ADVANCED EXERGETIC STUDY TO ASSESS THE EFFECTS OF RECTIFICATION AND DISTILLATION ON ABSORPTION REFRIGERATORS

by

Salek MALAINE[a*], Najib ABABSSI[a], Mohamed CHARIA[b], Jilali DARDOUCH[c], and Abdellah BOULAL[a]

[a] Laboratory of Mechanical Engineering, Industrial Management, and Innovation, Faculty of Sciences and Techniques, Hassan First University of Settat, Settat, Morocco
[b] Thermodynamic and Energetic Team, Faculty of Sciences, Mohammed V University in Rabat, Rabat, Morocco
[c] Laboratory of Solar Energy and Environment, Faculty of Sciences, Mohammed V University in Rabat, Rabat, Morocco

Original scientific paper
https://doi.org/10.2298/TSCI220402147M

In this paper, an advanced exergetic study is carried out to improve the exergy efficiency and minimize the exergy losses of an absorption refrigerator. Two thermal processes based on rectification and distillation were proposed to meet this critical requirement. A numerical simulation model was established in the FORTRAN language, building on the analytical Gibbs free energy equations. This model was validated from a thermodynamic point of view by previously published results. Preliminary results showed that when the efficiency of the absorber and boiler is increased, the vapors produced by the boiler become enriched in ammonia, and the overall exergy efficiency increases, which reduces considerably the irreversibility of the components of the studied absorption system. A comparative study of the effect of these two thermal processes on the overall exergy efficiency and total exergy losses was evaluated simultaneously. The results show that the refrigerator with a distiller has a higher exergy efficiency (η_{ex} = 24.37% at 86 °C), and at the same time has a lower total exergy loss ($Ex_{l,tot}$ = 457.45 kW) than the refrigerator with a rectifier (η_{ex} = 22.34% at 85 °C; $Ex_{l,tot}$ = 532.37 kW). This study reveals that the distillation process can contribute more to the exergy improvement and exergy loss minimization of the studied absorption refrigerator than the rectification process.

Key words: *exergetic study, absorption refrigerator, exergy efficiency, exergetic losses, rectification, distillation*

Introduction

Absorption cooling systems have been considered a promising alternative to conventional vapor compression systems [1] because of their ability to operate with low energy [2, 3] and environment-friendly working fluid mixtures [4], such as aqua-ammonia [5]. The improvement of refrigerating machines can represent an important research axis [6] in the field of energy engineering, either at the level of the development of the mathematical thermodynamic models and/or in the proposal of thermal processes and mechanisms dedicated to the purification of refrigerants. Among these processes are distillation and rectification. Distillation is one of the physico-chemical separation and purification processes for fluid mixtures [7], which has been studied most extensively in the past [8] and is widely used in various industrial applications, especially in the chemical, biochemical and pharmaceutical industries. Meanwhile, the fluid rectification process is frequently employed as a separation and purification process in oil, gas, and especially petrochemical processing industries to generate high efficiencies [9]. In ammonia-water absorption systems, the ammonia purification process by rectification and distillation is a critical issue that has attracted the attention of several researchers to achieve efficient and reliable systems [10]. The rectifier can be considered as a single-stage distillation column. The distiller, on the other hand, can be constructed from one or more rectifiers using different plateaus or mediums and with stripping and rectification sections [11]. In the following, a brief review of the literature will be made, focusing on the use of a rectifier and a distiller as two mechanisms for purifying ammonia as a refrigerant in an absorption refrigerator.

Fernandez-Seara and Sieres [10] studied the influence of the ammonia purification process by the distillation column on the efficiency of the NH_3-water absorption system using experimental data to analyze the impact of the distillation column on the refrigerant concentration and particularly on the COP of the system. Sieres and Fernandez-Seara [12] developed a differential mathematical model based on a numerical analysis using mass and energy balances and heat transfer equations of an NH_3-water absorption refrigeration system with a distillation column to evaluate the influence of the size of a distillation column on vapor enrichment and system performance in this type of machine. This study proved that ammonia enrichment is necessary throughout the distillation column to eliminate the accumulation of water droplets in the evaporator, which strongly deteriorates the efficiency of this system. The results obtained showed that the ammonia distillate concentrations and COP values of the absorption system are significantly higher. Adewusi *et al.* [13] studied and compared two configurations (one-stage and two-stage) of absorption refrigerating machines using a rectification system to purify ammonia vapor. They investigated and evaluated the COP and total entropy generation of the two systems studied, based on the Second law of thermodynamics by using engineering equation solver (EES) software. They showed that the two-stage of the studied systems is more efficient in terms of COP and less efficient in terms of total entropy generation than the single stage. To maximize the efficiency and optimize the operating conditions of this machine using a Pareto genetic algorithm, Guzman *et al.* [14] investigated theoretically a new NH_3-water cooling system known as the Goswami cycle, which is equipped with a rectifier. An energy analytical model has been proposed by Belman-Flores *et al.* [15] to evaluate the performances of a diffusion-absorption refrigerator with an integrated rectifier system and bubble pump, using ammonia and water as fluid mixture and hydrogen as an inert gas. In this study, the effects of geometrical parameters, especially tube length and diameter ratio, on the cooling capacity and performance of this system were investigated. They concluded that the COP and cooling capacity are greater for higher geometric parameters. Another study reported by Aman *et al.* [16] to improve the performance and cooling capacity of an absorption air conditioner included a bubble pump with a separator, using a new pair $LiCl-H_2O$ as the working fluid compared to $LiBr-H_2O$, and powered by a solar thermal collector specifically for domestic uses. In the last study, the highest cooling effect and performance are obtained for the system using $LiCl-H_2O$ as the working fluid pair. However, concerning reducing the manufacturing conditions of the absorption refrigeration machine, Ahachad *et al.* [17] conducted a study of a single-stage absorption machine equipped with a bubbling system that can serve as a distillation stage. Recently, Dardouch *et al.* [18] precisely in Rabat site with co-ordinates: latitude 34°02′ North and longitude 06°48′ West developed and analyzed a simulation program written in FORTRAN to evaluate a solar NH_3-H_2O absorption refrigeration system equipped with a distillation column in the Moroccan climate. The comparative study between the proposed system and the single-stage absorption system showed that the system equipped with a distillation column and powered by flat plate collectors is the most suitable for the weather conditions in Morocco. Therefore, it should be noted that this last study does not consider the exergetic concept. The specificity of this research lies in the effort to overcome this last problem, in the development of a new methodological approach that takes into account the energetic and exergetic aspects in advanced methods.

This study aims to investigate the effect of two thermal processes, namely rectification, and distillation, on the performance and irreversibility of an absorption refrigeration system, and to present an exergetic comparison of these two thermal processes for ammonia purification with a rectifier or distiller. This comparison has not been the subject of any previous paper following an extensive literature search. At the same time, the main objective is to achieve an exergetic improvement of exergy efficiency and minimization of exergy losses by rectification and distillation, and to select from these two thermal processes the most efficient and suitable process for refrigeration by absorption way.

Materials and methods

Absorption system description

The single absorption refrigerator object of the improvement, in this study, is composed of a boiler, a condenser, an evaporator, an absorber, two expansion valves, and two heat exchangers as shown in fig. 1. It's about an absorption refrigeration machine operating with the NH_3-water couple; ammonia being the refrigerant [19] and water being the absorbent [20]. There are two circuits in this refrigeration installation: the refrigerant circuit, which is positioned between the boiler and the condenser, the evaporator and the absorber, and the refrigerant-absorbent solution circuit, which is located between the absorber and the boiler.

Figure 2(a) shows a representative diagram of an absorption refrigerator with a rectifier implanted between the outlet of the boiler and the absorber. Whereas, fig. 2(b) shows a diagram of an absorption refrigerator equipped with a distiller installed at the outlet of the boiler. The difference between these two thermal process mechanisms lies in the power source for the rectifier and the distiller. The rectifier is powered simultaneously by the vapor produced by the boiler at high pressure on one side and by the rich solution sent by the absorber, delivered using a pump and undergoing preheating via a heat exchanger on another side. While the distiller is powered by the vapor produced directly by the boiler at high pressure. These three configurations of absorption machine can follow an operation known as the *thermal driven ammonia- water refrigeration cycle* [21], describing a thermodynamic cycle in which the heat input is absorbed by the boiler at a high pressure level [22].

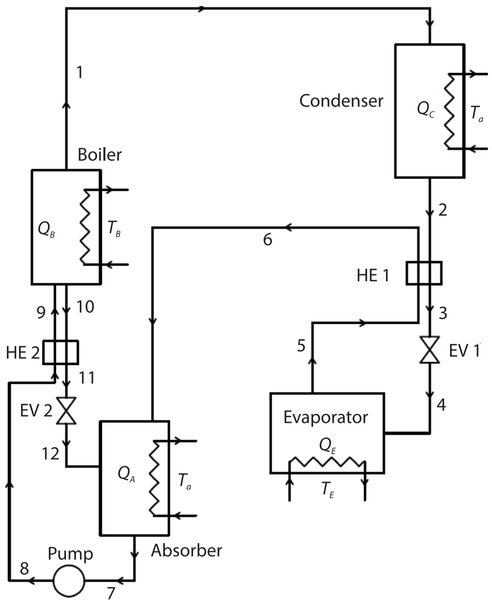

Figure 1. Schematic view of the single-stage absorption refrigerator

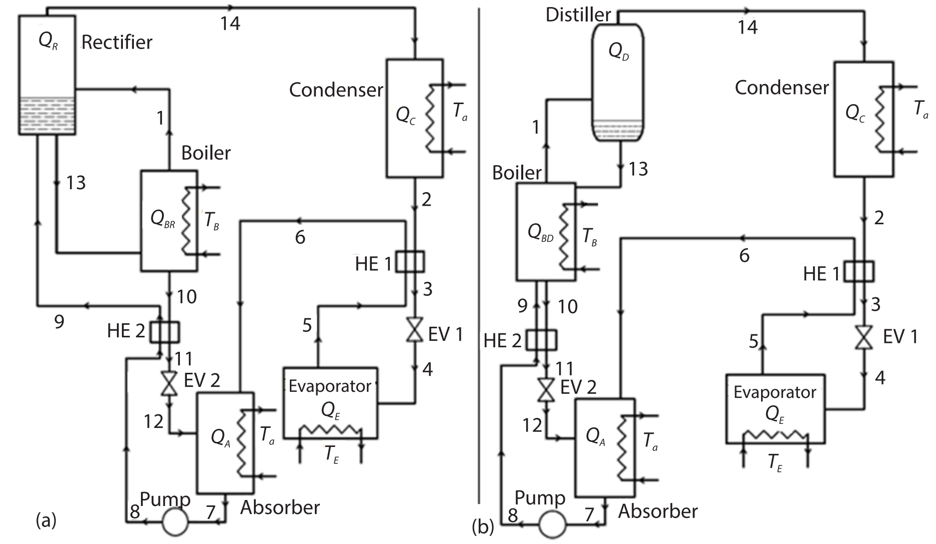

Figure 2. Schematic representation of the two proposed absorption refrigerators: with rectifier (a) and with distiller (b)

Thermodynamic model design

Before commencing any numerical simulation of any refrigeration systems object of this study, the development of a thermodynamic model constitutes a fundamental preliminary step. This model was built using the techniques for differentiating the analytical Gibbs free energy expressions given by SCHULTZ. We suggest as a second step, the development of a new methodological approach taking into account the energetic and exergetic aspects according to the first and second principles of thermodynamics.

Energetic modelling

The knowledge of the characteristics and physical properties of the refrigerant at the different points of the thermodynamic cycle to calculate, in particular, the enthalpies [20] and the entropies [23] liquids and vapors, helps in calculating the heat fluxes exchanged at the level of each organ of the absorption refrigeration machine. For this purpose, the design of a mathematical model that facilitates this calculation is a fundamental step that leads to the numerical simulation of a thermodynamic process. In this sense, several researchers are using a ready-to-use tool called EES [13, 24, 25]. It is software that makes it possible to calculate the thermodynamic properties of couples of fluids from a database provided for hundreds of refrigerants, including nevertheless the couples of NH_3-water and LiBr-water which are the most used in the field of absorption refrigeration. But in the present work, the design of a thermodynamic model for this useful purpose is necessary and makes it possible to facilitate these calculations according to the phase considered at the different points of the thermodynamic cycle of the studied refrigerators. This model is founded on Gibbs free energy analytical formulas. However, the analytical Gibbs free energy formalism established by Schultz offers an implementation of a fundamental equation expressed in the integral form [26]. To simplify the modelling of thermodynamic properties [27] we only consider the reduced variables of Gibbs free energy expressed below. In the same vein, this model was developed considering the following two scenarios:

Scenario 1. Case of pure ammonia.

For pure ammonia, the expression of Gibbs free energy [27] in its reduced form is distinguished according to two-phases:

– Vapor phase

$$G_r^v = H_{0r}^v - T_r S_{0r}^v + \int_{T_{0r}}^{T_r} C_{pr}^v \mathrm{d}T_r - T_r \int_{T_{0r}}^{T_r} \frac{C_{pr}^v}{T_r} \mathrm{d}T_r + T_r \ln\left(\frac{P_r}{P_{0r}}\right)$$
$$+C_1\left(P_r - P_{0r}\right) + C_2\left(\frac{P_r}{T_r^3} - 4\frac{P_{0r}}{T_{0r}^3} + 3P_{0r}\frac{T_r}{T_{0r}^4}\right) +$$
$$+C_3\left(\frac{P_r}{T_r^{11}} - 12\frac{P_{0r}}{T_{0r}^{11}} + 11P_{0r}\frac{T_r}{T_{0r}^{12}}\right) + \frac{C_4}{3}\left(\frac{P_r^3}{T_r^{11}} - 12\frac{P_{0r}^3}{T_{0r}^{11}} + 11P_{0r}^3\frac{T_r}{T_{0r}^{12}}\right) \tag{1}$$

with

$$C_{pr}^v = D_1 + D_2 T_r + D_3 T_r^2 \tag{2}$$

– Liquid phase

$$G_r^l = H_{0r}^l - T_r S_{0r}^l + \int_{T_{0r}}^{T_r} C_{pr}^l \mathrm{d}T_r - T_r \int_{T_{0r}}^{T_r} \frac{C_{pr}^l}{T_r} \mathrm{d}T_r + \left(A_1 + A_0 T_r + A_4 T_r^2\right)\left(P_r - P_{0r}\right) + \frac{A_2}{2}\left(P_r^2 - P_{0r}^2\right) \tag{3}$$

with

$$C_{pr}^l = B_1 + B_2 T_r + B_3 T_r^2 \tag{4}$$

*Scenario 2***.** Case of NH_3-water fluids mixture.

This is the scenario that reflects, notably, the NH_3-water pair as a mixture of working fluids necessary for the operation of the absorption refrigerator. For this couple, Gibbs free energy in its reduced form is also distinguished according to the following two-phases:

– Vapor phase

$$G_r^v = \left(1-Y\right)G_{r\mathrm{H_2O}}^v + Y\,G_{r\mathrm{NH_3}}^v + T_r\left[\left(1-Y\right)\ln\left(1-Y\right) + Y\ln Y\right] \tag{5}$$

– Liquid phase

$$G_r^l = \left(1-X\right)G_{rH_2O}^l + XG_{rNH_3}^l + T_r\left[\left(1-X\right)\ln\left(1-X\right) + X\ln X\right] + (E_1 + E_2 P_r\left(E_3 + E_4 P_r\right)T_r +$$
$$+\frac{E_5}{T_r} + \frac{E_6}{T_r^2} + \left[E_7 + E_8 P_r + \left(E_9 + E_{10}P_r\right)T_r + \frac{E_{11}}{T_r} + \frac{E_{12}}{T_r^2}\right]\cdot$$
$$\cdot\left(2X-1\right) + \left[E_{13} + E_{14}P_r + \frac{E_{15}}{T_r} + \frac{E_{16}}{T_r^2}\right]\left(2X-1\right)^2\Bigg)X\left(1-X\right) \tag{6}$$

The coefficients appearing in the aforementioned eqs. (1)-(6) are determined by the least-squares method, and this is from the experimental values available to them by the Ziegler and Trepp [28].

Through the thermodynamic properties and from the Gibbs free energy expressions all the other thermodynamic quantities can be determined by differentiation techniques, in particular, enthalpy, eqs. (7) and (9), and entropy, eqs. (8) and (10), which are those that interest us in this work.

In this respect, we can distinguish the following two-phases:
– Vapor phase

$$H^v = -T^2 \left(\frac{\frac{G^v(T,P,Y)}{T}}{\partial T} \right)_{P,Y} \tag{7}$$

$$S^v = -\left(\frac{\partial G^v(T,P,Y)}{\partial T} \right)_{P,Y} \tag{8}$$

– Liquid phase

$$H^l = -T^2 \left(\frac{\frac{G^l(T,P,X)}{T}}{\partial T} \right)_{P,X} \tag{9}$$

$$S^l = -\left(\frac{\partial G^l(T,P,X)}{\partial T} \right)_{P,X} \tag{10}$$

To make the previous equations in the dimensionless form [28], the following reduced thermodynamic parameters are considered in tab. 1.

Table 1. The reduced thermodynamic properties

$T_r = \frac{T}{T_b}$	(11)	$G_r = \frac{G}{RT_b}$	(14)
$P_r = \frac{P}{P_0}$	(12)	$H_r = \frac{H}{RT_b}$	(15)
$C_{pr} = \frac{C_p}{R}$	(13)	$S_r = \frac{S}{RT_b}$	(16)

Note: T_b= 100 °C, P_0= 10 bars, R = 3.3143 kJ/kmolK

The energy simulation model developed is based on establishing the mass and energy balances of the various components of the absorption refrigerator [29] assumed to be in a steady-state. This model is based on the principles of conservation of mass, eq. (17), and energy, eq. (18) [30]:

$$\sum \dot{m}_{\text{int}} = \sum \dot{m}_{\text{out}} \tag{17}$$

$$\dot{Q}_{\text{int}} + \dot{W}_{\text{int}} + \sum \dot{m}_{\text{int}} \left(h + \frac{v^2}{2} + gz \right)_{\text{int}} = \dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \sum \dot{m}_{\text{out}} \left(h + \frac{v^2}{2} + gz \right)_{\text{out}} \tag{18}$$

where $\dot{m}$ [kgs^{-1}], $\dot{W}$ [kW], $\dot{Q}$ [kW], and h [kJkg^{-1}] are the mass-flow rate , mechanical power, heat transfer rate, and enthalpy, respectively. Abbreviations int and out refer to the system's input and output.

The energetic performance coefficient of the absorption refrigerator is given [31]:

$$COP = \frac{Q_E}{Q_B} \tag{19}$$

Exergetic modelling

The physical exergy at different points of the refrigeration cycle is expressed [32]:

$$\dot{E}x_i = \dot{m}_i \left[\left(h - h_0 \right) - T_0 \left(s - s_0 \right) \right] \tag{20}$$

where h [kJkmol^{-1}], s [kJkmol^{-1}K^{-1}], and T_0 [°C] are specific enthalpy, specific entropy, and temperature at reference state, respectively.

The exergy rate of the transferred heat quantity and work can be expressed, respectively [33]:

$$\dot{E}x_{\dot{Q}_i} = \left(1 - \frac{T_0}{T_i} \right) \dot{Q}_i \tag{21}$$

$$\dot{E}x_{\dot{W}_i} = \dot{W}_i \tag{22}$$

Following energetic modelling, the formulation of the exergetic model is built by using the principle of entropy conservation, eq. (23), as well as the fundamental equation characterizing the exergy balance, eq. (24) [33]:

$$\sum \dot{m}_{\text{int}} s_{\text{int}} + \sum \left(\frac{\dot{Q}_{\text{int}}}{T_{\text{int}}} \right) + \dot{S}_{\text{gen}} = \sum \dot{m}_{\text{out}} s_{\text{out}} + \sum \left(\frac{\dot{Q}_{\text{out}}}{T_{\text{out}}} \right) \tag{23}$$

$$\sum \dot{E}x_{\dot{Q}_{\text{in}}} + \sum \dot{E}x_{\dot{W}_{\text{in}}} + \sum \dot{E}x_{\text{flux}_{\text{in}}} = \sum \dot{E}x_{\dot{Q}_{\text{out}}} + \sum \dot{E}x_{\dot{W}_{\text{out}}} + \sum \dot{E}x_{\text{flux}_{\text{out}}} + \dot{E}x_d \tag{24}$$

where s_{in}, s_{out}, $\dot{S}_{\text{gen}}$, and $\dot{E}x_d$ are the entropy at the entry of the system, the entropy at the exit, the entropy generation, and the exergy destruction, respectively.

Exergy efficiency

Exergy efficiency is an indicator of the quality and technical development of a refrigerator. The exergy efficiency is expressed by the relation:

$$\eta_{ex} = \left| \frac{Q_E \left(1 - \dfrac{T_0}{T_E} \right)}{Q_B \left(1 - \dfrac{T_0}{T_B} \right) + \dot{W}_p} \right| \tag{25}$$

Exergy loss

The rate of thermal exergy losses in any component of the system is calculated as [34]:

$$Ex_l = \sum Ex_{\text{int}} - \sum Ex_{\text{out}} + \sum \left[Q \left(1 - \frac{T_0}{T} \right) \right]_{\text{int}} - \sum \left[Q \left(1 - \frac{T_0}{T} \right) \right]_{\text{out}} + \sum W_{\text{int}} - \sum W_{\text{out}} \tag{26}$$

Total exergy losses

The total exergy losses of a system are equal to the sum of the exergy losses of the elements that compose it, this is can be expressed as:

$$Ex_{l,\text{tot}} = \sum_{i=1}^{n} Ex_{l,i} \tag{27}$$

where n is the number of the considered components in the system.

Methodology

The methodology adopted in this work aims at the realization of a numerical simulation program, developed in FORTRAN language, of a thermodynamic cycle representing an absorption refrigerator. Initially, this program makes it possible to calculate the thermodynamic state at any point of the refrigeration cycle according to the phase considered (liquid or vapor) from the analytical expressions of Gibbs free energy given by SCHULTZ and Antoine's equations. This program allows to calculate, subsequently, the various quantities of heat exchanged at the level of each component of the absorption refrigerator from the mass and energy balances. Secondly, this program helps to calculate the various parameters characterizing the thermodynamic cycle, namely the variation in exergetic efficiency as well as the exergetic losses that may impact the proper functioning of the refrigeration installation. Finally, the developed program does not offer any difficulty since it implements a repetitive calculation on the different variables studied. It allows us to appreciate the influence of various parameters on the overall exergetic efficiency and the total exergetic losses.

Calculation assumptions

To analyze the thermodynamic cycle of the refrigeration machine, the thermodynamic characteristics of the NH_3-water fluid mixture were taken into consideration. To simplify the calculations at the level of each component of the refrigerator cycle, the following assumptions are taken into account:
- Operation of absorption system in steady-state conditions [35].
- Expansion valves are considered isenthalpic [36].
- Heat recuperator-exchangers are chosen ideal [37].
- The pump is considered isentropic [14].
- Heat losses to the environment are negligible [10].
- Kinetic and potential energies are negligible [22].
- Absorber is considered ideal and the exchanged heat loss is not considered [38].
- The temperature pinches imposed between the heat exchanging fluids are:

$$\Delta T_\mathrm{B} = \Delta T_\mathrm{C} = \Delta \mathrm{T}_{\mathrm{HE\,1}} = \Delta T_{\mathrm{HE\,2}} = 10\ ^\circ\mathrm{C}, \text{ and } \Delta T_\mathrm{E} = \Delta T_\mathrm{A} = 5\ ^\circ\mathrm{C}$$

- The liquid mass titer x_7 of the ammonia-rich solution (leaving the absorber) must be higher than the titer of the low ammonia solution x_{10} (leaving the boiler).

Numerical simulation results and discussions

Validation of the established model from a thermodynamic point of view

For numerical validation and verification of the created simulation program about the basic system, representing the single-stage absorption refrigerator, the results obtained in the present study have been analyzed and compared with those previously published in the literature by the authors Dardouch *et al.* [18]. Figure 3 shows a comparison of the evolution of the COP of the base system, which is the subject of this study, with the previously published work of the authors Dardouch *et al.* [18].

Under the same operating conditions given in [18], it can be seen that the two curves representing the COP of the basic system are perfectly synchronized and almost identical for the two works. Moreover, these two curves have almost the same starting point and the same optimum point. Therefore, the comparison of these results shows clearly the reliability of the thermodynamic model developed in this work.

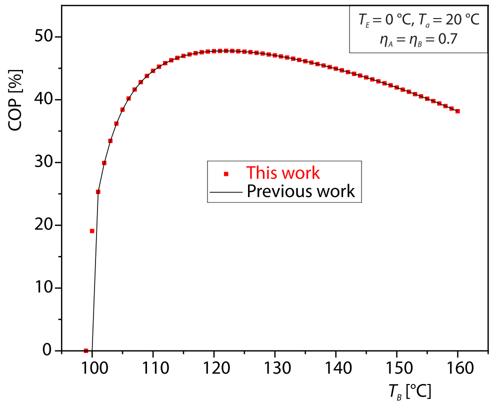

Figure 3. Comparison of the COP obtained in this work with the COP obtained and published previously in [18]

Influence of the absorber and the boiler efficiencies on the principal parameters simulated

In this section, the technical importance of the absorber and boiler performance was examined to assess their impact on the evolution of the overall exergy efficiency and total exergy losses of the single-stage absorption refrigerator. In the study, the following parameters were considered: T_a = 20 °C, T_E = 0 °C, and a vapor flow rate leaving from the boiler equal to the unity ($\dot{m}$ = 1 kg/s). Figure 4 shows the evolution of the mass-flow rates of the rich solution leaving the absorber of the single absorption refrigeration machine as a function of the temperature of the hot source T_B. The mass-flow rate curves decrease rapidly with the temperature of the hot source T_B when the efficiency of the absorber and boiler is low. On the contrary, as these efficiency levels increase, it can be seen that the mass-flow rates of the rich solution emanating from the absorber start to decrease slightly with increasing hot source temperature T_B. With this in mind, we considered equipping the absorption refrigerator with either a rectifier or a distiller, both placed at the outlet of the boiler according to the configurations shown in figs. 2(a) and 2(b), to improve the exergy efficiency of this system.

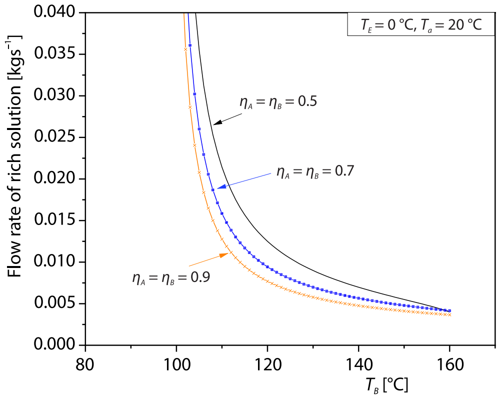

Figure 4. Representation of the mass-flow rates of the rich solution as a function of the temperature of the hot source T_B

Figure 5 depicts the influence of varying the efficiency of the absorber and boiler on the total exergy efficiency, η_{ex}, as a function of the boiler temperature, T_B, of the single absorption refrigerator. We see that if we increase the efficiency of the absorber and boiler, the total exergy efficiency, η_{ex}, increases simultaneously. The maximum values of the exergy efficiency of the single absorption refrigerator that corresponds to the variation of the absorber and boiler efficiency are obtained:

– For $\eta_A = \eta_B = 0.5$; $\eta_{ex,max}$ = 12.40% at T_B = 114 °C
– For $\eta_A = \eta_B = 0.7$; $\eta_{ex,max}$ = 14.06% at T_B = 114 °C
– For $\eta_A = \eta_B = 0.9$; $\eta_{ex,max}$ = 15.45% at T_B = 114 °C

As the absorber and boiler efficiencies increase, the exergy efficiency of the absorption refrigerator increases to its maximum and then begins to decrease with the temperature of the hot

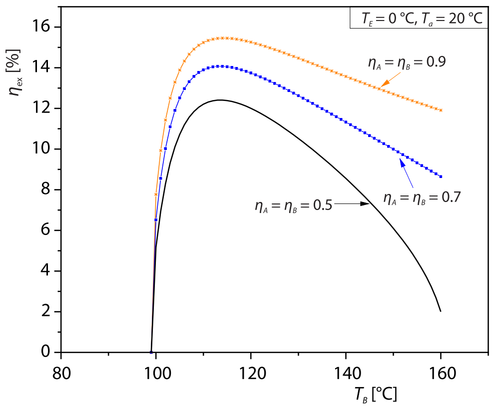

Figure 5. Effect of varying the efficiency of the absorber and the boiler on the exergy efficiency of the single absorption machine

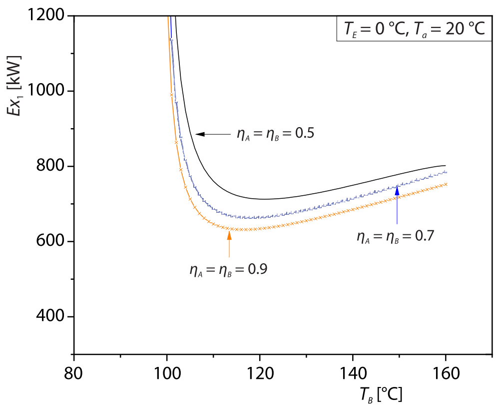

Figure 6. Effect of varying the efficiency of the absorber and the boiler on the exergy losses of the single absorption refrigerator

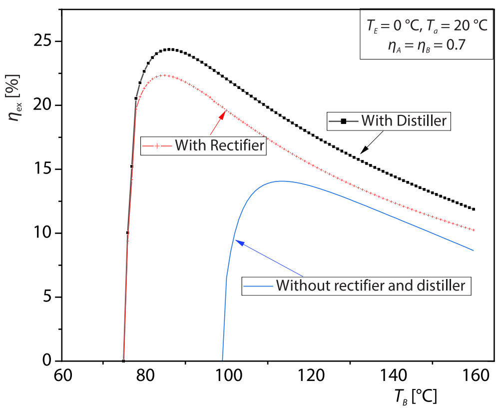

Figure 7. Variation of the overall exergy efficiency of the three analyzed configurations of the absorption refrigerator

source T_B. This can be explained by the fact that by increasing the efficiency of the absorber and boiler, at the same time, the vapors produced and discharged by the boiler are increasingly enriched with ammonia, which at the same time increases the cooling effect as well as the exergy efficiency of the absorption refrigerator.

The influence of the variation in the flow rate of the rich solution leaving the absorber and the vapors leaving the boiler on the evolution of the exergy losses was studied. Figure 6 gives an overview of the variation in the evolution of these losses as a function of the temperature T_B of the basic absorption chiller for different values of the efficiency of the absorber and the boiler. We find that the exergy loss as a function of T_B decreases significantly with increasing the absorber and the boiler efficiencies. Thus, we find the significant influence of varying absorber and boiler efficiencies on the operating conditions of the absorption refrigeration in terms of efficiency and exergy loss.

Comparison of the effect of distillation and rectification on the overall performance of the installation

After a series of analyses and evaluations of the established numerical simulation program, and to study the absorption refrigerator later under the best operating conditions, the efficiencies of the boiler, the absorber, and the pump have been taken equal to 0.7. For this purpose, fig. 7 shows the evolution of the exergy efficiency, η_{ex}, depending on the three studied configurations of the absorption refrigerator. At the beginning of the operation of these three machines, the curves representing the exergy efficiency evolve at the same rate for the three systems studied, with a shift towards a fairly large maximum reached by the refrigerator with the distiller. According to fig. 7, the absorption refrigerator equipped with a distiller has the highest exergy efficiency (equivalent to 24.37% at 86 °C) when compared to the refrigerator equipped with a rectifier and the single refrigerator (without a distiller and rectifier). So, the refrigerating machine with a rectifier possesses a better exergy efficiency (22.34% at 85 °C) than the single refrigerator (14.07% at 114 °C). In addition, the refrigerator equipped with a distiller and the refrigerator equipped with a rectifier can start its operation from a hot source temperature equal to 75 °C. While the single refrigerator (without a distiller and rectifier) can only work from a hot source temperature equal to 99 °C. However, the distiller and the rectifier can reduce the operating conditions of the absorption refrigerators.

To explain this situation, these two thermal processes can be distinguished as follows:

– In the case of the refrigerator with a rectifier: the rich solution comes out of the absorber, and before entering the boiler power the rectifier. While the vapors coming out of the boiler murmur in a solution rich with ammonia in the rectifier, which has a lower temperature than that of the boiler. As a result, the vapors coming out of the boiler become more and more enriched with ammonia via the rectifier.
– In the case of a refrigerator with a distiller: the rich solution coming from the absorber is conveyed to the boiler using a pump, and the vapors sent by the latter supply the distiller. The vapors coming out of the boiler can hamper the rich solution in the lower part of the distiller, which has a lower temperature than the boiler.

As a result, the vapors coming from the distiller are increasingly enriched and purified with ammonia, justifying this remarkable increase in exergy efficiency.

Given assessing the impact of these two thermal processes on the evolution of the total exergy losses of the absorption machine, fig. 8 shows a comparison of the variation of the total exergy losses, $Ex_{l,\mathrm{tot}}$, concerning the three configurations of the refrigerator studied. It is observed that the lowest values are obtained for the refrigerator with a distiller, followed by those obtained with a rectifier, and followed finally by those obtained by the single absorption machine without a rectifier and a distiller. The optimum minimum value of the total exergy losses equal to 457.45 kW was obtained by the refrigerator with a distiller (at the temperature of the hot source equal to T_B = 88 °C) compared to the refrigerator with a rectifier (equal to 532.37 kW at 86 °C). Then, we have demonstrated that maximizing the overall exergy efficiency of the absorption refrigerator has a positive impact on minimizing the total exergy losses caused by the irreversibility of the different components of these refrigerators, which are the subject of this study. Therefore, we have shown that distillation is the most suitable thermal process for refrigerating by absorption way.

Referring to fig. 8 and leaning on the total exergy loss curve of the refrigerator with the distiller, fig. 9 represents the distribution of the

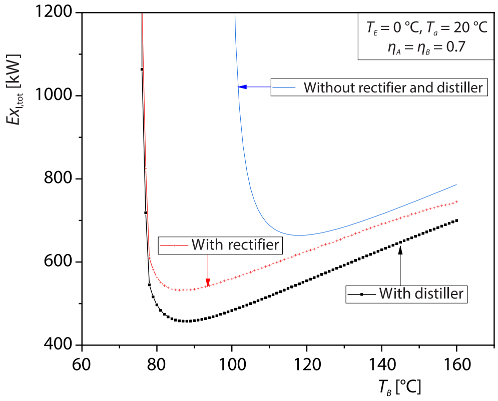

Figure 8. Variation in the total exergy losses relating to the three configurations studied of the absorption refrigerator

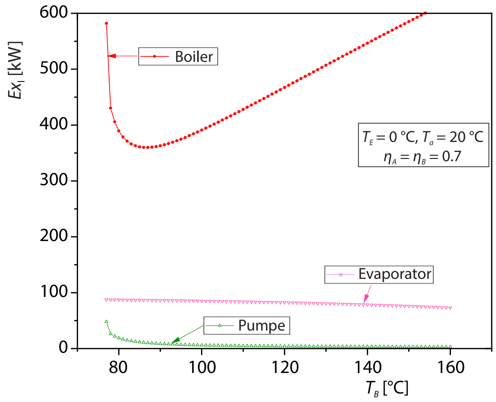

Figure 9. Exergy loss of the main components of the absorption refrigerator with the distiller

exergy losses among the main components of the absorption refrigerator cycle with a distiller as a function of the temperature of the hot source T_B. It can be seen that the major exergy losses occur in the evaporator, the pump, and the boiler. The majority of these energy losses take place in the boiler. This is due to the increase in the useful exergy required to supply and operate the boiler. It should be noted that the exergy losses decrease as the exergy consumption decreases and increase as useful exergy increases. Thus, improving the exergy efficiency means to minimizing the exergy consumption, which in turn minimizes the energy losses.

Conclusions

In this contribution, it was demonstrated that the variation in the absorber and the boiler efficiencies has a significant impact on the evolution of the absorption refrigerator's exergy efficiency and exergy losses. This allowed us to propose two thermal purification processes of ammonia fluid, relying on distillation and rectification, and to compare the effect of these two thermal mechanisms on the variation of the overall efficiency and total exergy losses of the refrigerator. It can be concluded that the distiller and the rectifier contribute to reducing the operating conditions of the absorption refrigerator. Thanks to the flexibility of the simulation program established, all parameters that can contribute to improving the exergetic performances of the refrigerator with these two thermal processes were highlighted. This improvement has helped to reduce immediately the total exergy losses due to the irreversibility of the different components of the refrigeration system under study, which can affect its proper functioning. The results demonstrated also that the distillation process is the most efficient and the best suited to the refrigeration by absorption way since it has made it possible to reduce as much as possible the droplets of water that may be contained in the vaporized part emanating from the boiler. When the overall exergy efficiency of the refrigerator with a distiller starts to increase because of the decrease in the exergy consumption at the boiler, this can lead to a decrease in their total exergy losses and vice versa. This results in a reduction of the exergies are required to ensure consequently the proper functioning of the boiler.

Despite the remarkable improvement of the refrigerator under study, it was necessary to carry out a technical and economic study to evaluate more efficiently the most suitable and economically profitable thermal process. The results obtained in this work must be validated and valorized experimentally, in order to take them into consideration in the future design and realization of new boilers integrating a rectifier or rather a distiller. In a second stage of this work, we plan to study, under real climatic conditions, the autonomous operation of an absorption refrigerator equipped with a latent heat storage system.

Nomenclature

Ex_l – exergy loss, [kW]
$Ex_{l,\text{tot}}$ – total exergy losses, [kW]
h – specific enthalpy, [kJkg^{-1}]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
n – number of cmponents in system
Q_A – heat exchanged by absorber, [kW]
Q_B – heat exchanged by boiler, [kW]
Q_{BR} – heat exchanged by boiler (in case with rectifier), [kW]
Q_{BD} – heat exchanged by boiler (in case with distiller), [kW]
Q_C – heat exchanged by condenser, [kW]
Q_D – heat exchanged by distiller, [kW]
Q_E – heat supplied to evaporator, [kW]
Q_R – heat exchanged by the rectifier, [kW]
s – specific entropy, [kJkg^{-1}K^{-1}]
T – temperature, [°C]
X – liquid mass title
Y – vapor mass title

Greek symbol

η_{ex} – exergy efficiency, [%]

Acronyms

EES – engineering equation solver
EV 1, EV 2 – expansion valve number 1 and 2

HE 1, HE 2 – heat exchanger number 1 and 2
LiBr-H_2O – lithium bromide-water mixture
LiCl-H_2O – lithium chloride-water mixture
NH_3-H_2O – ammonia-water mixture

References

[1] Nikbakhti, R., *et al*., Absorption Cooling Systems – Review of Various Techniques for Energy Performance Enhancement, *Alexandria Engineering Journal*, *59* (2020), 2, pp. 707-738

[2] He, Y., Chen, G., Equivalent Cycle and Optimization of Auto-Cascade Absorption Refrigeration Systems, *Journal Therm. Sci.*, *29* (2020), 4, pp. 1053-1062

[3] Wen, T., *et al*., Review on the Fundamentals and Investigations of Falling Film Dehumidification/Absorption Refrigeration Based on CFD Technology, *International Journal of Heat and Mass Transfer*, 171 (2021), 121042

[4] Papadopoulos, A. I., *et al.*, Systematic Assessment of Working Fluid Mixtures for Absorption Refrigeration Based on Techno-Economic, Environmental, Health and Safety Performance, *Energy Conversion and Management*, 223 (2020), 113262

[5] Yang, L., *et al*., Dynamic Characteristics of an Environment-Friendly Refrigerant: Ammonia-Water Based TiO_2 Nanofluids, *International Journal of Refrigeration*, *82* (2017), Oct., pp. 366-380

[6] Malaine,S., *et al*., Enhancing Benefits by Rectification in the Absorption Refrigeration Systems, *International Journal of Thermodynamics*, *25* (2022), 1, pp. 29-37

[7] Trubyanov, M. M., *et al*., A Hybrid Batch Distillation/Membrane Process for High Purification – Part 1: Energy Efficiency and Separation Performance Study for Light Impurities Removal, *Separation and Purification Technology*, *241* (2020), 116678

[8] Blahušiak, M., *et al*., Quick Assessment of Binary Distillation Efficiency Using a Heat Engine Perspective, *Energy*, *116* (2016), Dec., pp. 20-31

[9] Magaril, E., Magaril, R., Effect of Pressure on the Rectification Sharpness in Rectifying Sections of Tray Distillation Columns in Oil and Gas Refining, *Separation and Purification Technology*, *223* (2019), Sept., pp. 49-54

[10] Fernandez-Seara, J., Sieres, J., The Importance of the Ammonia Purification Process in Ammonia-Water Absorption Systems, *Energy Conversion and Management*, *47* (2006), 13, pp. 1975-1987

[11] Fernandez-Seara, J., *et al*., Distillation Column Configurations in Ammonia-Water Absorption Refrigeration Systems, *International Journal of Refrigeration*, *26* (2003), 1, pp. 28-34

[12] Sieres, J., Fernandez-Seara, J., Evaluation of the Column Components Size on the Vapour Enrichment and System Performance in Small Power NH_3-H_2O Absorption Refrigeration Machines, *International Journal of Refrigeration*, *29* (2006), 4, pp. 579-588

[13] Adewusi, S. A., Zubair, S. M., Second Law Based Thermodynamic Analysis of Ammonia-Water Absorption Systems, *Energy Conversion and Management*, *45* (2004), 15-16, pp. 2355-2369

[14] Guzman, G., *et al*., Thermal Optimization of a Dual Pressure Goswami Cycle for Low Grade Thermal Sources, *Entropy*, *21* (2019), 7, 711

[15] Belman-Flores, M., *et al*., Energetic Analysis of a Diffusion-Absorption System: A Bubble Pump under Geometrical and Operational Conditions Effects, *Applied Thermal Engineering*, *71* (2014), 1, pp. 1-10

[16] Aman, J., *et al*., Bubble-Pump-Driven LiBr-H_2O and LiCl-H_2O Absorption Air-Conditioning Systems, *Thermal Science and Engineering Progress*, *6* (2018), June, pp. 316-322

[17] Ahachad, M., *et al*., Study of an Improved NH_3-H_2O Solar Absorption Refrigerating Machine in Rabat (Morocco), *Solar Energy Materials and Solar Cells*, 28 (1992), 1, pp. 71-79

[18] Dardouch, J., *et al*., Study of a Solar Absorption Refrigeration Machine in the Moroccan Climate, *Materials Today: Proceedings*, *13* (2019), Part 3, pp. 1197-1204

[19] Rosa, A. C., *et al*., Quantitative Risk Analysis Applied to Refrigeration's Industry Using Computational Modelling, *Results in Engineering*, *9* (2021), 100202

[20] Sencan, A., *et al*., Prediction of Liquid and Vapor Enthalpies of Ammonia-Water Mixture, Energy Sources – Part A: *Recovery, Utilization, and Environmental Effects*, *33* (2011), 15, pp. 1463-1473

[21] Taboas, F., *et al*., Flow Boiling Heat Transfer of Ammonia/Water Mixture in a Plate Heat Exchanger, *International Journal of Refrigeration*, *33* (2010), 4, pp. 695-705

[22] Alsarayreh, A. A., *et al*., Energy and Exergy Analyses of Adsorption Chiller at Various Recooling-Water and Dead-State Temperatures, *Energies*, *14* (2021), 8, 2172

[23] Alamdari, G. S., Simple Equations for Predicting Entropy of Ammonia-Water Mixture, *IJE Transactions B: Applications*, *20* (2007), 1, pp. 97-106

[24] Khaliq, A., *et al*., Energetic and Exergetic Analyses of a Solar Powered Combined Compression-Absorption Refrigeration System, *International Journal of Exergy*, *34* (2021), 4, pp. 448-474
[25] Okwose, C. F., *et al*., Performance Analysis of Compressor-Assisted Two-Stage Triple Effect Absorption Refrigeration Cycle for Power and Cooling, *Energy Conversion and Management*, 227 (2021), 113547
[26] Kherris, S., *et al*., Contribution Study of the Thermodynamics Properties of the Ammonia-Water Mixtures, *Thermal Science*, *17* (2013), 3, pp. 891-902
[27] Mejbri, K., Bellagi, A., Modelling of the Thermodynamic Properties of the Water-Ammonia Mixture by Three Different Approaches, *International Journal of Refrigeration*, *29* (2006), 2, pp. 211-218
[28] Ziegler, B., Trepp, C., Equation of State for Ammonia-Water Mixtures, *International Journal of Refrigeration*, *7* (1984), 2, pp. 101-106
[29] Mazouz, S., *et al*., System Modelling and Parameter Effects on Design and Performance of a New Refrigeration Machine, *International Journal on Energy Conversion (IRECON)*, *2* (2004), 3
[30] Hasan, A., Dincer, I., An Ocean Thermal Energy Conversion Based System for District Cooling, Ammonia and Power Production, *International Journal of Hydrogen Energy*, *45* (2020), 32, pp. 15878-15887
[31] Bellos, E., *et al*., Yearly Investigation of a Solar-Driven Absorption Refrigeration System with Ammonia-Water Absorption Pair, *Thermal Science and Engineering Progress*, *23* (2021), 100885
[32] Chen, J., *et al*., A Sustainable Syngas Cryogenic Separation Process Combined with Ammonia Absorption Refrigeration pre-Cooling Cycle, *Journal of Cleaner Production*, *313* (2021), Sept., 127612
[33] Yilmaz, f., Energy, Exergy and Economic Analyses of a Novel Hybrid Ocean Thermal Energy Conversion System for Clean Power Production, *Energy Conversion and Management*, *196* (2019), Sept., pp. 557-566
[34] Kalla, S. K., Usmani, J. A., Performance Analysis of Vapour Compression Refrigeration System with R502 and its Substitutes, *International Journal on Energy Conversion (IRECON)*, *1* (2013), 4
[35] Valles, M., *et al*., Solar-Driven Absorption Cycle for Space Heating and Cooling, *Applied Thermal Engineering*, *168* (2020), 114836
[36] Satyro, M. A., *et al*., Temperature Change from Isenthalpic Expansion of Aqueous Triethylene Glycol Mixtures for Natural Gas Dehydration, *Fluid Phase Equilibria*, *305* (2011), 1, pp. 62-67
[37] Fakheri, A., Heat Exchanger Efficiency, *Journal of Heat Transfer*, *129* (2007), 9, pp. 1268-1276
[38] Ebrahimi, A., *et al*., Novel Integrated Structure Using Solar Parabolic Dish Collectors for Liquid Nitrogen Production on Offshore Gas Platforms (Exergy and Economic Analysis), *Sustainable Energy Technologies and Assessments*, *7* (2020), 100606

Paper submitted: April 2, 2022
Paper revised: July 11, 2022
Paper accepted: July 18, 2022