

Modeling Of Solar Absorption Air Conditioning System Applied To An Office Building In Morocco.

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Abstract: this paper consists of modeling a solar absorption air conditioning system for an office building in Morocco to replace conventional air conditioning systems whose power is already determined. This study will allow us to estimate the cooling production, the electricity consumption, the coefficient of efficiency (COP) and the overall performance of the solar air conditioning installation according to the external demands, in order to optimize and size the required installation. Dynamic modeling seems inevitable so to be able to take into account as precisely as possible all the physical phenomena related to our problem.

Index Terms: Modeling, Solar Absorption, Air Conditioning, Office Building, Dynamic Modeling, Cooling production, Electricity consumption, COP.

1. INTRODUCTION

Heating and air conditioning installations are part of the fields most concerned to achieve the objectives announced by the Moroccan government, which consist in achieving a primary energy saving of about 12% to 15% by 2020. Indeed, by considering the improvement in the standard of living and the decrease in the prices of heating and air conditioning equipment, the requirements of buildings occupants with regard to comfort have changed significantly and have become more and more demanding. This growth naturally leads to a significant increase in electricity consumption since the technologies used are energy-intensive. However, Morocco remains a country facing the virtual absence of fossil energy resources and a heavy dependence on imports, which cover more than 95% of energy needs [1]. In this difficult energy context, the use of solar energy in combination with solar air conditioning systems is one of the most attractive alternatives to conventional air conditioning systems. To this purpose, the choice of technology was made on one of the most widespread applications of solar air conditioning systems: absorption, this technology has the advantage to ensure continuous operation, a better COP compared to other technologies like adsorption, requires little external work and are more dominant in the solar technology market.

2 ABSORPTION MACHINE

LIQUID ABSORPTION REFRIGERATION MACHINES ALLOW THE PRODUCTION OF COLD WITH HEAT, DUE TO THE ABILITY OF CERTAIN LIQUIDS TO ABSORB AND DESORB A VAPOR AND THE FACT THAT THE SOLUBILITY OF THIS VAPOR IN THE LIQUID DEPENDS ON

TEMPERATURE.

Therefore, these refrigeration machines uses as working fluid a binary mixture, one of the components is very volatile compared to the other, which constitutes the refrigerant fluid.

We mainly meet two couples:

- Water + Lithium bromide: water being the refrigerant;
- Ammonia + Water: ammonia being the refrigerant.

For room air conditioning, the H₂O/LiBr is the most appropriate and the NH₃/H₂O is generally used to make negative cold [2].

2.1 Description of an Absorption Refrigerant Cycle

This machine is composed of four main equipment (a generator, a condenser, an evaporator, an absorber) and which can be assimilated to heat exchangers, two regulators, a pump and sometimes an inter-solutions heat exchanger. The points representing the various states of the solution (refrigerant-absorbent) are shown in the following diagram:

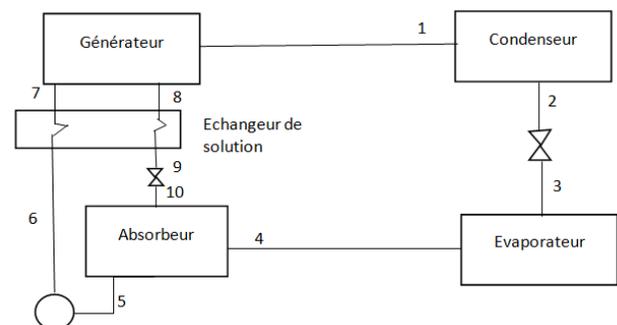


Fig.1: Schematic diagram of an absorption machine

The solar energy captured by the collectors is used to heat a coolant fluid, which fed the storage tank. The energy it has received is then transferred to the solution contained in the generator via the heat exchanger. Therefore, the lithium bromide solution that enters the generator in the diluted state (state7) is concentrated by vaporizing a quantity of water. The steam formed leaves the generator in the superheated state (state1) and cools down until condensation is reached by transferring the heat to the cooling water. The saturated condensate (state 2) undergoes iso-enthalpic expansion where its pressure changes from that prevailing in the condenser and generator to that of the evaporator and absorber (state 3). The low pressure in the evaporator allows

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the water to vaporize at low temperature by taking the energy necessary for its vaporization from the water to be cooled. The saturated steam then leaves the evaporator (state 4) and feeds the absorber where the concentrated lithium bromide solution flows, which dilutes by absorbing the water vapor until saturation. This is a physical absorption that is an exothermic operation, hence the need for cooling to reach the desired concentration of saturation at the output of the absorber (state 5). The diluted solution at low pressure is pumped to the pressure required for regeneration (state 6). This transformation is supposed to be isentropic since we operate at low pressures. Concerning the cycle of the lithium bromide solution (5-6-7-8-9-10). It makes sense to take advantage of the high temperature of the concentrated solution leaving the generator to heat the diluted solution coming from the absorber. Since otherwise this heat will only be dissipated, which will unnecessarily require a larger exchange surface in the absorber and a larger size of the cooling tower and will penalize the COP of the machine. This is why an economizer have to be installed in the circuit (generator-absorber) [3].

2.2 Data and Modeling Hypotheses

The knowledge of the input and output temperature of the cold water that we are trying to produce is essential to deduce the temperature and therefore the prevailing pressure in the evaporator. The regimes commonly used for chilled water are:

The 7-12°C mode: which is most often used in office spaces, as it has the advantage of requiring a lower investment cost compared to the 12-16°C regime. As a counterpart, such a regime will penalize the COP of the refrigeration machine, as will be demonstrated later.

The 12-16 °C mode: If the water loop operates at a higher temperature, the COP of the refrigeration machine will be improved and unnecessary condensation of ambient air humidity will be avoided. As a result, the operating cost will be lower. An oversizing of the cold battery will certainly be necessary, and therefore an additional investment cost. [4]

In our case, the objective is to replace the existing conventional water chillers with an absorption machine, so we will keep the same chilled water distribution system installed in our office building. However, we thought it would be useful to make a comparative study of the results obtained using the two regimes in order to make tangible the influence of the chilled water regime on the machine's performance.

We propose in this section a phenomenological model of absorption machine that will allow us to estimate the COP from the input temperatures of the main components, the energy efficiency of the absorption machine and the temperatures at the output of these components. Of course, for the application of the principles of thermodynamics, we have based on the following simplifying hypotheses [5], [6], [7]:

- The temperatures in the exchangers (generator, condenser, evaporator and absorber) are assumed uniform over the entire volume considered.
- In points: 1-2-3 and 7, the refrigerant is in its pure state. It is saturated in the liquid state (point 2) then in the vapor state (point 4).
- There is no pressure change in the piping except at the regulator and solution pump. In points, 7 and 8 the solution is saturated.

- Triggers are assumed isenthalpic.
- Heat exchanges with the environment and pressure drops are assumed negligible.

3 THERMODYNAMIC ANALYSIS OF THE CYCLE

3.1 Mass and Enthalpy Balance

The objective of this analysis is to estimate the quantities of energy exchanged of each component and to evaluate the COP and the energy efficiency of the installation, according to various solicitations.

For the following, we note:

- \dot{m}_i : Mass flow rate of component i (kg/s)
- W : Pump power (watt)
- Q_i : Thermal power of the equipment i (watt)
- h_i : Enthalpy of component i (kJ/kg)
- v_i : Masse volume (kg/m³)
- P_h : High pressure (bar)
- P_b : Low pressure (bar)

1. Evaporator analysis

$$\text{Mass balance : } \dot{m}_3 = \dot{m}_4 = \dot{m}_{ef} \quad (1)$$

$$\text{Enthalpic balance : } Q_{ev} = \dot{m}_{ef} \times (h_4 - h_3) \quad (2)$$

$$\text{So : } \dot{m}_{ef} = \frac{Q_{ev}}{(h_4 - h_3)} \quad (3)$$

2. Absorber analysis

$$\text{Global mass balance : } \dot{m}_4 + \dot{m}_{10} = \dot{m}_5 \quad (4)$$

$$\text{Partial mass balance on LiBr : } X_{10} \times \dot{m}_{10} = X_5 \times \dot{m}_5 \quad \text{Display}$$

From the equations (4) and (5), we conclude:

$$\dot{m}_{10} = \frac{X_5}{X_{10} - X_5} \times \dot{m}_4 \quad (6)$$

$$\text{Enthalpic balance : } \dot{m}_4 \times h_4 + \dot{m}_{10} \times h_{10} = \dot{m}_5 \times h_5 + Q_A \quad \text{Display}$$

3. Generator analysis

$$\text{Mass balance : } \dot{m}_7 = \dot{m}_1 + \dot{m}_8 \quad (8)$$

$$\text{Enthalpic balance : } Q_{gen} + \dot{m}_7 \times h_7 = \dot{m}_8 \times h_8 + \dot{m}_1 \times h_1 \quad (9)$$

4. Condenser analysis

$$\text{Mass balance : } \dot{m}_1 = \dot{m}_2 = \dot{m}_{ef} \quad (10)$$

$$\text{Enthalpic balance } Q_c = \dot{m}_{ef} \times (h_2 - h_1) \quad (11)$$

5. Analysis of the internal exchanger

$$\text{Mass balance : } \dot{m}_7 = \dot{m}_6 / \dot{m}_8 = \dot{m}_9 \quad (12)$$

$$\text{Enthalpic balance : } \dot{m}_7 \times (h_7 - h_6) = \dot{m}_9 \times (h_9 - h_8) \quad (13)$$

6. Analysis of the regulator of the refrigerant

$$\text{Mass balance : } \dot{m}_2 = \dot{m}_3 \quad (14)$$

$$\text{Enthalpic balance : } h_2 = h_3 \quad (15)$$

7. Analysis of the regulator of the solution

Mass balance : $\dot{m}_{10} = \dot{m}_9$ (16)
 Enthalpic balance : $h_{10} = h_9$ (17)

8. Analysis of the solution pump:

Mass balance : $\dot{m}_5 = \dot{m}_6$ (18)
 Enthalpic balance : $W_{pump} = \dot{m}_5 \times (h_6 - h_5) = v_5 \times (P_h - P_b)$
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3.2 Coefficient of performance of the machine:

A good index to quantify the thermal quality of the solar cooling process is the coefficient of performance (COP) defined as:

$$COP_{thermal} = \frac{Q_{ev}}{Q_{gen} + W_{pump}} \quad (20)$$

This coefficient indicates the amount of heat required to produce a cooling unit (or more precisely to remove a quantity of heat). It is closely related to the temperature levels of the machine (high, low, medium). For a chilled water system, the maximum performance according to temperature levels can be calculated as follows:

$$COP_{ideal} = \frac{TC}{TH} \times \frac{TH - TM}{TM - TC} \quad (21)$$

TH : temperature of the hot source in °K ;

- TC : temperature of the cold source in °K ;
- TM : average heat rejection temperature in °K ;

The efficiency of the system related to ideal conditions is expressed as follows:

$$\eta = \frac{COP_{thermal}}{COP_{ideal}} \quad (22)$$

According to the previously established balance equations, the COP as a function of enthalpies can be expressed as follows:

$$COP_{thermal} = \frac{h_4 - h_3}{h_1 + (Fr - 1) \times h_8 + Fr \times (h_6 - h_5 - h_7)} \quad (23)$$

With Fr : recirculation rate = $\frac{\text{Diluted solution flow rate}}{\text{Refrigerant flow rate}}$

4 APPROACH OF THE MODELIZATION:

After establishing the model equations, the next step would be to study the effect of the variation of the various system input parameters on machine performance. This study will allow us to choose the nominal parameters of the machine's operation. The thermodynamic properties of the solution has been taken from the references [8] ,[9].

Data :

- Cooling capacity to be produced by the machine: $Q_e=1.124$ MW
- The efficiency of the solution exchanger is considered: $\epsilon=0.7$
- Chilled water temperature regime: As we explained earlier, we will discuss a comparative study for the two temperature regimes 7/12 and 12/16°C.
- Cooling water temperature regime: 29/35°C. Indeed, the use of a cooling tower limits the return

temperature of the cooling water as long as the bulb temperature of the ambient air cannot be exceeded. Hence the proposed regime.

- The authors[Lasvignottes [6], Grosu [10]] proposed to introduce the notion of heat exchanger pinching, which makes it possible to determine the temperature of each component from its input temperature according to the equations:

$$\begin{cases} T_{gen} = T_{s, gen} - \Delta T_{pin, gen} \\ T_{ev} = T_{s, ev} - \Delta T_{pin, ev} \\ T_{cond} = T_{s, cond} + \Delta T_{pin, cond} \\ T_A = T_{s, ab} + \Delta T_{pin, ab} \end{cases} \quad (24)$$

With, respectively $T_{s,gen}$, $T_{s,ev}$, $T_{s,cond}$ and $T_{s,ab}$, the output temperatures of the generator, he evaporator, the condenser and the absorber.

- The pinch in the exchangers is considered equal to 5°C in the evaporator and condenser ,as for the absorber, we will select a pinch of 4°C and 8°C for the generator [11].

The approach followed in the modelling is explained below:

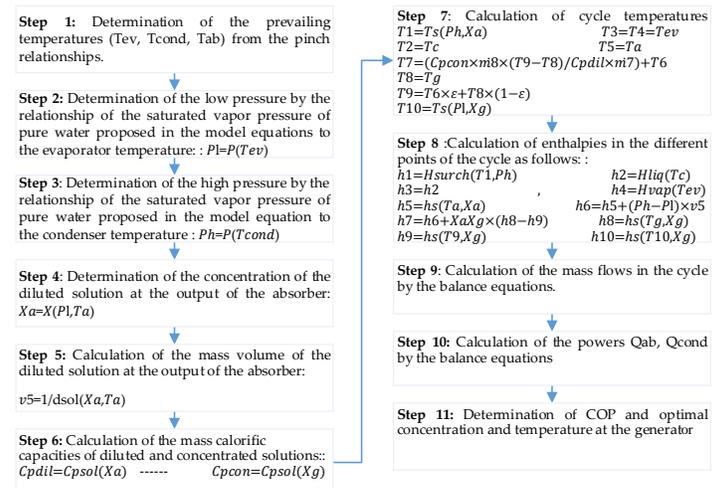


Fig.2: Approach followed in the modelling of absorption machine

5 PARAMETRIC STUDY

5.1 Influence of the variation of concentrations X_g - X_a on the COP

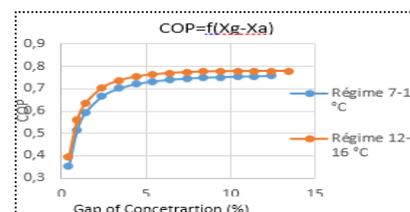


Fig.3: Influence of the variation of concentrations

Xg- Xa on the COP Evolution of the COP. Any increase of concentration at the generator results in an increase in the COP of the machine. However, this evolution of the COP begins to disappear as the concentration gap increases, and at some point, it becomes almost constant. This result shows us that it is not necessary to increase the concentration in the generator indefinitely, especially since another constraint arises, namely the crystallization of the solution, which is more likely to occur at high concentrations.

5.2 COMPARISON OF THE TWO REGIMES

- Influence of generator temperature on COP

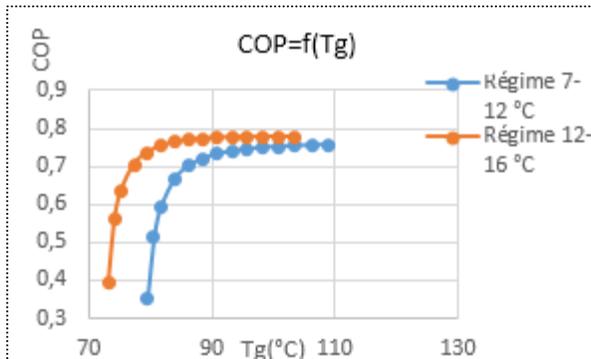


Fig.4: Influence of generator temperature on

COP THE 12-16°C OPERATING MODE ALLOWS THE SAME COP VALUES TO BE ACHIEVED FOR LOWER GENERATOR TEMPERATURES COMPARED TO THE 7-12°C OPERATING MODE. THEREFORE, THE ENERGY REQUIRED AT THE GENERATOR LEVEL WILL BE MINIMAL FOR THE 12-16°C REGIME. WHEREAS FOR THE 7-12°C REGIME THE COP STABILIZES FROM A TEMPERATURE OF 96°C, WHICH CORRESPONDS TO A COP OF 0.74, WHILE FOR THE 12-16°C REGIME IT BECOMES STABLE AT A TEMPERATURE OF 86°C CORRESPONDING TO A COP OF 0.77. HOWEVER, IT WILL BE NECESSARY TO ENSURE THAT THERE IS

NO RISK OF CRYSTALLIZATION.

- Influence of power variation at the generator level on COP

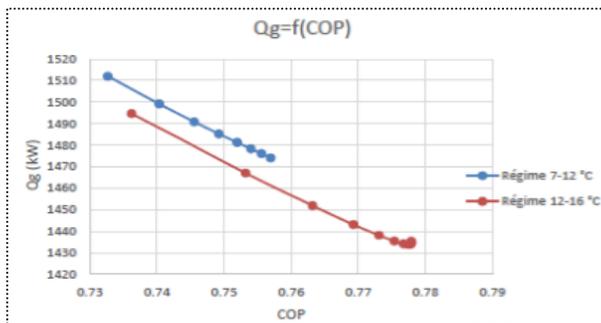
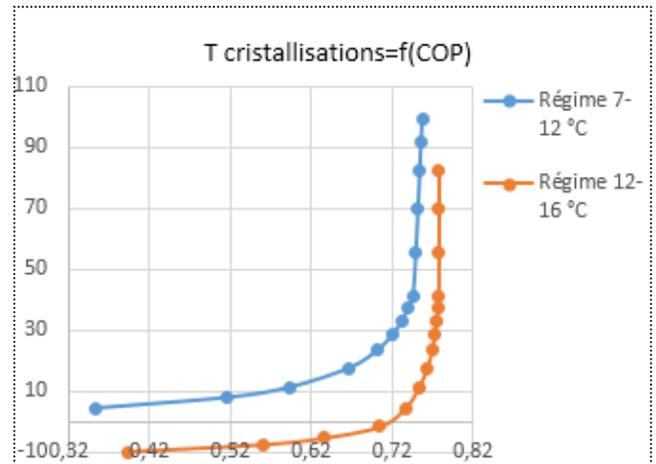


Fig.5: Influence of power variation at the generator level on COP

A better COP is the one corresponding to the operation using a minimum power for the same performance. From the curve in the following figure, we can see that for the same COP, the energy required at the generator level for the 12-16mode is lower than that required for the 7-12 mode.

- Influence of the crystallization temperature on the COP



Obtaining a better COP is penalized by the increased risk of crystallization. For the 12-16 ° C regime, the risks of crystallization are lower compared to the 7-12 ° C regime. We observe from the figures above that:

$$\begin{cases} COP = 0.74 \\ T_{gen} = 96^{\circ}C \\ T_{cri} = 55^{\circ}C \end{cases} \text{ Pour le régime } 7 - 12^{\circ}C$$

$$\begin{cases} COP = 0.77 \\ T_{gen} = 86^{\circ}C \\ T_{cri} = 23^{\circ}C \end{cases} \text{ Pour le régime } 12 - 16^{\circ}C$$

5.3 Determination of cycle characteristics for both regimes

The choice of operating parameters is made to obtain a better COP, while taking into account the problem of crystallization. For both regimes, we have:

- $Q_{ev}=1124 \text{ kW}$
- Cooling water regime: 29/35 °C, hence,

Temperature in the absorber: 4°C pinch, 2°C gap in cooling water temperature
Eq(24) $TA=35^{\circ}C$

Temperature in the condenser: 5°C pinch, 4°C gap in cooling water temperature.
Eq(24) $T_{cond}=40^{\circ}C$

Pressure in the condenser/generator (High Pressure): The condenser and the generator operate at the same pressure. It corresponds to the pressure of the saturated steam at the prevailing temperature in the condenser.

We find: $Ph=73,26 \text{ mbar}$

The other parameters are determined for each temperature regime separately.

- Chilled water regime 7-12 ° C

The parameters that have been chosen according to the previously studied graphs a Gap of Concentration (%)

- Mass title of the diluted solution: 0.576
- Mass title of the concentrated solution: 0.61
- Generator temperature: 86.14 ° C
- COP: 0.7
- Recirculation rate: 17.9

Calculation of the powers of the various components (Annex 1: Thermodynamic characteristics of the refrigerating cycle for the 7/12 ° C regime)

$$\left\{ \begin{array}{l} Q_{ev} = 1124 \text{ kW} \\ Q_{gen} = 1567.82 \text{ kW} \\ Q_a = 1541.65 \text{ kW} \\ Q_{cond} = 1184.44 \text{ kW} \\ W_{pompe} = 34.28 \text{ kW} \\ \text{Required cooling water flow : } 108.69 \text{ kg/s} \\ \text{Required coolant flow rate : } 40.20 \text{ kg/s} \end{array} \right.$$

- Chilled water regime 12-16 ° C

The parameters that have been chosen according to the previously studied graphs are:

- Mass title of the diluted solution: 0.546
- Mass title of the concentrated solution: 0.61
- Generator temperature: 83.86 ° C
- COP: 0.77
- Recirculation rate: 9.5

Calculation of the powers of the various components (Annex 1: Thermodynamic characteristics of the refrigerating cycle for the 12/16 ° C regime)

$$\left\{ \begin{array}{l} Q_{ev} = 1124 \text{ kW} \\ Q_{g\acute{e}n} = 1443.14 \text{ kW} \\ Q_a = 1405.35 \text{ kW} \\ Q_{cond} = 1179.72 \text{ kW} \\ W_{pompe} = 17.92 \text{ kW} \\ \text{D\acute{e}bit d'eau de refroidissement requis : } 103.07 \text{ kg/s} \\ \text{D\acute{e}bit de fluide caloporteur requis : } 37 \text{ kg/s} \end{array} \right.$$

- Comparison of the two regimes

Regime	7-12 ° C	12-16 ° C
Q_{gen} (kW)	1567,8	1443.1
COP	0.7	0.77
$Q_a + Q_c$ (kW)	2726,1	2585,1
W_{pompe} (kW)	34	18

Table1: Comparison of the powers of the various components of the refrigerating machine for the two chilled water regimes 7-12 ° C and 12-16 ° C.

The 12-16 ° C regime provides a COP higher than the 7-12 ° C regime, and requires less power at the generator, so a smaller area of solar collectors. Moreover the work of the solution pump for the 12-16 ° C regime, is lower than that of the 7-12 °

C regime. It turns out that working at a higher temperature level at the evaporator provides better performance. It remains to know the influence of the increase of the temperature on the surfaces of exchanges of the different equipment of the machine. We will then begin a comparative study of the two regimes in this direction.

4. CONCLUSION

This study was the subject of a modeling of a water-bromide lithium bromide absorption machine, which allowed us to choose the optimal operating parameters that will serve as the basic data for the rest of our thesis. Regarding the performance of the machine, which is influenced by the evaporation temperature of the refrigerant, we carried out a comparative study of two chilled water temperature regimes in order to quantify the gain that could be generated by changing the regime used in the existing installation. The conclusion was that any increase in the temperature of the chilled water regime would only be beneficial to the machine's performance. Indeed, the use of the 12-16 ° C instead of 7-12 ° C regime will result in an increase in the COP of the machine for the same cooling capacity of about 10%, which leads to a decrease in the energy required at the generator level and therefore in the collector surface area.

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