Effect of Pressure on Improving COP of a Refrigeration Installation Solar To Absorption

M. Benramdane(1), MEA. Ghernaout(1),
(1)Department of Mechanical Engineering.
ETAPlaboratory, Faculty of Technology, University of
Tlemcen, Tlemcen13000, Algeria

S. Abboudi(2)
(2)IRTES-M3M Laboratory, UTBM, Site of Sevenans,
90010, Belfort Cedex, France.

Abstract—the use of renewable energy creates more of interest in the world. Disruptions in oil prices, gas and environmental problems have led many countries to focus on renewable clean energy such as solar energy. However, cold production from this energy is possible for our country.

Among the various systems used to this, the absorption refrigeration machines seem to be very promising in this respect. A request for increased comfort and particularly high temperatures were led to strong growth of air conditioning in developed countries.

The objective of this work is to study a solar refrigeration double effect (two boilers, condenser, evaporator and two absorbers) operating with simple absorption panels, to see the influence of thermodynamic parameters in particular pressure on the improvement coefficient of performance (COP) and show that it produces the cold with interesting performances. It does not require distillation column and has the following advantages: possible operation from temperature in the boilers 70°C, possibility of using solar collectors cheaper which affects the overall cost of the installation.

Keywords—Cold Solar, absorption refrigeration, COP, improvement.

I. INTRODUCTION

The thermal energy required to operate these technologies comes from solar radiation. The solar system converted the solar thermal radiation through a capture by greenhouse effect is called solar radiation sensor.

Solar systems for refrigeration receive energy and convert it into coldstore and the rest is for use during the nights and the bad weather periods.

Generally, they are three necessary elements: solar, energy storage system, absorption machine[1].

In order to see the influence of the pressure on the improvement of the coefficient of performance. The configuration considered is that of an absorption refrigeration machine with double effect. It works with three pressures.

High-pressure of bouilleur 2 and condenser low pressure of the evaporator and the absorber 1.

The use of a second bouilleur 1 powered by binary solution can raise the third pressure absorber 2 then be in an intermediate pressure between the low pressure evaporator and the high pressure regenerator 2 (Figure 2).

II. ABSORPTION SYSTEM

Refrigeration absorption phenomenon that uses binary solutions absorb vapor and other solutions within this defined temperatures. It then calls absorption, the fixation of a substance by the entire volume of the absorbent material.

We know that the vapor of a pure substance can be absorbed by the body in the liquid state than in the case where the liquid temperature is lower than that of the vapor, unlike the pure substances, solutions having a remarkable property of absorbing the vapor of the liquid solution, different composition even if the liquid temperature is higher than the vapor [2].

Liquid absorption machines operate using the power of certain liquid absorbing (exothermic reaction) and desorption (endothemic reaction) vapor. An absorption system comprises a first assembly (condenser, expansion valve, evaporator) wherein the refrigerant is called refrigerant [1].

Two couples are mainly used:

- Water / Lithium Bromide (H₂O/LiBr)
- Ammonia / Water (NH₃/H₂O)

We can express the composition of the solution in the most volatile component (refrigerant), or the absorbent (absorbent).

- Diluted solution (absorbing) or rich solution (refrigerant).
- Concentrated (absorbing) or poor solution (refrigerant).
III. INSTALLATION

The absorption machine is, for us, the heart of the whole system of solar cooling. It therefore seemed interesting to see how these machines are modeled. In this regard, there are generally two approaches [3]: The first is based on a phenomenological description of each component of the machine. This approach is based on different energy balances to couple the four components of the machine. The second which is probably the most used is performance evaluation by an empirical model which is generally smoothing curves based on manufacturer's data. Indeed, these are tests that establish a simple correlation customary for the COP and the cooling capacity of the machine. Correlations are thus valid for the range of tests and tested the machine model studied.

To establish the heat balance of any component and to size the heat transfer surfaces that are associated with, it is necessary to know precisely the enthalpy of working fluid in the liquid and gaseous states as a function of temperature and concentration. Among the existing models, we have chosen the one proposed by M. Feidt [4] which combines the Gibbs free energy for the thermal properties and the equations that calculate the bubble point and the dew point of the mixture. This method combines the advantages of both and eliminates the need for iterations in order to have conditions of equilibrium phases.

The knowledge of climatic and geographical characteristics of the studied region is a very important factor in the study. These elements are variable, but we can estimate a monthly average [5]. After determining different mass and energy equations of the different elements of the installation, the table below presents the different equations for mass flow rates at different points of the installation.

### TABLE I. DETERMINATIONS

<table>
<thead>
<tr>
<th>elements</th>
<th>mass flow (kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$\dot{m}_1$</td>
</tr>
<tr>
<td>2</td>
<td>$\dot{m}_2 = \dot{m}_1$</td>
</tr>
<tr>
<td>3</td>
<td>$\dot{m}_3 = \dot{m}_1$</td>
</tr>
<tr>
<td>4</td>
<td>$\dot{m}_4 = \dot{m}_1$</td>
</tr>
<tr>
<td>5</td>
<td>$\dot{m}_5 = \dot{m}<em>1 \left(1 + \frac{(x_4 - x_5)}{(x_5 - x</em>{10})}\right)$</td>
</tr>
<tr>
<td>6</td>
<td>$\dot{m}_6 = \dot{m}_5$</td>
</tr>
<tr>
<td>7</td>
<td>$\dot{m}_7 = \dot{m}_5$</td>
</tr>
<tr>
<td>8</td>
<td>$\dot{m}<em>8 = \dot{m}</em>{10}$</td>
</tr>
<tr>
<td>9</td>
<td>$\dot{m}<em>9 = \dot{m}</em>{10}$</td>
</tr>
<tr>
<td>10</td>
<td>$\dot{m}_{10} = \dot{m}<em>1 \left(\frac{x_4 - x_5}{x_5 - x</em>{10}}\right)$</td>
</tr>
<tr>
<td>11</td>
<td>$\dot{m}_{11} = \dot{m}<em>1 \left(1 - \frac{(x_4 - x_5)}{(x_5 - x</em>{10})}\right)$</td>
</tr>
<tr>
<td>12</td>
<td>$\dot{m}_{12} = \frac{\dot{m}_7}{2} = \frac{\dot{m}<em>1}{2} \left(1 + \frac{(x_4 - x_5)}{(x_5 - x</em>{10})}\right)$</td>
</tr>
<tr>
<td>13</td>
<td>$\dot{m}<em>{13} = \dot{m}<em>1 \left(1 + \frac{(x_1 - x</em>{15})}{(x</em>{15} - x_{16})}\right)$</td>
</tr>
<tr>
<td>14</td>
<td>$\dot{m}<em>{14} = \dot{m}</em>{13}$</td>
</tr>
<tr>
<td>15</td>
<td>$\dot{m}<em>{15} = \dot{m}</em>{13}$</td>
</tr>
<tr>
<td>16</td>
<td>$\dot{m}<em>{16} = \dot{m}<em>1 \left(\frac{x_1 - x</em>{15}}{(x</em>{15} - x_{16})}\right)$</td>
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<tr>
<td>17</td>
<td>$\dot{m}<em>{17} = \dot{m}</em>{16}$</td>
</tr>
<tr>
<td>18</td>
<td>$\dot{m}<em>{18} = \dot{m}</em>{16}$</td>
</tr>
<tr>
<td>19</td>
<td>$\dot{m}_{19} = \frac{\dot{m}_7}{2} = \frac{\dot{m}<em>1}{2} \left(1 + \frac{(x_4 - x_5)}{(x_5 - x</em>{10})}\right)$</td>
</tr>
</tbody>
</table>
IV. THERMODYNAMIC STUDY OF THE SYSTEM STUDIED

For the application of the principles of thermodynamics on a real cycle, conditions and assumptions were used:
1. Temperatures in parts of the plant (boiler, condenser, absorber and evaporator) are assumed uniform throughout the volume in question.
2. Solution rich in refrigerant at the outlet of the absorber is saturated at the temperature and the concentration in the absorber liquid. Likewise, the weak solution leaving the refrigerant generator is connected by a balance of pressure and temperature relationship of the concentration generator.
3. The coolant leaving the boiler is taken as saturated vapor at the temperature and corresponding pressure.
4. The coolant leaving the condenser is taken as saturated liquid at the same temperature and pressure. The refrigerant at the outlet of the evaporator is the form of saturated steam at the temperature and low pressure of the evaporator. The isenthalpic expansions are assumed.
5. Heat exchange with the environment and losses are assumed negligible.

Calculations are based on the determination of the respective enthalpies of the liquid phase and vapor phase from the analytical expression of the Gibbs energy \[6\], knowing the pressure, temperature and the concentration of the solution. We must also determine the quantities of vapor-liquid equilibrium of binary pair ammonia-water from the Peng-Robinson equation \[7\] and the interaction coefficient \( K_{ij} \) characterizing the mixing torque.

For the determination of various system parameters, enthalpies (\( H_i \)) and titles (\( X_i \)), there are two methods: either from empirical equations or from the diagram of Meckel and Oldham. In our case, we used diagrams for the determination of these parameters.

Our work is spent on a pressure range that varies from 2 to 20 bars.

The choice of a high pressure of 20 bars, for this case, leads us to a title \( X \) extends to 1.

V. COEFFICIENT OF PERFORMANCE

\[
\text{COP} = \frac{Q_{\text{evap}}}{Q_{\text{bou1}} + Q_{\text{bou2}} + W_{p1} + W_{p2}} \quad (1)
\]

\[
\text{COP} = \frac{\dot{m}_3 (h_4 - h_3)}{\dot{m}_8 (h_8 - h_5) + \dot{m}_1 (h_1 - h_0) + \dot{m}_{12} (h_{12} - h_{11})}
\quad (2)
\]

VI. SIMULATION

The simulation is based on the heat balance for the different phases of the absorption cycle \[6\]: Our simulation is made with the aim of seeing the variation of COP depending on the different temperature levels and the elements of the installation with constant pressure absorption.

Fig 3: Evolution of COP on the temperature variation of boiler 2 (T1).

Fig 4: Evolution of COP on the temperature variation of boiler 1 (T11).

Fig 5: Evolution of COP on the temperature variation of the absorber 1 (T5).
VII. INTERPRETATION OF RESULTS

It is important to choose the right temperatures and pressures of running an absorption machine (NH₃/H₂O) at boiler, evaporator and absorber.

Figure [3]

It is noted that the variation of COP is inversely proportional with the temperature at the boiler 2: The COP decreases with increasing temperatures of the boiler 2 (T₁) in the temperature range 70 °C ≤ T₁ ≤ 88 °C. In this interval, it is noted that decreases with the pressure decrease.

Figure [4]

It is noted that the variation of COP is inversely proportional with the temperature at the boiler 1: The COP decreases with increasing temperatures boiler 1 (T₁₁) in the same temperature range. We note also that the COP decreases with the pressure decrease.

Figure [5]

It is noted that the variation of the COP is proportional with the temperature at the absorber 1: the COP increases with increasing temperatures absorber1 (T₅) in the temperature range 30°C ≤ T₅ ≤ 48°C and in this interval it is noted that the variation of the COP is inversely proportional with pressure: COP decreases with increasing pressure.

Figure [6]

It is noted that the variation of COP is proportional with the temperature at the evaporator: the COP increases with increasing temperatures of the evaporator (T₄) in the temperature range -20°C ≤ T₄ ≤ 0°C and decreases with increasing pressure.

VIII. GENERAL CONCLUSION

In order to improve COP and reduced cost for solar absorption refrigeration, this work is devoted to geometric changes to an absorption chiller (dual engine effects constitute two boilers, condenser, evaporator and two absorbers) as the latter can operate with solar panels plans. This study allowed us to analyze the absorption refrigeration systems coupled with solar energy. Thermodynamic analysis of the refrigeration cycle with double effect absorption running torque (NH₃/H₂O) showed that the boiler temperature is inversely proportional to the coefficient of performance of the installation the latter more(COP) with the proportional pressure level of boilers. On the other hand, the temperature of evaporator and absorber are in proportion with the COP of the installation. The COP is very important for low pressure at the evaporator.

The geometric modifications we made on this system (number of boilers) does not require distillation column. Their operation is possible from temperatures which vary between 70°C. The possibility of using solar collectors, less expensive and available in the market, allows us to reduces the overall cost of this type of facilities.
### NOMOCATURE

<table>
<thead>
<tr>
<th>Nomenclatures</th>
<th>Désignations</th>
<th>Unité</th>
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<tbody>
<tr>
<td>T</td>
<td>Température</td>
<td>K</td>
</tr>
<tr>
<td>P</td>
<td>Pression</td>
<td>Pa</td>
</tr>
<tr>
<td>X</td>
<td>Titre du frigorigène ou de l’absorbant dans la solution</td>
<td>%</td>
</tr>
<tr>
<td>Q</td>
<td>Puissance</td>
<td>Kw</td>
</tr>
<tr>
<td>Qe</td>
<td>Puissance frigorifique de l’évaporateur</td>
<td>Kw</td>
</tr>
<tr>
<td>Qb</td>
<td>Puissance frigorifique de bouilleur</td>
<td>Kw</td>
</tr>
<tr>
<td>Qc</td>
<td>Puissance frigorifique de condenseur</td>
<td>Kw</td>
</tr>
<tr>
<td>Wp</td>
<td>Puissance de la pompe</td>
<td>Kw</td>
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<tr>
<td>H</td>
<td>Enthalpie</td>
<td>KJ/kg</td>
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<tr>
<td>m</td>
<td>Débit massique</td>
<td>Kg/s</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient de performance</td>
<td>-</td>
</tr>
</tbody>
</table>

### REFERENCE

1. J. Castaing-lavignottes2001; http://jc.castaing.free.fr/jc.castaing@free.fr.