{Q}_G = m_r h_1 + \dot{m}_s h_8 - \dot{m}_w h_7 \tag{11}$$

The destroyed exergy in the generator is written as follows:

$$\dot{E}x_{D,G} = T_0 \left[\dot{m}_w \left(s_8 - s_7 \right) + \dot{m}_r \left(s_1 - s_8 \right) + \dot{m}_G \left(s_{18} - s_{17} \right) \right] \tag{12}$$

Absorber

$$\dot{Q}_A = \dot{m}_r h_4 + \dot{m}_s h_{10} - \dot{m}_w h_5 \tag{13}$$

The destroyed exergy in the absorber can be written as follows:

$$\dot{E}x_{D,A} = T_0 \left[\dot{m}_r \left(s_{10} - s_4 \right) + \dot{m}_w \left(s_5 - s_{10} \right) + \dot{m}_A \left(s_{16} - s_{15} \right) \right] \tag{14}$$

Solution Pump

The work input to the pump is given by

$$\dot{W}_{PUM} = \dot{m}_w v (p_6 - p_5) \tag{15}$$

The exergy destruction rate of pump is given as

$$\dot{E}x_{D,PUM} = T_0 \dot{m}_w \left(s_6 - s_5 \right) \tag{16}$$

Solution heat exchanger

$$\dot{m}_w (h_7 - h_6) = \dot{m}_s \left(h_8 - h_9 \right) \tag{17}$$

The effectiveness of exchanger is expressed as follows:

$$\varepsilon_{HEX} = \frac{T_9 - T_8}{T_6 - T_8} \tag{18}$$

The exergy destruction rate in the recuperator is obtained from

$$\dot{E}_{D,HEX} = T_0\left[\dot{m}_s\left(s_9 - s_8\right) + \dot{m}_w\left(s_7 - s_6\right)\right] \tag{19}$$

Solution expansion valve

The energy balance is written.

$$h_{10} = h_9 \tag{20}$$

The exergy destruction rate during the expansion process is expressed as follows:

$$\dot{E}x_{D,SV} = T_0\dot{m}_s\left(s_{10} - s_9\right) \tag{21}$$

Condenser

$$\dot{Q}_C = \dot{m}_r(h_1 - h_2) \tag{22}$$

The destroyed exergy in the condenser can be written as follows:

$$\dot{E}x_{D,C} = T_0\left[\dot{m}_r\left(s_2 - s_1\right) + \dot{m}_C\left(s_{12} - s_{11}\right)\right] \tag{23}$$

Refrigerant expansion valve

$$h_3 = h_2 \tag{24}$$

The destroyed exergy in the Refrigerant expansion valve can be written as follows:

$$\dot{E}x_{D,RV} = T_0\dot{m}_r\left(s_3 - s_2\right) \tag{25}$$

Evaporator

The evaporator refrigeration capacity can be calculated as

$$\dot{Q}_E = \dot{m}_r(h_4 - h_3) \tag{26}$$

The exergy destruction rate for the evaporator is given as

$$\dot{E}x_{D,E} = T_0\left[\dot{m}_r\left(s_4 - s_3\right) + \dot{m}_E\left(s_{14} - s_{13}\right)\right] \tag{27}$$

3.5 Total exergy destroyed

The total exergy destruction equation for the solar absorption cooling system is given as follows:

$$\dot{E}x_{D,total} = \sum_{j}^{N} \dot{E}x_{D,j} \tag{28}$$

3.6 Coefficient of Performance

In the absorption refrigeration system, the total energy supplied to the system is the total of the heat supplied in the generator and work done by the pump. The actual COP of the absorption chiller is calculated from (Gomri, 2010).

$$COP_{Cycle} = \frac{\dot{Q}_E}{\dot{Q}_G + \dot{W}_{PUM}} \tag{29}$$

3.7 Exergetic efficiency

The overall cycle exergetic efficiency has been evaluated as the ratio between the useful exergy output and the exergy input of the system (Bejan, 2006; Feidt, 1996).

$$\eta_{ex} = \frac{\dot{Q}_E\left(1 - \frac{T_0}{T_{CS}}\right)}{\dot{Q}_u(1 - \frac{T_0}{\overline{T}_p}) + \dot{W}_{PUM}} = 1 - \frac{\dot{E}x_{D,total}}{\dot{Q}_u(1 - \frac{T_0}{\overline{T}_p}) + \dot{W}_{PUM}} \tag{30}$$

T_{CS} is the medium logarithmic temperature of the cold source (at the evaporator).

4 Validation of simulation

In order to validate the present model, the simulation results have been compared with the available numerical data in the literature. The results of this study were compared with the simulation data published by (S.C. Kaushik and Arora, 2009). Table 1 presents the system performances for the same operating conditions used by S.C. Kaushik and Arora. According to comparison, the agreement between the two simulation results is good.

Table 1 Validation of the model with the results of the S.C. Kaushik and Arora.

Parameters: $T_G = 87.8°C$, $T_C = T_A = 38°C$, $T_E = 7.2°C$, $\dot{m}_r = 1\,kg/s$, $\varepsilon_{HEX} = 0.7$			
Components	Power [kW]		Difference (%)
	S.C. Kaushik and al.	Present work	
Generator	3095 .70	3093	0.08
Absorber	2945.27	2943	0.07
Condenser	2505 .91	2506	0.00
Evaporator	2355.45	2355	0.01
Exchanger	518.72	522,6	0.74
Pump	0 .0314	0,0309	1.5
COP $_{cycle}$ [-]	0 .7609	0,7615	0.07
η_{ex} [%]	11.75	11.78	0.25

5 Results and discussion

The different results presented in this work are obtained from solving the equations of the model developed above. We used the EES software (Klein, 2004) to solve these equations. Our model allows analysing and studying the effect of various parameters on system efficiency and the performance of each component constituting the latter. Table 2 exhibits the parameters and the performance of single effect absorption cooling system of the present work with the cooling capacity of 10 kW. The mean temperatures of the external sources are: $T_{ES} = 16°C$, $\overline{T}_p = 100°C$, $T_{SC} = T_{SA} = 30°C$, $T_0 = 25°C$

Table 2 parameters and performance of system

Parameters: $\dot{Q}_E = 10\,kW$, $T_G = 90°C$, $T_A = T_C = 36°C$, $T_E = 7°C$, $\varepsilon_{HEX} = 0.85$				
$\dot{Q}_G$ (kW)	$\dot{Q}_C$ (kW)	$\dot{Q}_A$ (kW)	$\dot{Q}_{HEX}$ (kW)	$\dot{W}_{PUM}$ (kW)
12.60	10.66	11.94	1.97	8.68 e-5
$COP_{Cycle} = 0.79$		$\eta_{ex} = 12.35$ %		

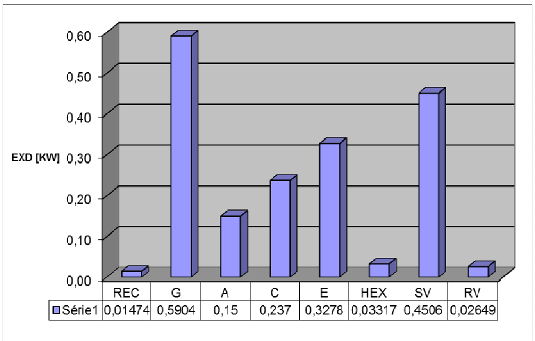
5.1 Destroyed exergy distribution

Fig.2. shows the destroyed exergy distribution in the solar absorption cooling system. The results show that the strongest exergy destruction appears in the generator 32.26%. Other values are 24.62% for the solution expansion valve, 17.91% for the evaporator, 12.95% for the condenser and 8.20 % for the absorber, 1.81 % for the heat exchanger,

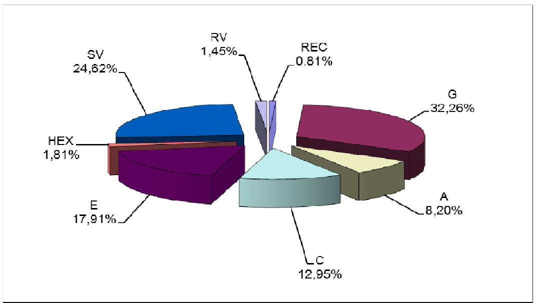
1.45 % for the refrigerant expansion valve and 0.81 % for the solar collector, the exergy destruction for other components is almost negligible.

The obtained results from Kaushik et al, is that considering only the refrigerating machine, the maximum of exergy destruction appears in the absorber component of the machine. Contrarily the solar system study we have performed, shows that the maximum exergy destruction appears at the hot side of the system, namely in the generator heat exchanger.

Figure 2 Destroyed exergy distribution in the system
$\dot{Q}_E = 10\ kW$, $T_G = 90°C$, $T_C = T_A = 36\ °C$, $T_E = 7°C$, $\varepsilon_{HEX} = 0.85$



(a)



(b)

5.2 Effect of generator temperature

The effects of the generator, condenser and absorber temperatures on the destroyed exergy of the components are shown in Figs.3-10.

Figs. 3-10 show the effect of generator temperature on the destroyed exergy in the solar collector, generator, absorber, condenser, pump, solution throttle valve and refrigerant throttle valve when the evaporator temperature and heat exchanger effectiveness are constants. The results show that the generator temperature has a great importance on the variation of exergy destruction in the solar collector, generator, absorber, condenser, pump and solution expansion valve Figs. 3-6 and Figs. 8,9. On the other side, we note that there is no influence on the variation of exergy destruction in the evaporator and refrigerant expansion valve Figs.6,10. The increasing of the generator temperature can increase the exergy destruction in the solar collector, condenser and solution expansion valve. On the other side this temperature allows the decreasing of the exergy destruction in the pump; it can appear a minimum regarding the pump. But we note that the mean value of destroyed exergy differs essentially a wing to the component. These values are important in the generator, absorber, condenser, evaporator and solution expansion valve, but small in the collector, refrigerant expansion valve, and solution pump.

Figure 2 Variation of destroyed exergy of the solar collector with the generator temperature.

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

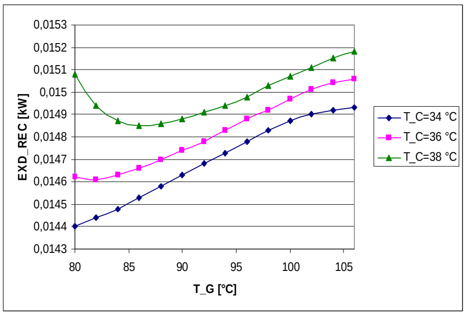


Figure 4 Variation of destroyed exergy of the generator with the generator temperature.

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

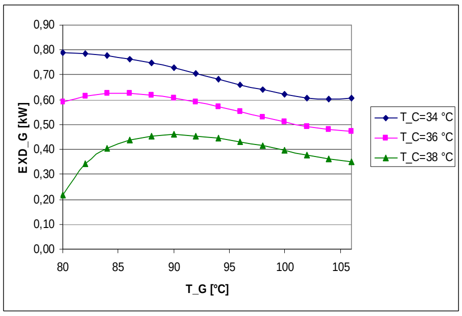


Figure 5 Variation of destroyed exergy of the absorber with the generator temperature

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

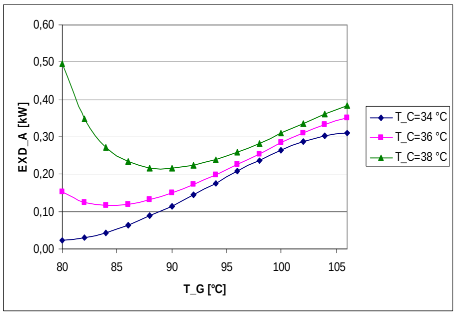


Figure 6 Variation of destroyed exergy of the condenser with the generator temperature

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

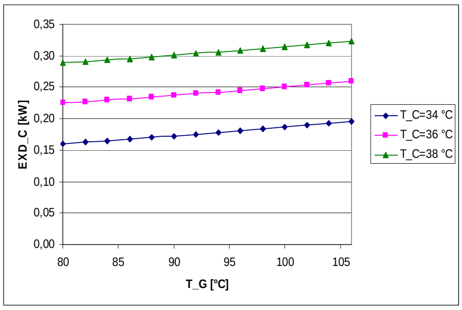


Figure 7 Variation of destroyed exergy of the evaporator with the generator temperature.

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

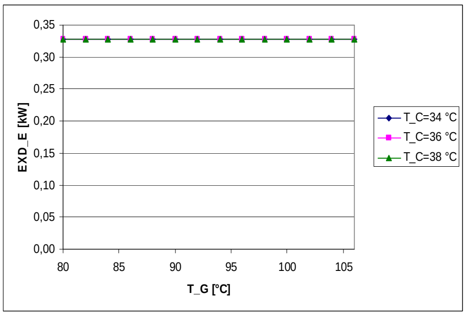


Figure 8 Variation of destroyed exergy of the pump with the generator temperature

$$\dot{Q}_E = 10\,kW\ , T_E = 7\,°C\ , \varepsilon_{HEX} = 0.85$$

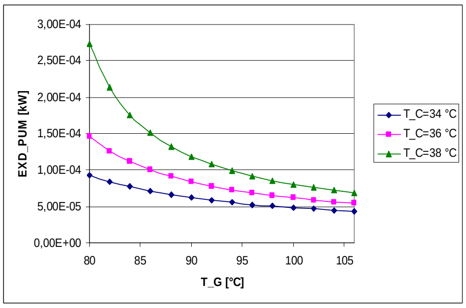


Figure 9 Variation of destroyed exergy of the solution valve expansion with the generator temperature.

$\dot{Q}_E = 10\ kW$, $T_E = 7\ °C$, $\varepsilon_{HEX} = 0.85$

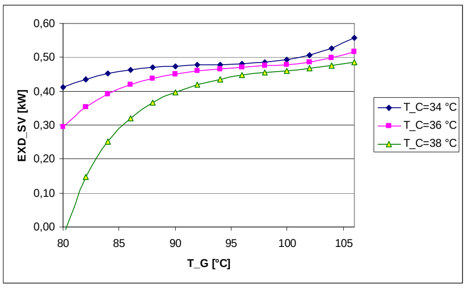
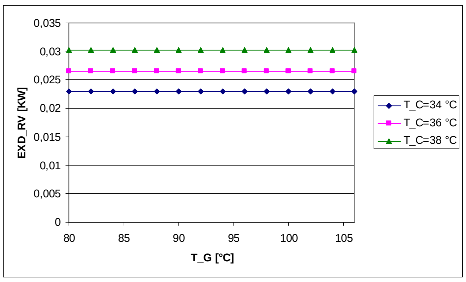


Figure 10 Variation of destroyed exergy of the refrigerant valve expansion with the generator temperature.

$\dot{Q}_E = 10\ kW$, $T_E = 7\ °C$, $\varepsilon_{HEX} = 0.85$



5.3 Effect of Evaporator temperature:

The effects of the evaporator, condenser and absorber temperatures on the exergy destruction in the components are shown in Figs.11-18.

Figs. 11-18 show the effect of evaporator temperature on the destroyed exergy in the solar collector when the generator temperature and heat exchanger effectiveness are constants. Concerning the effect of evaporator temperature on the exergy destruction and according the Figs. 11-18, we note that it also has a great importance on all system components except in the condenser where there is no variation exergy destruction. The increasing of evaporator temperature allows the decreasing of exergy destruction in the solar collector, the absorber, the evaporator, the pump and the refrigerant expansion valve and increases the exergy destruction in generator and the solution expansion valve. A gain we note the same tendency regarding the mean value of destroyed exergy with main components as in section 5.2.

Figure 11 Variation of destroyed exergy of the solar collector with the evaporator temperature.

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

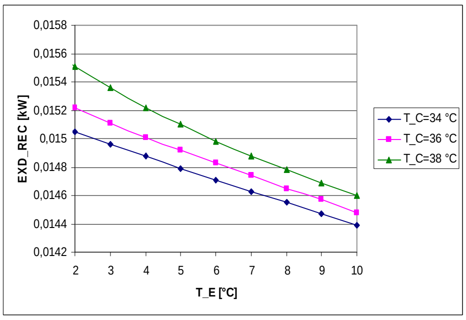


Figure 12 Variation of destroyed exergy of the generator with the evaporator temperature.

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

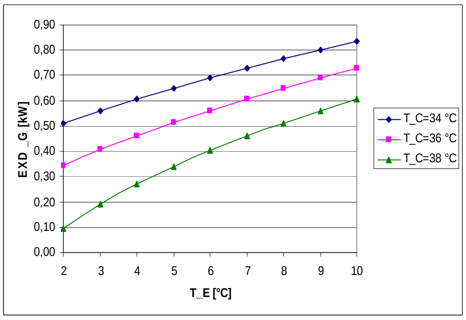


Figure 13 Variation of destroyed exergy of the absorber with the evaporator temperature

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

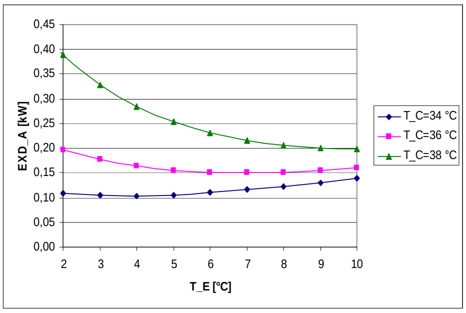


Figure 14 Variation of destroyed exergy of the Condenser with the evaporator temperature

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

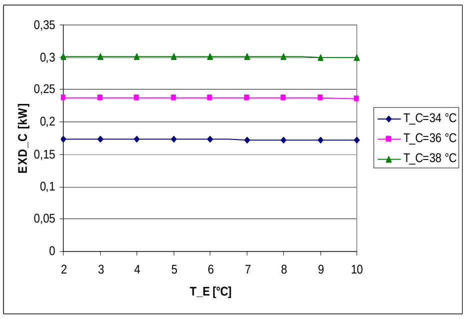


Figure 15 Variation of destroyed exergy of the evaporator with the evaporator temperature

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

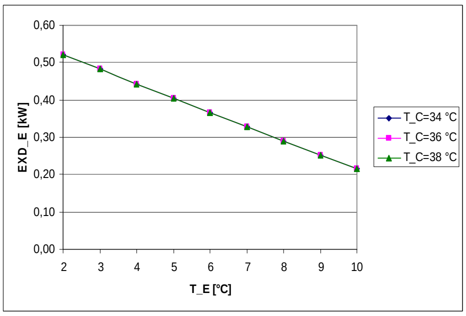


Figure 16 Variation of destroyed exergy of the pump with the evaporator temperature

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

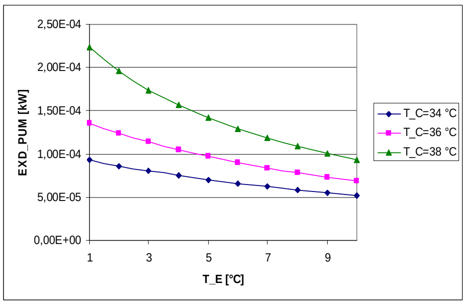


Figure 17 Variation of destroyed exergy of the solution throttle valve with the evaporator temperature.

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$

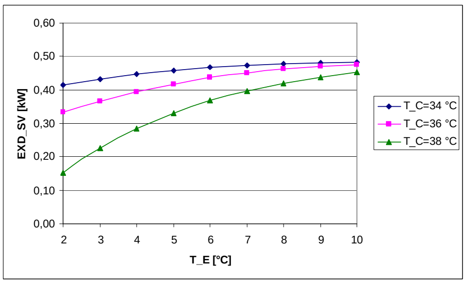
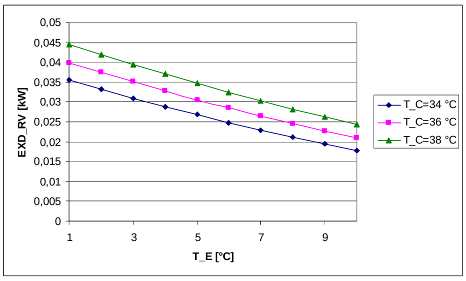


Figure 18 Variation of destroyed exergy of the refrigerant throttle valve with the evaporator temperature.

$\dot{Q}_E = 10\ kW$, $T_G = 90\ °C$, $\varepsilon_{HEX} = 0.85$



We note also that when the temperature increases in condenser, the exergy destruction for all components of system increases except for the generator.

5.4 System Performance

The Figs.19, 20 show the effects generator and evaporator temperature on the exergy efficiency of the absorption cooling system.
The results show that the generator temperature and evaporator temperature are parameters that play an important role on the exergy efficiency of the system. It varies from 12 % to 12.65 % with generator temperature (approximately 3 % , when the generator temperature moves from 80 °C to 105 °C; it appear that an optimum value of generator temperature exists) Fig.19, even the influence of the evaporator temperature on exergy efficiency varies from 11.6 % to 12.65 % Fig. 20. This influence is more pronounced and monotonous (approximately 10 % more when the evaporator temperature moves from 1 to 10 °C).

Figure 19 Variation of exergy efficiency with generator temperature

$$\dot{Q}_E = 10\,kW, T_E = 7\,°C, \varepsilon_{HEX} = 0.85$$

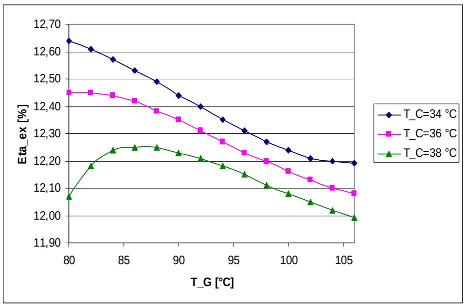
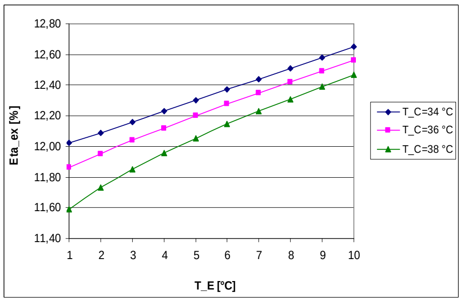


Figure 20 Variation of exergy efficiency with evaporator temperature

$$\dot{Q}_E = 10\,kW, T_G = 90\,°C, \varepsilon_{HEX} = 0.85$$



6. Conclusion

Exergy analysis and parametric study of a solar absorption cooling system are conducted. The system is examined under the variation of the generator temperature, condenser temperature and evaporator temperature. The main findings are: The most significant sources of exergy destruction rates are first in the generator, second in the solution throttle valve, third in the evaporator and finally in the condenser. Therefore, further improvements in designing these four components are needed while the maximum exergy is destroyed during desorption of solution in the generator; it represents over 32.26 % of the total exergy destruction in the overall system. The contribution of some components of system to exergy destruction is negligible like the pump. Finally the overall exergy efficiency could present an optimum with the generator temperature but increases significantly with evaporator temperature. The optimization of the whole system regarding these results is in due course, concerning generator temperature at first.

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Nomenclature

$\dot{m}$	mass flow rate (kg/s)
h	specific enthalpy (kJ/kg)
s	specific entropy (kJ/kg. K)
T	temperature (K)
$\dot{W}$	mechanical power (kW)
$\dot{Q}$	thermal energy rate (kW)
COP	Coefficient of performance (-)
x	mass fraction (-)
v	specific volume (m^3/kg)
p	pressure (kPa)
$\dot{E}x$	exergy rate (kW)
A	area (m^2)
U_l	overall loss coefficient (W/ m^2.K)
Greek symbols	
ε	effectiveness (-)
η	efficiency (-)
τ	transmittance (-)
α	absorptance (-)
Subscripts	
REC	Receiver
A	Absorber
G	Generator
HEX	Heat exchanger
PUM	Pump
E	Evaporator
C	Condenser
SV	Solution expansion valve
RV	Refrigerant expansion valve
ex	Exergetic
D	Destroyed
r	Refrigerant
s	Strong
w	Weak
in	Inlet
out	Outlet
0	Reference state