

SIMULATION AND ANALYSIS OF A SOLAR COOLING SYSTEM

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ABSTRACT:

The present paper focuses on a study model of a trithermal absorption machine, operating at H₂O/LiBr couple, powered by a planar solar collector. Therefore, the mathematical model is based on the principles and laws of thermodynamics as well as the heat and mass transfer equations, expressed for the refrigeration cycle. In fact, the results of the complete numerical simulation of the machine, developed within the framework of this study, allow predicting the performances of this type of installation according to the various parameters as well as the operating temperatures and the flow rate in the absorber.

1. INTRODUCTION

During the last decade, the increase in living standards, the growing demand for comfort and the increase in summer temperatures have led to a strong development of air conditioning, with the development of the global economy, the energy consumption of buildings has increased and represents about 30% of all used energy (Aebischer, 2007; Grignon-masse, L., 2010). In addition and consequently, buildings (residential and tertiary) alone account for 66% of direct GHG emissions in 2010 (PEB heating regulations, Brussels Environment - IBGE, Heating and air conditioning department, November 2014). The interest of solar cooling lies in the simultaneous demand for cooling and sunshine. When the heat necessary for the operation of the refrigerating machine is provided by the sun, the cold provided is free (no cost, no pollution). In the other hand, to predict the unsteady behavior of the system as a whole and induced by the variable character inherent to the operation of each of the subsystems and the sources/sinks to which they are connected (Mugnier D, Meunier F., 2013), the different dynamics involved concern successively:

1. The solar source ;
2. The means by which it is captured (nature and technology of the sensor: plane, vacuum) and with what quality (in terms of temperature and power) ;
3. The refrigeration unit configuration in charge of transforming the calorific impact into a refrigeration effect by means of a trithermal absorption machine ;
4. Nature of the cooling requirement (cooling loads) ;
5. Analyze its behavior and be able to deduce its performance at any time and quantify its seasonal efficiency.

Modeling and simulation of absorption refrigeration cycles have always been a constant concern in research in the field of air

conditioning and cold production (Poggi, 2018; Colbourne, 2018). In this study, we present a model of an absorption cycle, operating between three heat sources. The obtained results will be confronted with those of a complete numerical simulation of the real absorption machine operating with lithium bromide/water couple.

2. OPERATING PRINCIPLE

2.1 Absorption principle

Liquid absorption refrigeration machines work thanks to the ability of some liquids to absorb (exothermic reaction) and desorb (endothermic reaction) a vapor. They also use the fact that the solubility of this vapor in the liquid depends on the temperature and pressure. The most used couples are Ammonia+Water (NH₃/H₂O, where ammonia is the refrigerant), and Water+Lithium Bromide (H₂O/LiBr, water being the refrigerant (Lee et al., 2000; Sencan, 2005; Aniès, 2011). The first one allows to make negative cold for air-conditioning or industrial cooling needs while the second one, which is the couple used in our study, can only make positive cold (triple point at 0°C) so it is exclusively intended for building cooling. The transformation cycle is described as follows:

The water vapor created in the evaporator is fed to the absorber. The absorber contains the absorbing solution (LiBr) which is continuously pumped to the bottom of the vessel for spraying. The LiBr absorbs the water vapor out of the evaporator and thus maintains the low pressure in the evaporator necessary for vaporizing the refrigerant. As it absorbs water vapor, the absorbing solution becomes more and more diluted. Eventually, it would become saturated and unable to absorb any more. Therefore, the solution is regenerated in the concentrator (heated to approximately 85°C), some of the water evaporates.

The regenerated solution returns to the absorber. Finally, the water vapor extracted from the concentrator is led to the condenser, where a cold-water circulation cools it. The condensed water returns to the evaporator (Figure 1).

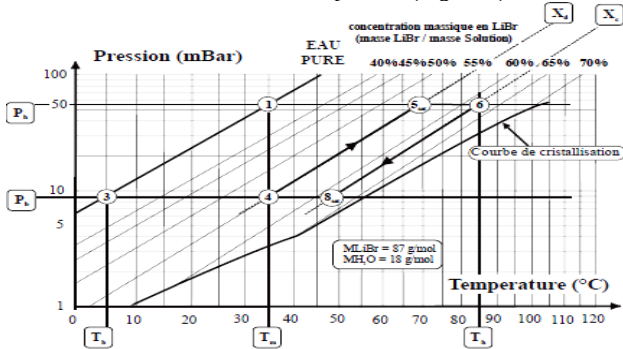


Figure 1. Absorption cycle representation on an Oldham diagram, H₂O/LiBr couple

The temperatures of the sources T_m (environment) and T_b (cold production) fix the two pressures P_h and P_b. These two pressures correspond to the saturated vapor pressures of water (isotitres LiBr = 0) at temperatures T_m (point 1) and T_b (point 3).

2.2 Variation of the titer during the cycle

The titer X_c of the concentrated solution at the disorder outlet is given by the intersection of the T_h isotherm and the P_h isobar (point 6). In a similar way, the titer x_d of the diluted solution at the absorber outlet corresponds to the intersection of the isotherm T_m and the isobar P_b (point 4). The absorber receives the concentrated solution from the disorder, after it has been expanded in the expansion valve (6-8). This pressure drop results in a drop in temperature due to partial vaporization of the solution. The absorber thus receives a mixture of liquid solution and vapor (8), for which it is not possible, with the help of this diagram, to specify either the temperature or the relative mass of the two phases. At the inlet of the absorber, this mixture is cooled, on the one hand by mixing with the cold steam coming from the evaporator, on the other hand, directly by the heat transfer fluid. The schematic diagram of a solar cooling system is shown in figure 2. The heat produced by the solar thermal collector field (1) is stored in the hot water tank (2) before being fed to the absorption machine generator (3). The cold water produced by the evaporator of the absorption machine is stored in the cold-water tank (4) before being distributed to the building to be cooled (6). Finally, the absorber and the condenser of the absorption machine are cooled by the cooling tower (5).

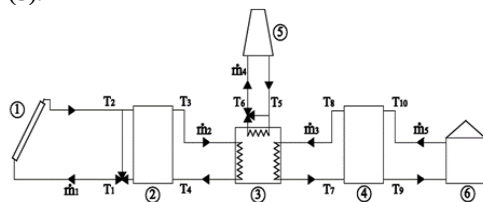


Figure 2. Schematic diagram of a solar cooling system; 1. Solar collector field; 2. Hot water tank; 3. Absorption machine; 4. Chilled water tank; 5. Cooling tower; 6. Building

3. INSTALLATION MODELLING

The ultimate objective of this modelling is to evaluate the cooling production and the global performances of our solar cooling installation according to the external solicitations (solar radiation, external temperature, building occupation). These parameters being very variable from one moment to another, the modeling in dynamic regime seems inevitable to be able to take into account as precisely as possible all the physical phenomena related to our problem (Lucas F, 2006).

3.1 Simplifying assumptions used in the model

For the application of the principles of thermodynamics to a real cycle, the following basic conditions and assumptions are applied:

1. The temperatures in the exchangers (generator, condenser, evaporator and absorber) are assumed uniform over the entire volume considered.
2. The fluid in the condenser and evaporator is pure water;
3. Thermodynamic equilibrium is reached in each component;
4. It is assumed that the liquid fluid in each component leaves in a saturated state;
5. The variables are assumed uniform in space.

3.2 Plant elements modeling:

3.2.1 The cooling tower

In order to model the cooling tower, we have considered a counter flow heat exchanger. The primary fluid (1) that circulates inside the exchanger is the water to be cooled, while the air, the secondary fluid (2), blows around it. We have chosen to use the efficiency method whose formula is the following:

$$\phi_{\text{échange}} = \left(\dot{m} \cdot C_p \right)_{\min} \cdot \eta \cdot \Delta T_{\max} = \dot{m}_2 \cdot C_2 \cdot (T_{2s} - T_{2e})$$

$$= -\dot{m}_1 \cdot C_1 \cdot (T_{1s} - T_{1e})$$

(1)

$$\Delta T_{\max} = T_{1e} - T_{2e}$$

(2)

$$\left(\dot{m} \cdot C_p \right)_{\min} = \min \left(\dot{m}_1 \cdot C_1, \dot{m}_2 \cdot C_2 \right) \quad (3)$$

$$\eta = \frac{1 - e^{[-N \cdot (1-C)]}}{1 - C \cdot e} \quad (4)$$

$$N = \frac{U \cdot S}{\left(\dot{m} \cdot C_p \right)_{\min}}$$

$$C = \frac{\left(\dot{m} \cdot C_p \right)_{\min}}{\left(\dot{m} \cdot C_p \right)_{\max}}$$

3.2.2 The absorption machine

In order to model the absorption machine we used the operating curves given by the manufacturer (Schüco, 2007a). Thanks to these curves and depending on the water temperature at the generator inlet, we will be able to determine the cooling power of the machine, the power at the generator and the cooling power.

3.2.3 The solar collector field

The modeling of our solar collector is based on the efficiency method. The efficiency equation (Aebischer, 2007b) is given by the manufacturer and depends on the inlet and outlet temperatures of the collector, and on the climatic conditions (the outside temperature T_{ext} and the sunshine E):

$$\eta_{\text{capteur}} = 0.798 - 2.275 \cdot \frac{T_{\text{moy}} - T_{\text{ext}}}{E} - 0.022 \cdot \frac{(T_{\text{moy}} - T_{\text{ext}})^2}{E} \quad (5)$$

$$T_{\text{moy}} = \frac{T_{\text{ecapteur}} + T_{\text{scauteur}}}{2}$$

$$\Delta U = \phi_{\text{reçu}} - \phi_{\text{transmis}} \quad (6)$$

$$\phi_{\text{reçu}} = \eta_{\text{capteur}} \cdot S_{\text{capteur}} \cdot E$$

$$\phi_{\text{transmis}} = \dot{m} \cdot C_p \cdot (T_e - T_s)$$

$$dU = M_{\text{eau}} \cdot C_{\text{peau}} \cdot \frac{dT}{dt} = \eta_{\text{capteur}} \cdot S_{\text{capteur}} \cdot E - \dot{m} \cdot C_p \cdot (T_e - T_s) \quad (7)$$

3.2.4 Storage tanks

In order to model the storage tanks, we will proceed in the same way by applying the first principle of thermodynamics. Example for the hot water tank (Solarfocus.2018):

$$M \cdot C_p \frac{dT_{\text{eauballon}}}{dt} = \phi_{\text{transmis}} - \phi_{\text{perdu}} - \phi_{\text{générateur}} \quad (8)$$

The assumptions are as follows:

1. The temperature and density of the fluid in each stratum are uniform and constant at each time step,
2. The heat loss to the outside of the tank and the conduction in the tank walls is small enough to avoid the formation of temperature gradients on the other two dimensions,
3. The pump of the generator circuit is switched on when $T_{\text{eau-balon}} > 80^\circ\text{C}$.
4. The flow rates in the solar and generator circuit are constant when the pumps are running,
5. It is considered that the pipes are perfectly insulated (no heat loss).

3.2.5 The building

We present here a simplified model of the building. For this, we will consider a single zone building with the following main characteristics:

Length = 13m Glazing: Single glazing ($S = 25\text{m}^2$) Occupants: 105 persons; Width = 13m; Wall: reinforced concrete (thickness = 16cm); Lighting: 22 neon lights of 58W

Height = 3m; Air infiltration = 1 vol/h; Occupancy: from 8h00 to 18h00

Let us now apply the first principle:

$$dU = \phi_{\text{apport extérieur}} + \phi_{\text{charges internes}} - \phi_{\text{rafraichissement}} \quad (9)$$

4. SIMULATION ENVIRONMENT

We have decided to simulate a model of a solar cooling system under the TRNSYS environment. This chosen model is functional at the Institute of Civil Engineering at the University of Reunion in France (Figure 3). To operate (Castaing-lasvignottes and al, 2008), the model needs climatic data (sunshine and outdoor temperature), the site chosen for these data is that of Bechar-Algeria. The simulated configuration is as follows:

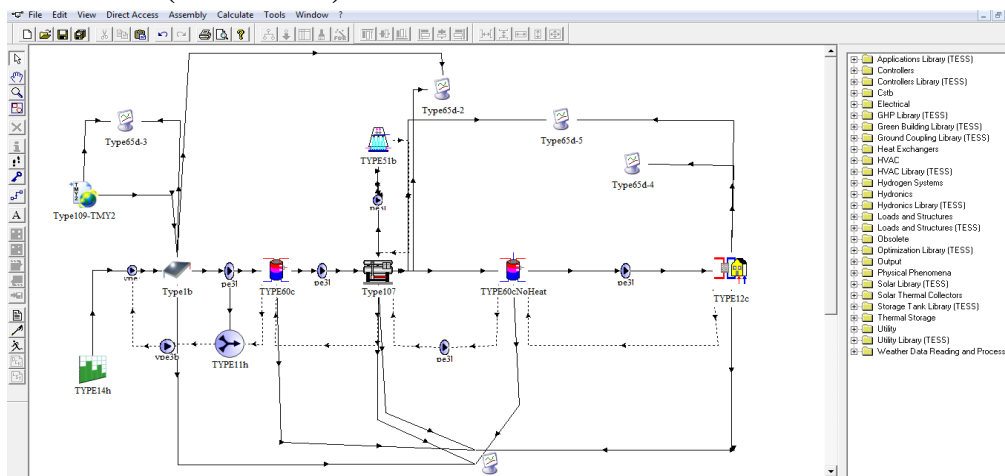


Figure 3. Model of the solar cooling system in TRNSYS

5. SIMULATION RESULTS

5.1 Performance coefficient COP:

COP coefficient of performance is given by the relationship:

$$COP = \frac{Q_{evap}}{Q_{gen}}$$

So than:

$$COP = \frac{T_e}{T_g} * \frac{T_g - T_e}{T_c - T_e}$$

245

The performance coefficient COP of the absorption machine is shown in Figure 4, it is 0.6254. It can be seen that there are two regimes, the first one is a transient regime at the start of the machine, and the second one is a steady state representing the normal operation of the machine.

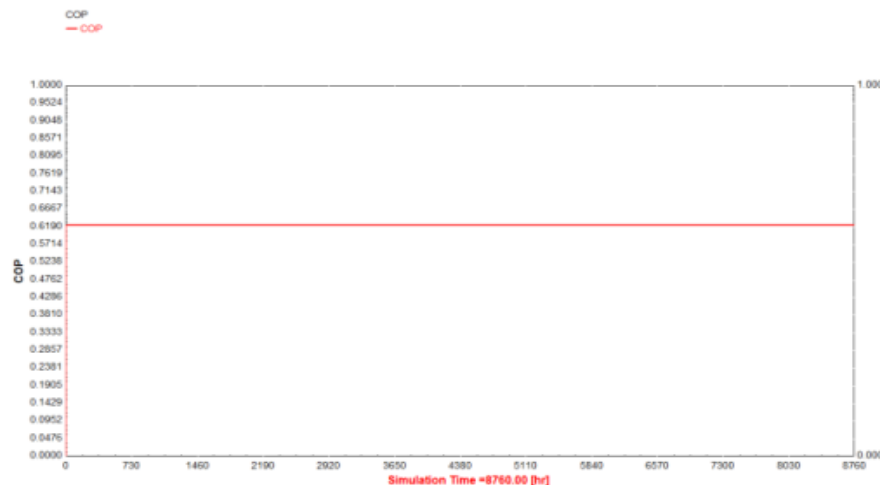


Figure 4. Machine performance coefficient

5.2 Temperatures of the different components of the installation:

Thus, the temperatures of the various components of the machine are shown in the following figures:

For the generator temperature, it is between 41 and 89 °C, the condenser temperature is between 17 and 38°C and the evaporator temperature is between 8 and 13°C. The major problem that arises in this case of refrigerating machine is the

crystallization of Lithium Bromide/Water solution, to overcome this problem, a control of the generator temperature is imposed and limits it to 90°C, that of the condenser is limited to 45°C (Schüco, 2007a). The temperature difference between the evaporator and the interior is 12°C, which allows the climate to be centralized inside the room. This difference is sufficient at the evaporator and ensures a comfortable temperature in the room.

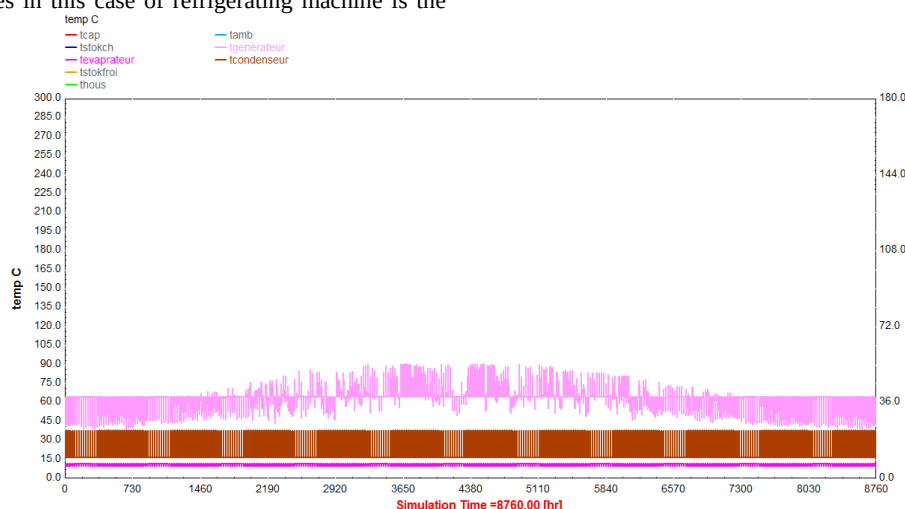
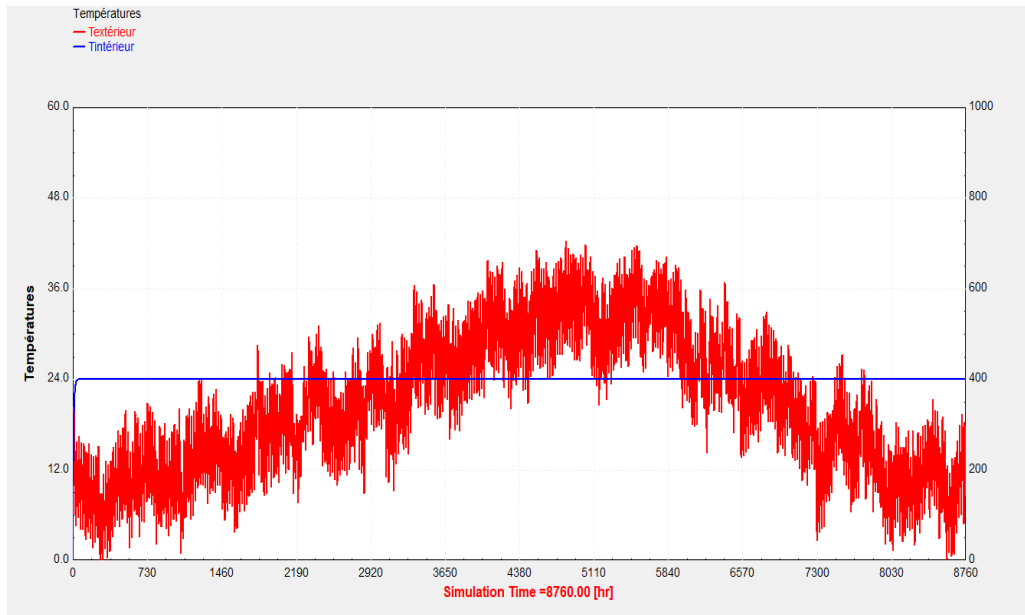


Figure 5. Temperatures of the different parts of the machine

Figure 6 illustrates the annual variation of the indoor and outdoor temperatures (T) of the studied room. This same figure shows us the air conditioning needs. It can be seen that,

whatever the outside temperature, the inside temperature is around 24°C (comfort temperature). It should be noted that for the winter and autumn case, we have activated the winter balance.



246

Figure 6. Indoor and outdoor temperatures

5.3 Effect of the sensor surface on the sensor output and generator temperature

We can see that as we increase the collector surface, the

temperatures increase as the heat transfer fluid has enough time to exchange the heat absorbed by the absorber of the solar collector (Figures 7-9).

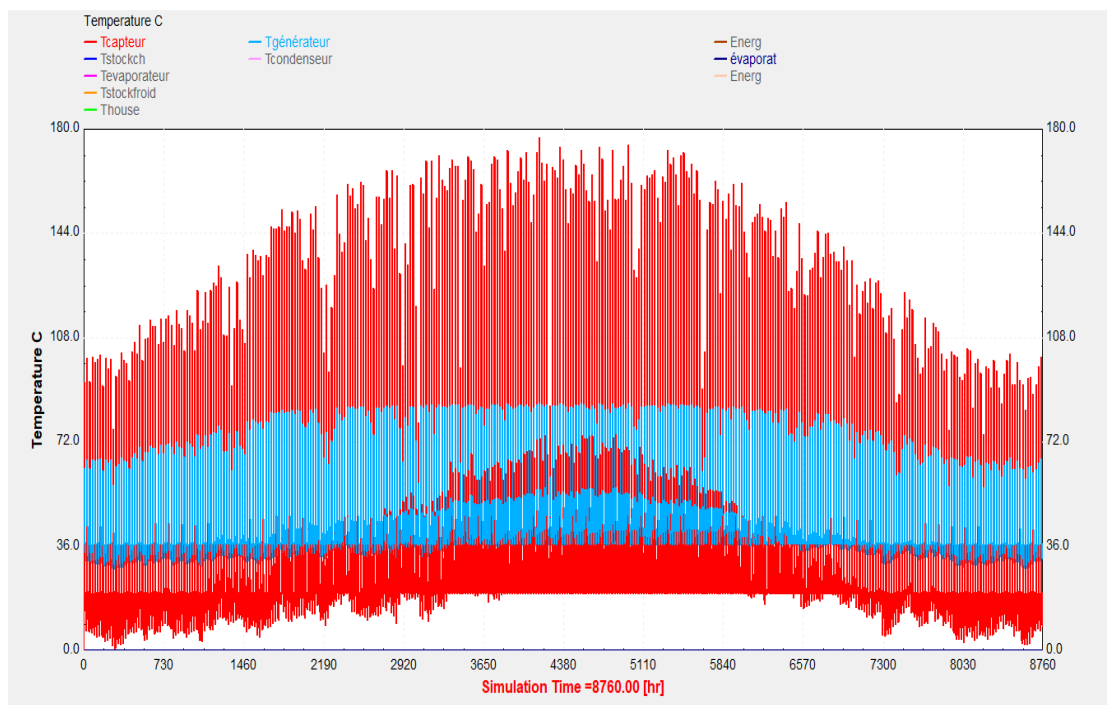


Figure 7. Influence of collector area on collector and generator temperature for a collector area of 2 m² de captage sur la température du capteur et du générateur pour une surface de captage de 2 m²

247

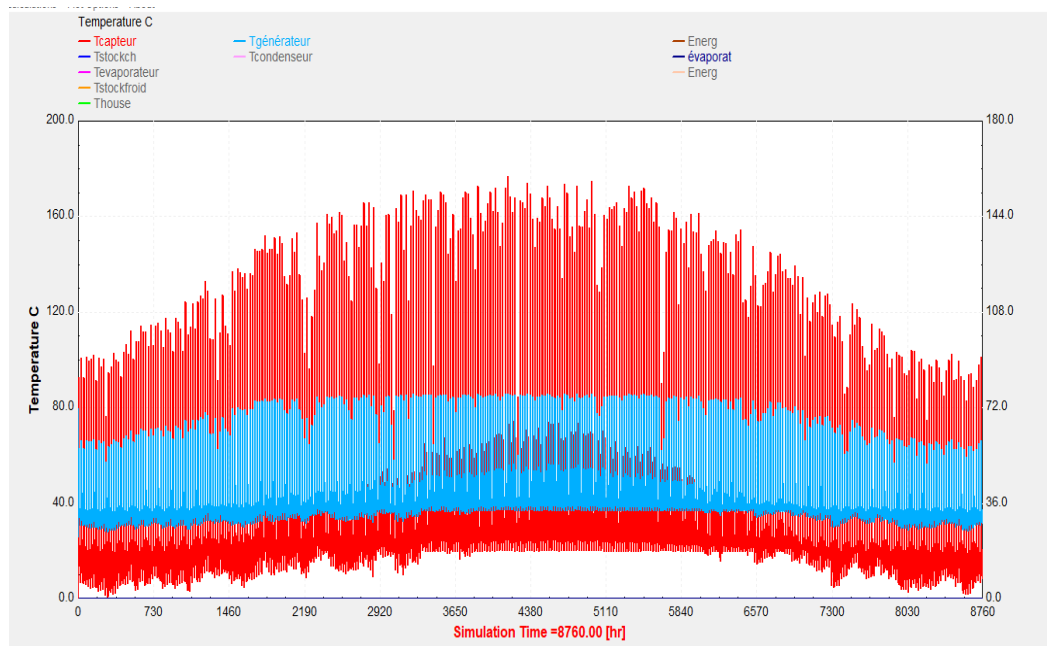


Figure 8. Influence of collector area on collector and generator temperature for a collector area of 38 m²

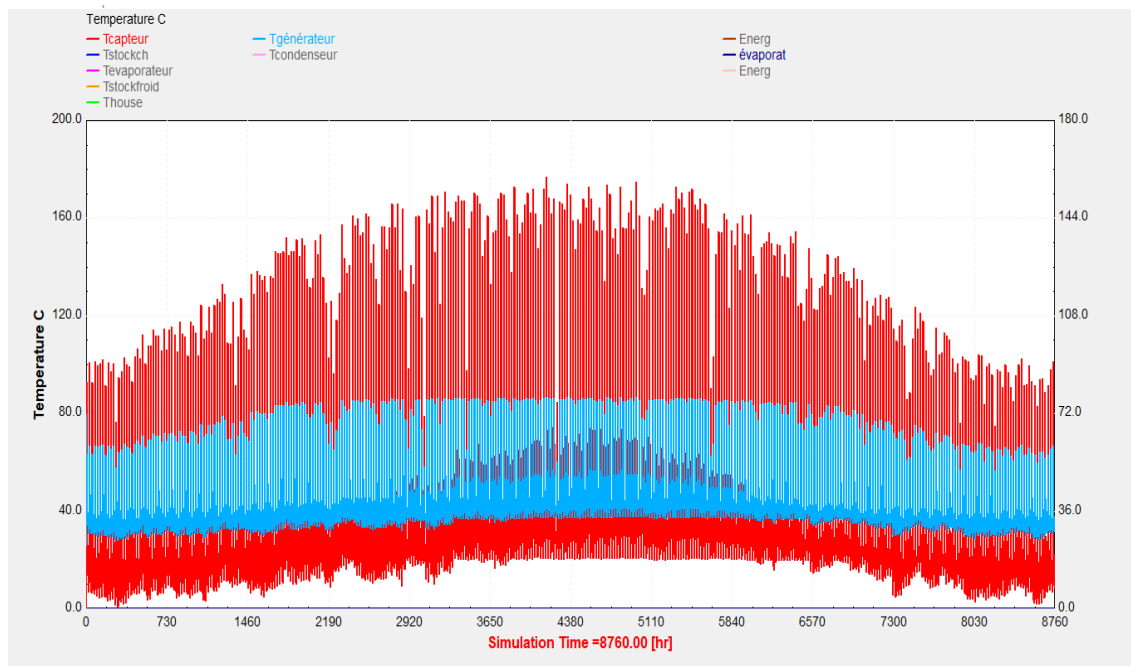


Figure 9. Influence of collector area on collector and generator temperature for a collector area of 90m²

5.4 Effect of collector slope coefficient on collector, generator and storage tank temperatures

It can be seen that as the collector's slope coefficient increases,

the collector temperature decreases, which leads to a decrease in the temperature of the storage tank as well as the generator (Figures 10-12).

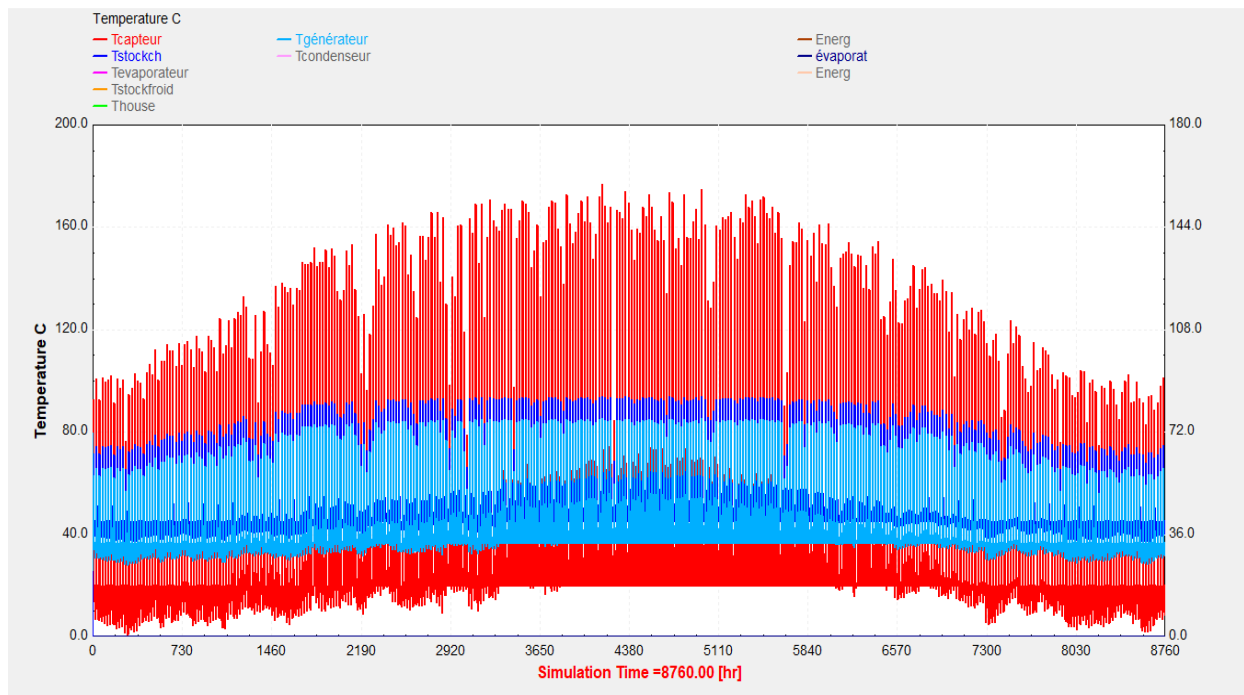


Figure 10. Collector, generator and storage tank temperatures at a slope coefficient of 18

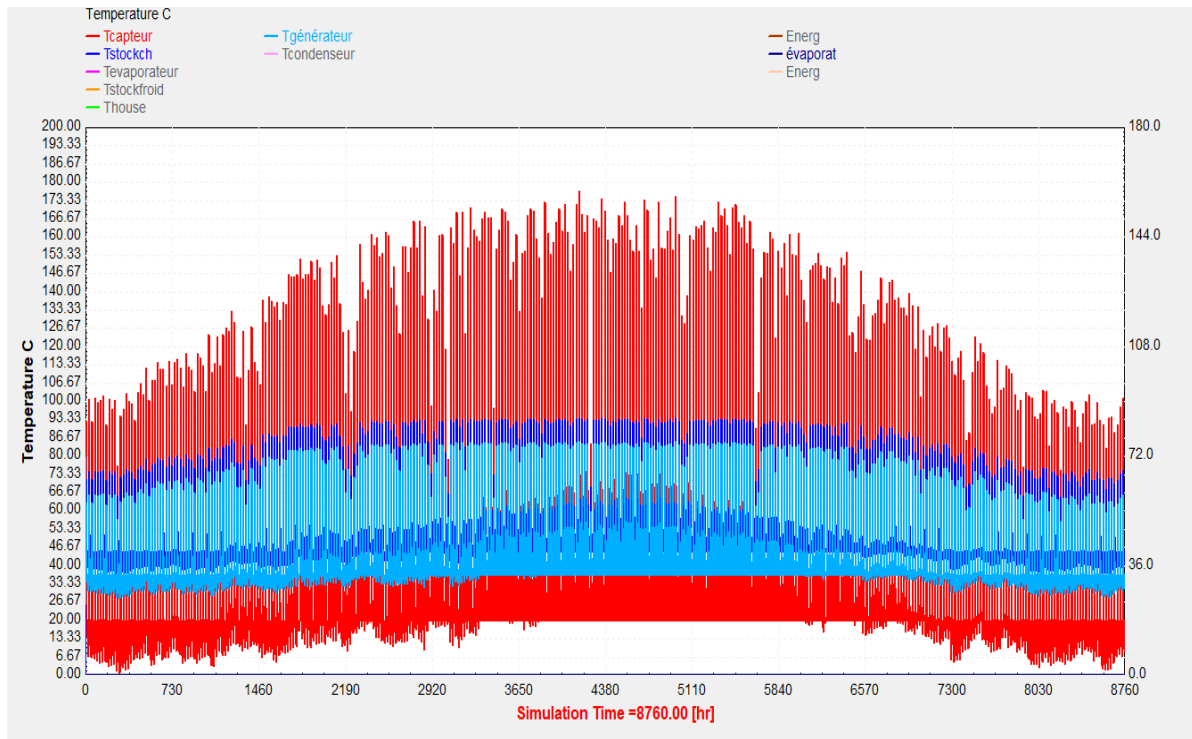


Figure 11. Collector, generator and storage tank temperatures at a slope coefficient of 22

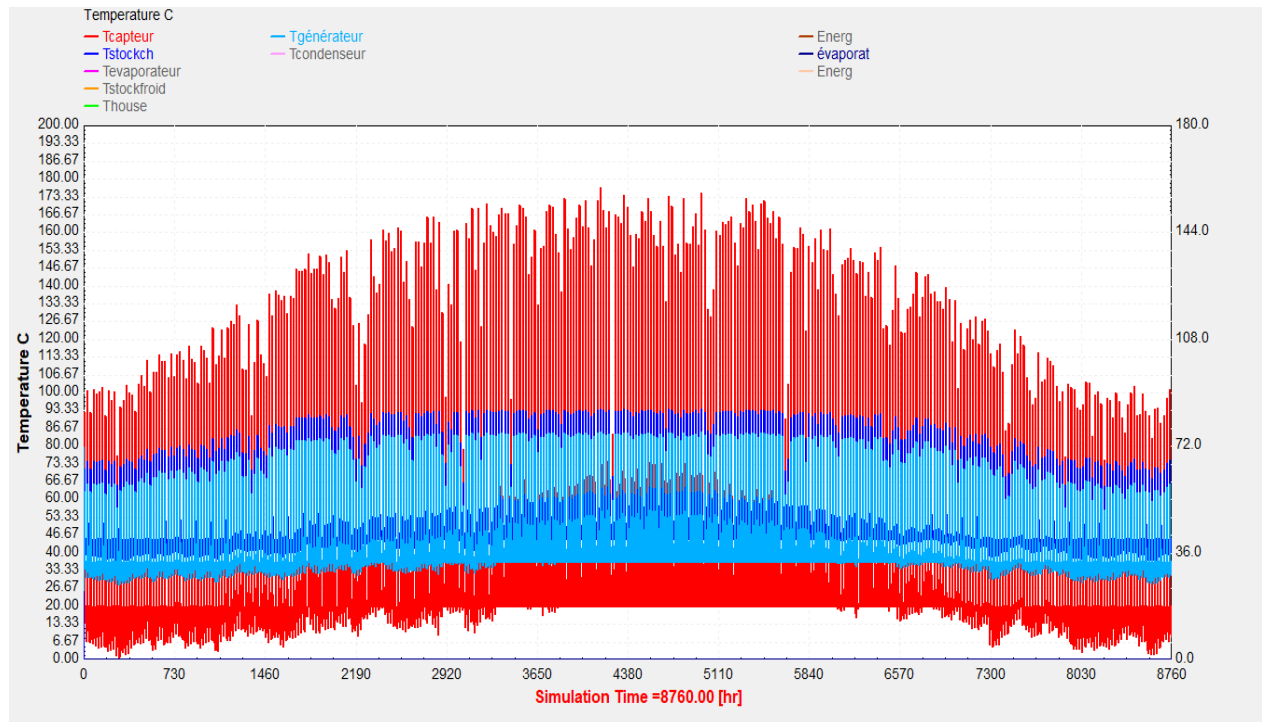


Figure 12. Collector, generator and storage tank temperatures at a slope coefficient of 25

6. RESULTS VALIDATION

The simulation results plotted the following COP variations: According to these graphs, we can see that the COP increases with the increasing temperature of the generator until it reaches a maximum value, while it decreases with the increasing temperature of the condenser (Figure 13). In fact, the obtained results by our model are in good agreement with those of

Romero et al (Romero and al 2001). For the couple water - lithium bromide. However, the differences, which remain negligible, are due to the assumptions used as a basis for these calculations and the choice of equations for the calculation of thermodynamic properties of the used fluid couples.

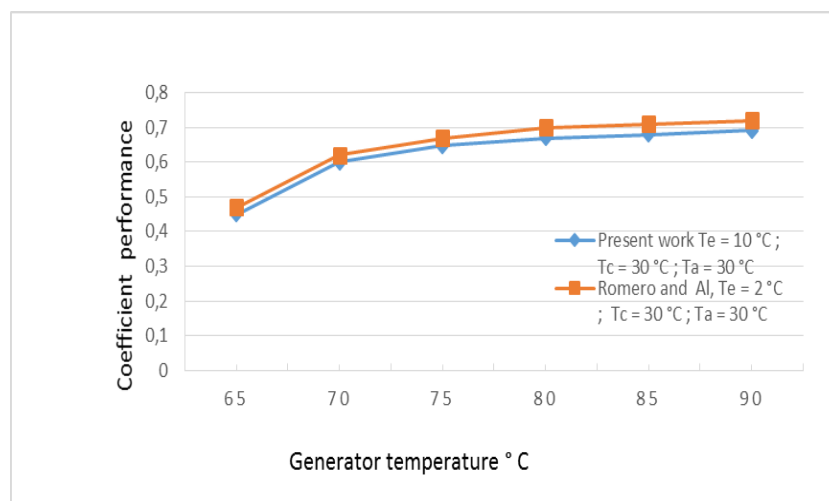


Figure 13. COP comparison of the present investigation with the data of Romero and al (Romero and al 2001)

7. CONCLUSION

The studied simulations during this work allowed us to draw the following conclusions:

1. The obtained results show that the C.O.P of the absorption system is about 0.6254 which corresponds to an operating temperature below 80 ° C ensuring the operation of the installation in the chosen site;
2. The temperature balance of the building in relation to the sunrise and sunset and stabilization by the evaporator effect. The evaporator temperature is composed of two regimes, transient which presents the absorption of heat and permanent which presents the total vaporization ;
3. By varying the collector area, the collector and generator temperatures vary between 80 °C and 160 °C respectively. This shows that the machine is working properly ;
4. The increase of the slope coefficient of the collector leads to a decrease of the outlet temperature of the storage tank as well as that of the generator. This leads us to propose a good orientation of the collector in order to ensure the increase of the output temperature and consequently the needs of uses ;
5. The climatic conditions of Bechar site allowed us to have interesting results, an energy saving (free energy), without forgetting the contribution to the environmental protection ;
6. The COP is proportional to the temperature of the generator and inversely proportional to the temperature of the condenser.

Nomenclature

$\dot{m}$	Mass flow, kg.s ⁻¹
C_p	Heat capacity, J.kg ⁻¹ .K ⁻¹
T	Temperature, °C
M	Mass, kg
Φ	Heat flow, W
S	Surface, m ²
U	Internal energy, J
eau_{sol}	Water in collector circuit
eau_{ballon}	Water in the tank
Air_{Bat}	Air in the building
e	Input
s	Output
$capt$	Sensor
gen	Generator
$cond$	Condenser
$evap$	Evaporator
ext	Exterior
bal_{ch}	Hot water tank
bal_{fr}	Cold water tank

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