Exergetic Optimization of Absorption Chiller Single Stage H₂O-NH₃ by Experiments Design Method

Mohamed Izzedine Serge Adjibade¹, Kokouvi Edem N’Tsoukpoe², Ababacar Thiam¹, Christophe Awanto³ and Dorothé Azilinon¹

1. Laboratoire d’Énergétique Appliquée (LEA), Ecole Supérieure Polytechnique, Université Cheikh Anta Diop (UCAD), BP 5085 Dakar-Fann, Sénégal
2. Laboratoire Energie Solaire et Economie d’Energie (LESEE), Département Génie Electrique, Énergétique et Industriel, Institut International d’Ingénierie de l’Eau et de l’Environnement (2iE), 01 BP 594 Ouagadougou 01, Burkina Faso
3. Laboratoire d’Énergétique et de Mécanique Appliquée (LEMA), Ecole Polytechnique d’Abomey-Calavi, Université d’Abomey-Calavi (UAC), 01 BP 2009 Cotonou, Bénin

Abstract: Single stage absorption chillers using H₂O-NH₃ have received increasing research interest in recent years, in order to make them competitive with conventional refrigeration machines. This work presents a study on the performance of such tri-thermal machines, used for negative temperature refrigeration. The objective is to determine the values of the system operating temperatures that minimize the irreversible losses in the various heat exchangers. To do this, the overall exergy efficiency of the system has been expressed as a function of the various operating temperatures. This objective function is to be maximized with experimental design method. The normal probability plot of the residual indicates that the random errors for the process are drawn from approximately normal distributions. From the results, with exergy efficiency greater than 0.4, two operating modes are presented which can be with condensation temperatures below 32 °C and above 38 °C.

Key words: Absorption chiller, exergy, thermodynamic irreversibility, exergetic efficiency, water-ammonia.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Evaporation temperature</td>
</tr>
<tr>
<td>B</td>
<td>Heat source temperature</td>
</tr>
<tr>
<td>C</td>
<td>Condensation temperature</td>
</tr>
<tr>
<td>h</td>
<td>Specific enthalpy (kJ·kg⁻¹)</td>
</tr>
<tr>
<td>Ir</td>
<td>Irreversibility (kW)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow (kg·min⁻¹)</td>
</tr>
<tr>
<td>P</td>
<td>Pressure (kPa)</td>
</tr>
<tr>
<td>Ṅ</td>
<td>Flow of heat exchanged (kW)</td>
</tr>
<tr>
<td>s</td>
<td>Specific entropy (kJ·kg⁻¹·K⁻¹)</td>
</tr>
<tr>
<td>ŝ</td>
<td>Entropy generation (kJ·K⁻¹)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>Ṥ</td>
<td>Power (kW)</td>
</tr>
<tr>
<td>X</td>
<td>Ammonia mass fraction</td>
</tr>
<tr>
<td>P-value</td>
<td>Probability value</td>
</tr>
<tr>
<td>F-ratio</td>
<td>Fisher statistic</td>
</tr>
<tr>
<td>Symbol</td>
<td>Subscripts</td>
</tr>
<tr>
<td>θ</td>
<td>Carnot coefficient</td>
</tr>
<tr>
<td>η</td>
<td>Exergetic efficiency</td>
</tr>
<tr>
<td>ψ</td>
<td>Specific exergy (kJ·kg⁻¹)</td>
</tr>
<tr>
<td>eₓₜ</td>
<td>Exergy destroyed by the process (kJ·kg⁻¹)</td>
</tr>
<tr>
<td>α</td>
<td>Constant</td>
</tr>
<tr>
<td>ε</td>
<td>Error</td>
</tr>
<tr>
<td>a</td>
<td>Reference</td>
</tr>
<tr>
<td>abs</td>
<td>Absorber</td>
</tr>
<tr>
<td>bp</td>
<td>Low pressure</td>
</tr>
<tr>
<td>c</td>
<td>Condenser</td>
</tr>
<tr>
<td>e</td>
<td>Intake cooling water or chilled water</td>
</tr>
<tr>
<td>g</td>
<td>Heat source</td>
</tr>
<tr>
<td>hp</td>
<td>High pressure</td>
</tr>
<tr>
<td>p</td>
<td>Ammonia-poor solution</td>
</tr>
<tr>
<td>r</td>
<td>Ammonia-rich solution</td>
</tr>
<tr>
<td>s</td>
<td>Outlet cooling water or chilled water</td>
</tr>
<tr>
<td>sys</td>
<td>System</td>
</tr>
<tr>
<td>o</td>
<td>Evaporation</td>
</tr>
</tbody>
</table>

Corresponding author: Mohamed I. ADJIBADE, research fields: absorption chiller, internal combustion engine and computational fluid dynamics.
1. Introduction

Cold generation is mainly achieved by compression refrigeration machines, which require large energy consumption. In addition, these machines use, for their operation, refrigerants that deplete the ozone layer or contribute to a considerable extent to the increase in the greenhouse effect.

Many industries generate heat from fossil fuels for their processes. After using, this heat is released into the environment as waste. Waste heat can be used to operate an absorption system for cold production [1].

Absorption chillers have several advantages compared to compression machines and appear as a promising alternative [2] for cold generation, with various advantages:

- use of thermal energy (thermal effluents, solar, biogas, etc.);
- use of environmentally friendly refrigerants;
- low maintenance due to the absence or the presence of very few mechanical moving parts; therefore, they exhibit limited operating cost;
- limited electricity consumption and hence, limited operating cost.

The power consumption required to operate the pump of an absorption system, only electrical elements of the system, is less than 1% of the cooling capacity, unlike the compression systems that require at least 20% to 50% of cooling capacity [3].

The performance of absorption chillers depends not only on the performance of various heat exchangers of the system but also on the thermodynamic and chemical properties of the used fluids [4-11].

Zhi-Gao-Lin Nuo Sun and Xie [6] worked on an experimental study of the performance of an SCCP (small combined cold and power) for air conditioning system with 24.5 kWe micro gas turbine. A lithium bromide-water (LiBr-H$_2$O) double-effect absorption refrigeration unit is used in the work. To compensate the problem of eventual high cooling demand, a burner is installed in the double-effect absorption chiller. To evaluate the performance of the cogeneration system, PER (primary energy rate) and comparative saving of primary energy ($\Delta$SCCP) demand are used. The results show that the PER and $\Delta$SCCP of the system are 0.867% and 12.3% with burner working, and 0.726% and 19.4% with no burner working, respectively.

Martinez and Pinazo [4] worked on LiBr-H$_2$O single effect absorption chiller for a design task. In this work, the effect of the variation of the surface of each component on the system performance is analyzed. This not only allows knowing the operating conditions, but also stability of the system for optimal performance. To do this, the variance analysis is done. To this end, the inlet cooling water temperature, the chilled water and the hot water of the generator are maintained respectively at 29.4 °C, 11.5 °C and 82.2 °C. Thus, for a machine whose nominal COP (coefficient of performance) equals 0.717, they obtained 0.786, representing an improvement of 9.6%, without varying nominal capacity and total heat transfer area.

Lostec et al. [8] presented numerical model of H$_2$O-NH$_3$ absorption chiller single stage operating with solar system under steady state conditions. The objective of the study is to analyze the effect of absorption and desorber temperature on system performance. The results are compared and validated with experimental data of solar absorption chiller with coolant and heat source temperature respectively equal to 19.42 °C and 86.2 °C. The results showed a 25% reduction of the COP for a decrease of 10 °C of the coolant temperature when there is an increase of 4% for an elevation of 10 °C of heat source temperature.

Adewusi and Zubair [5] made an entropy analysis of H$_2$O-NH$_3$ absorption chillers single-stage and two-stage in order to reduce the entropy generated. They computed some parameters such as the COP of the machine and the total entropy generated flow in the system. This study is performed with an evaporation temperature of -10 °C, condensation temperature of 40 °C, and heat source temperature of
about 135 °C and 200 °C respectively for single- and two-stage systems. The two-stage system with a COP of 0.734 is more efficient than the single-stage system with a COP of 0.598. Moreover, the total entropy generated for the two-stage system is greater than that for the single stage system. The authors showed that the increase in total entropy generated two-stage level is due to the irreversibility in the desorber as it produces about 50% of the total entropy generated in the system. To correct this, they advocated reducing the heat to the generator so as to have a better performance factor.

Bazzo et al. [9] worked on experimental study of combining 28 kWe natural gas micro turbine, a thermosyphon HRSG (heat recovery steam generator) and 133 kW H₂O-NH₃ absorption chiller fired by steam, initially operating with natural gas. The aim of this work is to evaluate the chiller performance with different micro turbine power outputs. When comparing the initial COP of chiller system to the modified system, a reduction of 0.34 is recorded. They have shown that this is due to the reduction of heat transfer modes, convective and radiative for initial system and only convective for the modified system.

Aman and al. [10] realized energy and exergy analyses to evaluate the performance of the 10 kW H₂O-NH₃ absorption chiller for residential air conditioning application under steady state conditions. The system operates at an evaporation temperature of 2 °C, condensation temperature of 30 °C, and heat source temperature of 80 °C. One of the aims of their study is to maximize the exergetic efficiency of each component that can be avoided or not. The results show that the desorber and the absorber present the greatest unavoidable irreversibility. For temperatures of the heat source ranging from 343 °C to 364 °C, the irreversibilities of the desorber and absorber represent respectively 70-84% and 75-84%.

This literature review shows that the second law approach is widely used in the analysis of absorption cooling machines. Second law approaches and exergy analysis are currently gaining increased attention because of the additional information they provide on the quality of the energy conversion processes. However, to the authors’ knowledge, no published work is available detailing the implementation of the experimental planning method to predict the internal parameters having a significant influence on the H₂O-NH₃ simple effect absorption operating with evaporation temperature below 0 °C to optimize the exergetic efficiency.

During absorption chiller operation, four external temperatures can be defined which are: the heat source temperature, the cold source temperature, and the temperatures of the cooling mediums for the absorber and the condenser. These four external temperatures, correspond four internal temperatures which are respectively, the desorber, evaporation, absorber and condensation temperature. In this work, in order to improve the performance of a H₂O-NH₃ single stage absorption cooling system, the triple effect absorption presents the best COP and exergy efficiency. Indeed, for a COP ranging from 0.73 to 0.79 for the single effect, it ranges from 1.62 to 1.9 for the triple effect. Similarly, the exergy efficiency varies from 12.5% to 23.2% for the single effect and 17% to 25.2% for the triple effect absorption cooling system.

Gong and Boulama [11] worked on advanced exergy analysis of a single-effect water-lithium bromide absorption refrigeration machine. A numerical model is developed and validated from experimental results in literature. The aim is to determine the irreversibility in each component that can be avoided or not. The results show that the desorber and the absorber present the greatest unavoidable irreversibility. For temperatures of the heat source ranging from 343 °C to 364 °C, the irreversibilities of the desorber and absorber represent respectively 70-84% and 75-84%.
absorption machine, the exergetic efficiency, irreversibility in each heat exchanger is calculated. The simulation is run with Matlab and the optimization is performed using the statistical method of experimental design. The problem to be solved can be summarized as follows: with the four external temperatures and fixed cooling capacity, what are the values of the established internal temperatures that minimize irreversible losses in heat exchangers? This is an optimization problem, where the exergetic efficiency is the objective function to be maximized and the independent variables are the four internal temperatures of the system.

2. Description of the System

The system in study is a H₂O-NH₃ pair single-stage absorption cooling system. The system is composed of four essential components: desorber, condenser, evaporator and absorber (Fig. 1).

The NH₃-rich solution (5) absorbs a quantity of heat Qₘ at the temperature Tₘ of the heat source, causing the vaporization of a portion of the NH₃. At the outlet of the desorber, are obtained NH₃ vapor (8) and a NH₃ poor solution (6), which returns to the absorber. The NH₃ vapor reaches the condenser. The latter is cooled by a heat transfer fluid which may be water or air, which extracts the heat necessary for the condensation of NH₃. The condensed NH₃ (1) is expanded and injected into the evaporator (2) where it absorbs heat from the medium to be cooled. The NH₃ vapor from the evaporator (3) is admitted to the absorber where it mixes with the NH₃-poor solution coming from the desorber to produce the rich solution (4). The heat from this exothermic transformation is removed by a cooling fluid at ambient temperature.

Maintaining a pressure difference between the whole absorber/evaporator (low pressure) and desorber/condenser (high pressure) further requires the presence of a pump in the circuit of the NH₃-rich solution. Also, the presence of two regulators on refrigerant circuits and refrigerant-poor solution is required. To avoid the pump cavitation, the ammonia-rich solution must be subcooled before it leaves the absorber (its temperature should be lower than its boiling start temperature).

---

Fig. 1  Simplified diagram of an absorption chiller.
The benefits of inserting recovery heat exchangers between the desorber and the absorber, between the condenser and the evaporator, or a combination are examined by Sözen [12]. The author concluded that having both heat exchangers or only having the heat exchanger between the desorber and absorber resulted in comparable performances, while only having a heat exchanger between the condenser and evaporator did not have a noticeable effect on the performance of the machine. A solution heat exchanger installed on the system of Fig. 1 allows an increase of approximately 22% of the initial COP [13]. So, to reduce both external heat and entropy generation, the solution heat exchanger is important for heat recovery. It will reduce both the power supply to the desorber, preheating the rich solution of ammonia before its admission into the desorber, and the evacuation to the absorber. The power that it will evacuate to the intermediate source for a given absorber outlet temperature will be lower. Decreasing the power to the desorber will result in increased efficiency of the refrigeration even about 60% [14]. Fig. 2 illustrates H₂O-NH₃ absorption cycle study.

3. Modeling Absorption Chiller H₂O-NH₃

A simulation program, with Matlab, is developed for the assessment of thermodynamic performance of the single-stage absorption chiller. The mass, energy and exergy balances of each component are carried out by following the model in Fig. 2. The thermodynamic properties of H₂O-NH₃ are used to determine the physical parameters of each system point [1, 15-23].

The mass balance of the mixture, the refrigerant enthalpy and entropy components are obtained from the following equations.

Mass balance of the mixture:
\[ \sum \dot{m}_\text{in} = \sum \dot{m}_\text{out} \]  
(1)

Mass Balance of the refrigerant:
\[ \sum \dot{m}_\text{in} \chi_\text{in} = \sum \dot{m}_\text{out} \chi_\text{out} \]  
(2)

Energy balance:
\[ \dot{Q} = \sum \dot{m}_\text{in} h_\text{in} - \sum \dot{m}_\text{out} h_\text{out} \]  
(3)

Entropy balance:
\[ \dot{S} = \sum \dot{m}_\text{out} s_\text{out} - \sum \dot{m}_\text{in} s_\text{in} + \frac{\dot{Q}}{T} \]  
(4)

where \( \dot{m} \) is the mass flow rate, \( h \) and \( s \) are the specific enthalpy and entropy respectively at temperature \( T \) and \( \dot{Q} \) is the heat transfer rate.

The COP is the ratio of the useful energy gained
from the evaporator to the primary energy supply to the generator and mechanical work absorbed by the pump of the system.

$$COP = \frac{\dot{Q}_0}{Q_g + W}$$  \hspace{1cm} (5)

The second law of thermodynamics can be expressed assuming reversible transformation as follows.

$$\frac{Q_0}{T_0} + \frac{Q_g}{T_g} + \frac{Q_c + Q_{abs}}{T_c} = 0$$  \hspace{1cm} (6)

From Eq. (6), one can express the ratio \(\frac{\dot{Q}_0}{\dot{Q}_g}\) based on the temperatures involved in the system.

$$\frac{\dot{Q}_0}{\dot{Q}_g} = \frac{T_a}{T_g} \frac{T_g - T_c}{T_c - T_0}$$  \hspace{1cm} (7)

Exergetic balance:

$$I_r = \sum (\dot{m}_{in} \psi_{in} - \dot{m}_{out} \psi_{out}) + \dot{Q} \left(1 - \frac{T_a}{T}ight)$$  \hspace{1cm} (8)

where \(\psi\) the specific exergy and \(T_a\) environmental temperature. In this analysis, \(T_a\) is set to 298.15 K.

4. Exergy Balance of the System

The exergy balance system gives us:

$$\theta_g \cdot Q_g + \theta_c \cdot Q_c + \theta_0 \cdot Q_0 - e_{xD} + W = 0$$  \hspace{1cm} (9)

with:

$$\theta_g = 1 - \frac{T_a}{T_g}$$  \hspace{1cm} (10)

$$\theta_c = 1 - \frac{T_a}{T_c}$$  \hspace{1cm} (11)

$$\theta_0 = 1 - \frac{T_a}{T_0}$$  \hspace{1cm} (12)

\(W\) is the pump power consumption and \(e_{xD}\) is the exergy destroyed by the process.

The maximum thermal performance of an absorption refrigeration system is determined by assuming that the entire cycle is totally reversible (i.e., the cycle involves no irreversibility nor any heat transfer through a differential temperature difference) [9], it is possible to obtain:

$$COP_E = \left(1 - \frac{T_a}{T_g}\right) \left(\frac{1}{\frac{T_a}{T_c} - \frac{T_0}{T_0}}\right)$$  \hspace{1cm} (13)

The second law efficiency of the absorption system leads to computing the exergetic efficiency, which is defined as the ratio of the useful exergy gained from a system to that supplied to the system [24]. The ratio between the two COP definitions in Eqs. (5) and (13) is the exergetic efficiency of the machine, and gives an extent of the thermodynamic irreversibilities associated with the absorption cooling process.

$$ECOP = \frac{COP}{COP_E}$$  \hspace{1cm} (14)

By using Eqs. (5), (7) and (13), the exergetic efficiency of the system can be calculated as follows with \(W\) negligible.

$$ECOP = \left|\frac{T_0 - T_a}{T_g - T_a}\right| \left(\frac{T_c - T_a}{T_c - T_0}\right)$$  \hspace{1cm} (15)

In order to simulate the single effect absorption cooling system, several assumptions are made [25]:

- The analysis is made under steady conditions;
- The temperatures in components (desorber, condenser, evaporator and absorber) are uniform throughout the volume considered;
- The mixture at the outlet of the absorber and the desorber is in the saturated state. The respective mass fractions and temperatures are on equilibrium values corresponding to the pressure in the heat exchangers;
- The coolant leaving the evaporator and the condenser is in the saturated state;
- The process through the pump is isenthalpic;
- The heat exchange with the environment surrounding and the pressure losses are assumed to be negligible;
- The desorber and the condenser are at the same pressure; it is the same case for the absorber and the evaporator.

5. The Input Data and Thermodynamics Properties
The input data are parameters which are assumed to be known. These are:

1. the cooling medium which is the same in the absorber, the condenser and its temperature;
2. the heat source and temperature;
3. the medium to be cooled and the temperature to keep it there.

In Fig. 2, States (9)-(12) require the thermodynamic properties for NH₃ and States (1)-(8) are based on H₂O-NH₃ mixtures. The two-phase equilibrium pressure and temperature of NH₃, the specific enthalpies of saturated NH₃ liquid and NH₃ vapor in terms of temperature, the relation between saturation equilibrium pressure, mass fraction and temperature of a water-ammonia mixture and the specific volume of the mixture have been calculated using equations from the literature [20, 22, 26]. The entropy of a water-ammonia mixture in the saturated liquid phase in terms of temperature, and mass fraction has been calculated by using equation from Alamdari’s work [21].

6. Method for Resolution

For the simulation with Matlab, a method of resolution scanning system parameters is used. Thus,

- evaporation temperature: -15 to 0 °C in steps of 2.5 °C;
- condensation temperature: 30 to 40 °C in steps of 2.5 °C;
- heating medium temperature in the desorber: 80 to 200 °C with a step of 20 °C;
- solution heat exchanger effectiveness is 80%;
- cooling load.

The purpose of the program is to find the operating ranges of the absorption system for maximum exergy efficiency.

The temperature of the medium to be cooled imposes the condensation temperature and consequently the pressure in the whole condenser/desorber. The evaporation temperature and, consequently, the pressure in the evaporator/absorber assembly are fixed by the chilling water or temperature of the medium to be cooled.

7. Validation of the Numerical Model

A commercial absorption chiller, designed for solar air conditioning applications, was tested under various operating conditions to assess its performance. The experimental data reported by Lostec et al. [27] and the simulation results obtained by Lostec et al. [8] are used to validate our model. The validation parameters employed are presented in Table 1. It shows temperatures at 10 locations, the internal pressures, the thermal loads in the main components and the COP, of the absorption chiller reported by Lostec et al. [8, 27] and the values obtained in the present work. The simulation parameters for the validation are as follows:

- cooling temperature: 22.33 °C;
- temperature at the inlet of the desorber (26): 86.29 °C;
- evaporation temperature: 8 °C;
- cooling load: 10.5 kW.

In Fig. 3, the temperatures calculated using the present model are compared with the experimental data from Lostec et al. [27] and simulated temperatures reported by Lostec et al. [8].

In the case of comparison with experimental data from Lostec et al. [27], the maximum relative error obtained is 13.1% for our model and is 27.68% for Lostec et al. model [8]. The results obtained in this work agree well with the two sets of data. Fig. 4 shows a similar comparison with the heat duties involved in the system. The most difference was

Table 1 Comparison between experimental data [27], the simulation results [8] and the present model in terms of internal pressure in the system and chiller COP.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Experimental value</th>
<th>Simulation result [9]</th>
<th>Present model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bp (kPa)</td>
<td>530</td>
<td>570</td>
<td>560.7</td>
</tr>
<tr>
<td>Hp (kPa)</td>
<td>1,300</td>
<td>1,280</td>
<td>1,293</td>
</tr>
<tr>
<td>COP</td>
<td>0.60</td>
<td>0.60</td>
<td>0.66</td>
</tr>
</tbody>
</table>
Fig. 3  Temperature comparison at different locations in the chiller.

Fig. 4  Component heat duties comparison.
obtained for the pump power with a relative error of 65% for Lostec et al. and 15.45% for the present work. However, the results of the modeling of the absorption chiller are close to the experimental data.

Table 1 presents a comparison of the internal pressures of our model and Lostec et al. [8] with those of the test bench [27]. For internal pressure, the simulation results and those of Lostec et al. [8] compared with experimental results have very little difference with a maximum relative error of 8% for both simulations. The COP of the model is slightly different compared to the test bench with a relative error of 10%.

8. Quadratic Model of Experimental Design

The factorial design study allows analyzing the direct effects and interactions on the response function, which is the exergetic efficiency to be maximized. This method, based on the ANOVA (analysis of variance) determines the influence and interaction between significant factors in relation to the standardized effect.

We have three factors, the condensation/absorption temperature (A), heat source temperature (B) and evaporation temperature (C). To explore optimal operating region and to detect second order effects, factorial design with three levels, three blocks and two random center points per block is used. To do this, a mathematical function is selected which links the response, ECOP, to the factors. The model for such an experiment is:

$$ECOP_{ijk} = \alpha + A_i + B_j + AB_{ij} + C_k + AC_{ik} + BC_{jk} + ABC_{ijk} + \epsilon_{ijk}$$

$$\epsilon_{ijk}$$ represents the lack of fit due to the difference between the developed model and the actual model. Each factor is included as a nominal factor rather than as a continuous variable. In such cases, main effects have 2 degrees of freedom, two-factor interactions have $2^2$ degrees of freedom and $k$-factor interactions have $2^k$ degrees of freedom. In this model we have $i = 1, 2, 3$, and similarly for $j$ and $k$. If there is no replication, the fit is exact and there is no error term in the model and if one assumes that there are no three-factor interactions, we have 33 samples to analyze. The levels and values for the three temperatures are shown in Table 2.

The samples are simulated by the centurion Statgrahics software and Matlab simulation allowed us to extract the corresponding ECOP.

The Pareto chart (Fig. 5) shows the influence of the direct effects and their interaction on the exergy efficiency. The blue vertical line represents the probability equal to 0.05 thresholds for statistical significance. The analysis shows that 5 factors and their interactions have a significant influence on the exergy efficiency. This implies that these five factors are significantly different from zero at the 95% confidence level.

The R-squared statistic indicates that the model as fitted explains 98.87% of the variability in exergetic efficiency. The adjusted R-squared statistic, which is more suitable for comparing models with different numbers of independent variables, is 95.01%.

In the following analysis, no significant effects are excluded. The reduction of these effects allows having the adjusted R-squared at 96.88%.

Fig. 6 presented the effect of system temperatures and pressures on the ammonia mass fraction. For a given absorption temperature, the low pressure and the mass fraction of the rich solution of ammonia evolve proportionally; while for a given low pressure, the

<table>
<thead>
<tr>
<th>Level</th>
<th>A (°C)</th>
<th>B (°C)</th>
<th>C (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-1</td>
<td>30</td>
<td>90</td>
<td>-15</td>
</tr>
<tr>
<td>0</td>
<td>35</td>
<td>140</td>
<td>-7.5</td>
</tr>
<tr>
<td>1</td>
<td>40</td>
<td>190</td>
<td>0</td>
</tr>
</tbody>
</table>
absorption temperature and the mass fraction of the rich solution of ammonia are inversely proportional. It is the same for the mass fraction of the low solution of ammonia as a function of the high pressure and the temperature of the heat source. Indeed, increasing the pressure improves the condensation compared to the evaporation which is beneficial to the system for lower pressures than for higher pressures.

Fig. 6 Effect of system temperatures ($T_g, T_{abs}$) and pressures ($B_p, h_p$) on the ammonia mass fraction ($X_r, X_p$).

Fig. 7 shows that, for a given heat source temperature, the increase of the condensation temperature decreases the exergetic efficiency. Indeed, the increase of the condensation temperature leads to the increase of the temperature difference between the coolant and the refrigerant in the condenser. Thus for an evaporation temperature of 0 °C the exergetic efficiency is maximal for a
condensation temperature below 33 °C.

Fig. 8 shows that increasing the temperature of the heat source, involves that the exergetic efficiency reached a maximum before decreasing. Thus, the heat source temperature for which the exergy efficiency is maximal. The change of this heat source temperature depends on the condensation temperature. For a heat source temperature below 95 °C and a condensation temperature below 31 °C, maximum exergetic efficiency is also obtained.

The normal probability plot of the residual (Fig. 9) indicates that the random errors for the process are...
drawn from approximately normal distributions. In each case there is a strong linear relationship between the residuals and the theoretical values from the standard normal distribution.

In order to delineate optimal areas operating, Fig. 10 shows a three-dimensional plot of the relationship between the three internal temperatures of the system. The analysis shows that the internal temperatures of the system have a strong interaction on the exergy efficiency. So, the choice of these temperatures depends on the application to be realized.

Thus, with exergy efficiency greater than 0.4, two operating modes are possible according to the condensation temperature:

\[
30 \degree C \leq T_c \leq 32 \degree C \\
90 \degree C \leq T_{ch} \leq 150 \degree C \\
-6 \degree C \leq T_0 \leq 0 \degree C \\
38 \degree C \leq T_e \leq 40 \degree C \\
160 \degree C \leq T_{ch} \leq 190 \degree C \\
-15 \degree C \leq T_o \leq -10 \degree C
\]

Definitively, taking into account parameters that affect exergy efficiency, the regression equation of the ECOP can be written:

\[
ECOP = 2.403 - 0.077 T_c - 0.0054 T_g + 0.025 T_o + 0.0004 T_c \cdot T_g - 0.000036 T_g^2 - 0.00018 T_o \cdot T_g
\]

\[(17)\]

9. Conclusion

The aim of this study is to investigate by experiment design method the values of the internal temperatures in the system allowing exergetic efficiency to be maximized. In regards to this, exergy analysis of a 10 kW water-ammonia absorption chiller is performed and exergetic efficiency is calculated.

The second law efficiency of the system is investigated and compared under different system operating conditions. The results show that the exergetic efficiency of the system decreases with increasing absorption and condensation temperatures. Increasing the heat source temperature makes the exergetic efficiency reach a maximum before decreasing. The analysis reveals that the cycle is more thermodynamically efficient when the absorption cooling system is operated at a low evaporation temperature rather than a high evaporation temperature. Indeed, the maximum exergetic efficiency is obtained when the condensation temperature is low (about 32 \degree C) or high (about 38 \degree C). It is low for a heat source temperature and evaporation respectively ranging between 90 \degree C and 150 \degree C and -6 \degree C and 0 \degree C. It is high for a heat source temperature and evaporation respectively ranging between 160 \degree C and 190 \degree C and -15 \degree C and 10 \degree C.
Acknowledgments

We are grateful to the ICTP (The International Centre for Theoretical Physics) and ANSOLE (African Network for Solar Energy) for financial support in the frame of the ANSUP (ANSOLE SUR-PLACE Fellowship Program).

References


Exergetic Optimization of Absorption Chiller Single Stage H₂O-NH₃ by Experiments Design Method


