Evaluation of Performance of a Commercial Absorption Chiller

I. Boukholda, S. Elmay and A. Bellagi
Unité de Recherche Thermique et Thermodynamique des Procédés, URTThP
Ecole Nationale d'Ingénieurs de Monastir, E.N.I.M.
Avenue Ibn ElJazzar, 5019 Monastir
Tunisia

Abstract: This study describes and analyses the performance of a water ammonia absorption refrigeration system. This analysis is based on the first and second thermodynamic laws, so a steady state model using the exergy method is developed and some working parameters are varied to evaluate the system performance. The used model involves operating parameters such as overall heat transfer coefficient of each component (UA), air temperature, and inlet chilled water. We have examined the coefficient of performance (COP) and the exergetic efficiency (ξ) under various operating conditions. The results show that the performance of the ARS increases with decreasing of air temperature, which means also a decreasing of both condenser and absorber temperatures. Moreover, the performance of the ARS goes to better with an increasing of inlet water temperature which means an increasing of evaporator temperature at a same flow rate. Under various operating conditions of the ARS, The highest destroyed exergy of the system is examined in generator, whereas the lowest exergy destroyed is evaluated in the expansion valves, the pump solution and in the refrigerant heat exchanger compared to the other components.

Key-Words: Absorption; Refrigeration; Exergy; Ammonia-water

1 Introduction

The growing use and the increasing cost of electricity as well as the serious ecological problems caused by the use of CFC has made absorption refrigeration system (ARS) an alternative of mechanical vapor compression refrigerators. Since ARS, especially at low temperatures can exploit a renewable energy sources such as solar energy, biomass or geothermal for functioning, moreover the employed refrigerants are known by their environment friendliness. These interesting opportunities offered by the ARS systems have led to the increase of theoretical and experimental researches on this kind of refrigeration. These researches are divided into two classes: energy analysis and exergy analysis.
The energy analysis is based on the first law of thermodynamics which describes the energy conservation. Among these works we can cite: Omer Kaynakli et al [1] have studied theoretically the effect of operating conditions on the performance of a water/lithium bromide absorption refrigeration system by varying different parameters: These parameters are the temperature of condenser, absorber, evaporator and generator also the effectiveness of heat exchangers. F. Asdrubali and al [2] have experimentally evaluated the performance of the same type of absorption refrigeration system for different service conditions in order to determinate the optimum case of functioning. In the same context, I. Hrouz et al [3] have carried out an experimental study of water ammonia absorption chiller producing 10 kW cooling utility. They emphasize the effect of the variation of the ambient and chilled water temperatures and the input heat to generator on the performance of system. Adnan Sozen [4] has interested in the effect of heat exchangers on the performance of ARS, however he has discussed three different cases including heat exchangers: Firstly he considered a system operating only with a refrigerant heat exchanger, secondly with only solution heat exchanger and finally including both in the same cycle. M.de Vega and J.A. Almendros-Ibanez [5] have chosen a special type of heat exchangers (plate of heat exchanger) and they have examined their effects on the performance of a water/lithium bromide absorption system.

The exergy analysis known as the second law method is extensively analysed in the literature. In fact, some searchers have used the principle of entropy generation minimization to improve the system performance; among them, we can cite Bejan [6]. While others have used the exergy criteria based also on the second law: Talbi et al [7] have established a general exergy analysis of water/LiBr absorption refrigeration system. Arzu Sencan et al [8] have examined the exergy loss of each component, the coefficient of performance (COP) and the exergetic efficiency (ξ) of single effect water/lithium bromide absorption system designed for cooling and heating applications.

In this study, we are interesting on GAX absorption system using water ammonia mixture; water as absorbent solution and ammonia as the refrigerant. We have made a theoretical evaluation of its performance and the effect of some design parameters and operating conditions on the performance coefficient (COP), the total destroyed exergy ($\dot{E}_d$) and the exergetic efficiency (ξ). This study is based on experimental investigation realized by Klein [9] who has treated a commercial absorption chiller able to produce 10.55 KW cooling capacity at a nominal condition (T_{air} = 35°C).

2 Exergy analysis

Conventional absorption system consist of four basic components; a generator (Gen), a condenser (Cond), an absorber and an evaporator (Evap) separated by expansion valves and solution pump. The system case of study is modified by including some heat exchangers that improve the efficiency as showing Fig.1.

These heat exchangers are the rectifier (Rect), refrigerant heat exchanger (Rhe), solution cooled absorber (Sca), and the air-cooled absorber (Aca). The absorption process in the system is realised in the two last heat exchangers (Sca and Aca).

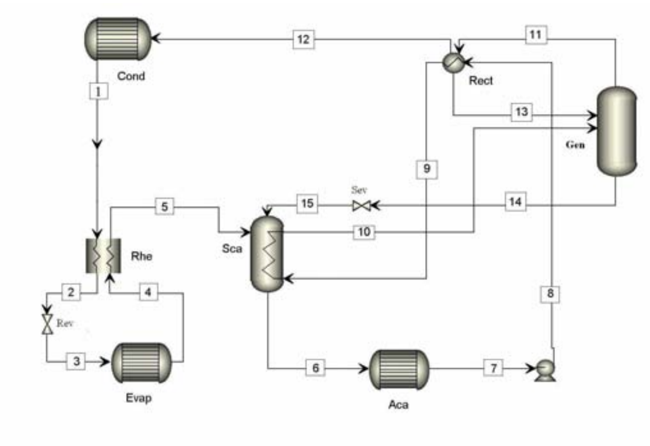

Fig.1 A schematic diagram of the ARS.

The first law is concerned with the energy conservation, and it gives no information on how, where, and how much the system performance is degraded. Only the second law can answer to these questions. However it assists in increasing of efficiency system and in reducing of system cost. In literature there are two criteria based on the second law which used to detect the irreversibility's occurred in each component of system and permit consequently the remedy to their causes, the first is entropy generated and the second is the exergy concept. These criteria are also used for comparing the performance of various systems. The exergy notion is defined by the maximum amount of work potential of a material or an energy stream in relation in to the surrounding environment

As knowing, there is no chemical reaction in the components of the ARS, the exergy (e) of stream flow can be written as:

$$e = h - T_0 S \tag{1}$$

Where T_0 is the reference temperature, h is e the enthalpy and S is the entropy of the mixture. Considering a steady state modeling of the ARS, and neglecting the pressure loss, and the variation of potential and kinetic energies in all heat exchangers and pipelines, the exergetic balance applied to fixed control volume can be written accordingly to Bejan [6] as:

$$\sum \dot{m}_i e_i - \sum \dot{m}_o e_o + \dot{Q}(1 - \frac{T_0}{T}) + \dot{W} - \dot{E}_d = 0 \tag{2}$$

3 Modeling and simulation

The thermodynamic modeling of the ARS is established by the application of the mass conservation, the energy conservation and the exergetic balances to each component of the ARS. The energetic and exergetic studies of the ARS were provided by some parameters that reflect the performance and the efficiency system. The parameters used in this work are the following:

The Coefficient of performance (COP) of ARS, which defined as ratio of the cooling produced in the evaporator of the heat input to generator. It can be written accordingly to Bejan [6] as:

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \tag{3}$$

The exergetic efficiency of ARS, which defined as the ratio of the chilled water exergy at evaporator to the exergy of the heat source at generator and can be written accordingly to Bejan [6] as:

$$\xi = \frac{-\dot{Q}_{evap}(1 - \frac{T_0}{T_e})}{\dot{Q}_{gen}(1 - \frac{T_0}{T_{gen}})} \tag{4}$$

In according to the experimental study realized by Klein [9] at 35°C air temperature, some parameters model were chosen as been showing Table 1.

Besides to these parameters, the modeling was performed to evaluate ARS with the following assumptions:

- The vapor leaving the rectifier to the condenser and the condensate returning to generator are assumed in equilibrium thermodynamic state.
- The internal heat exchange in generator between the incoming rich solution and the exiting

streams are characterized by the liquid effectiveness for the exiting weak solution and the vapor effectiveness for the exiting vapor.

- The refrigerant at the generator outlet is assumed a saturated vapor.
- The high and the low pressure used in the model have taken from the experimental study of Klein [10].

Parameter	Value
Overall heat transfer cefficient of rectifier	0.04220 (kW/°C)
Overall heat transfer coefficient of condenser	0.39564 (kW/°C)
Overall heat transfer coefficient for evaporator	1.58258 (kW/°C)
Overall heat transfer coefficient of solution cooled absorber	0.29014 (kW/°C)
Overall heat transfer coefficient of air cooled absorber	0.52752 (kW/°C)
Overall heat transfer coefficient of refrigerant heat exchanger	0.03428 (kW/°C)
Mass flow rate of pumped solution	0.02519 (kg s^{-1})
Mass flow rate of chilled water	0.45424 (kg s^{-1})
Liquid efficiency of generator	0.8
Vapor efficiency of generator	0.9
Pump efficiency	0.95

Table 1. Parameters used in the model

For more understand the modeling, appendix1 summarizes expressions that used to calculate model parameters.

From this model, an independent non linear equations system have determined and resolved with a program developed with Mathematica@6. The thermo physical proprieties of water ammonia mixture used in this work are taken from correlations provided by M. Barhoumi et al [10].

4 Results and discussion

In the part of work, we have interested in the first step on the evaluation of the performance system at the nominal condition (T_{air} =35°C).

Air temperature T_{air} = 35°C

P^t	P (MPa)	$\dot{m}$ (kgs^{-1})	T (°C)	x	y	h ($kJkg^{-1}$)	e ($kJkg^{-1}$)
1	*2.24*	*0.0097*	46.85	0.98	0	548.9	130.8
2	0.49	0.0097	28.62	0.98	0	456.4	-131.6
3	0.49	0.0097	4.420	0.97	0.99	456.4	-139.9
4	0.49	0.0097	8.630	0.84	0.99	1436.	-245.8
5	0.49	0.0097	15.30	0.72	0.99	1529.	-253.3
6	0.49	0.0252	84.81	0.26	0.91	703.7	-178.2
7	0.49	0.0252	47.99	0.42	0	119.9	-230.7
8	2.24	0.0252	48.23	0.42	0	121.1	-228.6
9	2.24	0.0252	66.16	0.42	0	201.2	-223.1
10	2.24	0.0252	109.1	0.42	0	401.3	-192.9
11	2.24	0.0104	117.6	0	0.94	1857.	-15.72
12	2.24	0.0096	95.25	0	0.98	1774.	-39.61
13	2.24	0.0008	95.25	0.52	0	369.6	-215.3
14	2.24	0.0156	126.2	0.08	0	493.1	-35.73
15	2.24	0.0156	126.4	0.08	0	494.2	-37.23

Calculated results

$COP = 0.56$	$\dot{Q}_{Aca} = 14.45\ kW$
$\dot{Q}_{evap} = 9.50\ kW$	$\dot{Q}_{Gen} = 17.132\ kW$
$\dot{Q}_{Cond} = 12.0\ kW$	$\dot{W}_{p} = 0.055\ kW$
$\dot{E}_{d,P} = 0.0033\ kW$	$\dot{E}_{d,Cond} = 0.8758\ kW$
$\dot{E}_{d,\mathrm{Rev}} = 0.0795\ kW$	$\dot{E}_{d,Sca} = 0.719035\ kW$
$\dot{E}_{d,Gen} = 1.379\ kW$	$\dot{E}_{d,\mathrm{Rect}} = 0.255618 kW$
$\dot{E}_{d,Aca} = 1.323\ kW$	$\dot{E}_{d,Sev} = 0.023295\ kW$
$\dot{E}_{d,Rhe} = 0.080\ kW$	$\dot{E}_{d,Evap} = 0.12662\ kW$

Table 2. Thermodynamic evaluation of the ARS obtained from numerical simulation

With the given parameters, the developed program calculates the thermodynamic proprieties of the mixture at all inlets and outlets of components system as showing Table 2.

At 35°C air temperature, it can be seen from table 2 that the ARS is able to produce 9.50 KW cooling capacity and it attaints a COP equal 0.56. The outlet chilled water temperature in this condition is 7.2°C.

From the same table, the simulation results based on the exergetic analysis of the ARS shows that the highest destroyed exergy is occurred in the generator (1.379kW) and in the air absorber cooled (1.323kW) respectively, whereas the lowest exergy destroyed is examined in the expansion valves, the pump solution and in the refrigerant heat exchanger. The rest of components have a significant effect on the total destroyed exergy. The sum of the destroyed exergy in the generator and solution-cooled absorber represent 55.54% of the total destroyed exergy which is remarkably a high percentage, this because these components gather a many factors which cause the irreversibility's in ARS (imperfect heat and mass transfer, mixing losses).

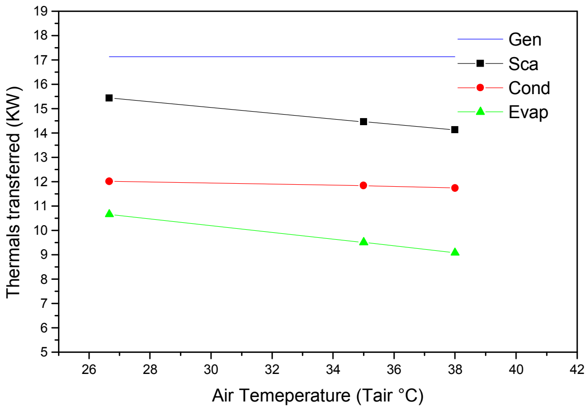

Fig.1 Variation of heat transferred by components system with air temperature

Fig.1 gives a broad view of the heat transferred by the components system with the increasing of air temperature. As knowing that the Sca and Cond are cooled by a same source (ambient air), the increasing of air temperature, increase both the condensing and the absorption temperatures and hence causes a rising of heat transferred in Sca and Cond for a fixed input heat to generator as showing Fig.1. Consequently the cooling capacity must be decrease by a consideration of the first law.

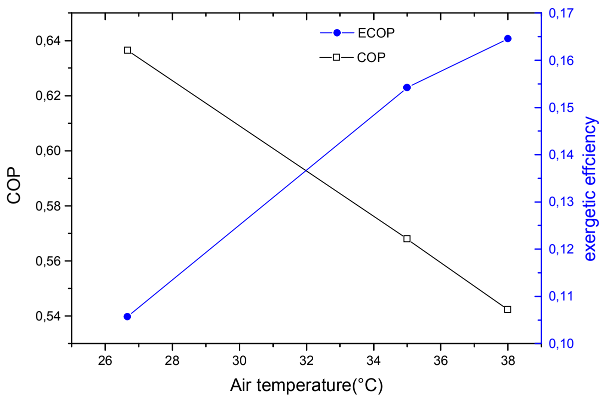

Fig.2 Variation of COP and exergetic efficiency ξ with air temperature.

Fig.2 shows the effect of the surrounding air on the performance of the ARS, it can be seen from the figure that the increasing of air temperature reduces the COP values; this is due to the decreasing of external heat transferred by the ARS with the increasing of air temperature as shown Fig.1. It also proves the effect of surrounding air on some parameters which affect the performance system. As the air temperature is increased, both the high pressure and the low pressure increase in such way and the heat rejected by Cond and Sca decrease, consequently the COP and the cooling produced rise.

Contrarily to the behavior of COP, the exergetic efficiency (ξ) goes to better with the increasing of air temperature. This proves, the ARS have a bigger potential to reduce the destroyed exergy more than that occurred when the air temperature increases. This compensates for the decrease in cooling capacity associated with the first law.

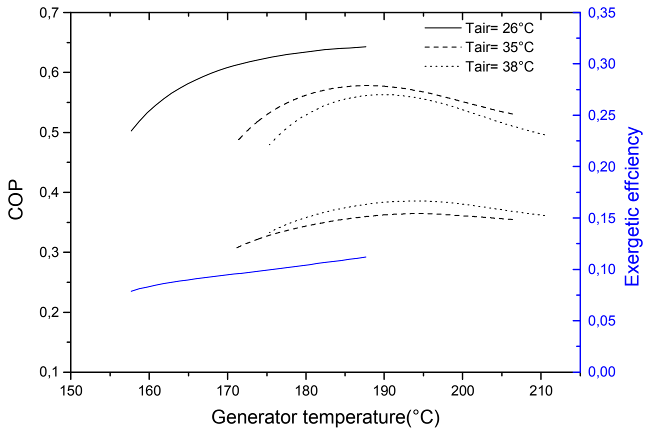

Fig.3. Variation of COP and efficiency ξ with generator temperature.

Fig.3 presents both the COP and the exegetic efficiency of system as function of generator temperature at three cases of surrounding air (T_{air} = 26°C, T_{air} = 35°C, Tair = 38°C). In the three cases, it can be observed that both the COP and the exergetic efficiency ξ have similar behaviors with the increasing of generator temperature. In fact as the generator temperature increase, as the concentration and the mass flow rate of the leaving vapor refrigerant the generator increase, and consequently the cooling capacity and the COP of system increase as be showing Fig.3, but this increase of generator temperature is not infinitely beneficial to the COP. For example, in the cases of T_{air} =35°C and T_{air} = 38°C, the COP decreases slightly and remains approximately stable above the values of generator temperature.

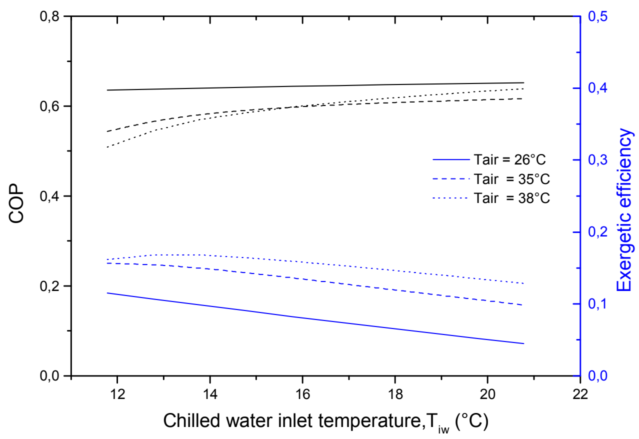

Fig.4. Variation of COP and efficiency ξ with inlet water temperature

Figure.3 shows the effect of inlet water temperature on both the COP and the exegetic efficiency ξ of ARS. As can be seen, the COP and the ξ have a dissimilar behavior with the increasing of the inlet water temperature. When the inlet water temperature increases, more cooling capacity can be produced and consequently as showing the figure the COP increases slightly. The decrease of ξ with the increasing of inlet water temperature proves that the chilled water at a lower temperature has a bigger potential to create cooling effect at a same flow rate. This compensates for the decrease in cooling capacity associated with the first law.

5 Conclusion

In this study, an exegetic analysis of an absorption refrigeration system is evaluated. From this above study, the following results can be drawn: For the water ammonia absorption refrigeration system which needs a high generator temperature to functioning, the COP goes to better with the decreasing of air temperature whereas, the exergetic efficiency goes to worse if the air temperature is reduced. The increasing of the generator temperature can improve the COP of system but as the generator temperature further increases the COP of the system levels off. This phenomenon is also is revealed on the exergetic efficiency. The increasing of the chilled water temperature increases as proven the COP as a more cooling capacity can be produced whereas, the exergetic efficiency decreases when the chilled water temperature increases. The exergetic analysis of the system shows, the most fraction of the destroyed exergy is occurred in generator at different air .The contribution of the others components system on the destroyed exergy is very small for expansion valves, refrigerant heat exchanger and solution pump. The future development of the research will focus on the improvement of the present analysis by the including of the physical characteristics of the system components and the consideration of pressure losses under the operating conditions of the ARS.

Appendix 1.

The condenser (Cond)

$$\dot{m}_{12} h_{12} - \dot{m}_1 h_1 = (UA)_{Cond} \frac{(T_{12} - T_1)}{Ln(\frac{(T_{12} - T_{air})}{(T_1 - T_{air})})} \quad (5)$$

$$\dot{Q}_{Cond} = \dot{m}_1 h_1 - \dot{m}_{12} h_{12} \quad (6)$$

$$\dot{m}_{12} e_{12} - \dot{m}_1 e_1 - \dot{E}_{d,Cond} = 0 \quad (7)$$

The refrigerant expansion valve (Rev)

$$\dot{m}_2 h_2 = \dot{m}_3 h_3 \quad (8)$$

$$\dot{m}_2 e_2 - \dot{m}_3 e_3 - \dot{E}_{d,\mathrm{Rev}} = 0 \quad (9)$$

The solution expansion valve (Sev)

$$\dot{m}_{15} h_{15} = \dot{m}_{14} h_{14} \quad (10)$$

$$\dot{m}_{14} e_{14} - \dot{m}_{15} e_{15} - \dot{E}_{d,Sev} = 0 \quad (11)$$

The refrigerant heat exchanger (Rhe)

$$\dot{m}_2 h_2 - \dot{m}_1 h_1 = \dot{m}_4 h_4 - \dot{m}_5 h_5 \quad (12)$$

$$\dot{m}_1 h_1 - \dot{m}_2 h_2 = (UA)_{Rhe} \frac{(T_2 - T_4) - (T_1 - T_5)}{Ln(\frac{(T_2 - T_4)}{(T_1 - T_5)})} \quad (13)$$

The solution cooled absorber (Sca)

$$\dot{m}_{10} h_{10} - \dot{m}_9 h_9 = (UA)_{d,Sca} \frac{(T_{15} - T_9) - (T_6 - T_{10})}{Ln(\frac{(T_{15} - T_9)}{(T_6 - T_{10})})} \quad (14)$$

$$\dot{m}_6 e_6 + \dot{m}_{15} e_{15} + \dot{m}_9 e_9 - \dot{m}_6 e_6 - \dot{m}_{10} e_{10} - \dot{E}_{d,Sca} = 0 \quad (15)$$

The solution pump

$$\dot{W}_P = \dot{m}_8 h_8 - \dot{m}_7 h_7 \quad (16)$$

$$\dot{m}_7 (e_7 - e_8) - \dot{E}_{d,P} = 0 \quad (17)$$

The evaporator (Evap)

$$\dot{m}_4 h_4 - \dot{m}_3 h_3 = (UA)_{Evap} \frac{(T_{EW} - T_3) - (T_{EW} - T_4)}{Ln(\frac{(T_{EW} - T_3)}{(T_{EW} - T_4)})} \quad (18)$$

$$\dot{m}_4 h_4 - \dot{m}_3 h_3 = \dot{m}_W C_P (T_{I,W} - T_{O,W}) \quad (19)$$

$$\dot{m}_4 (e_3 - e_4) - \dot{E}_{d,Evap} = 0 \quad (20)$$

The rectifier (Rect)

$$Q_{Rect} = (UA)_{Rect} \frac{(T_{11} - T_{10}) - (T_{12} - T_9)}{Ln(\frac{(T_{11} - T_{10})}{(T_{12} - T_9)})} \quad (21)$$

$$(\dot{m}_{11} e_{11} + \dot{m}_8 e_8) - (\dot{m}_{13} e_{13} + \dot{m}_{12} e_{12} + \dot{m}_9 e_9) = \dot{E}_{d,Rect} \quad (22)$$

The generator (Gen)

$$\dot{m}_{14} + \dot{m}_{11} = \dot{m}_{13} + \dot{m}_{10} \quad (23)$$

$$\varepsilon_{Liq} = \frac{T_{Gen} - T_{14}}{T_{gen} - T_{10}} \quad (24)$$

$$\varepsilon_{Vap} = \frac{T_{Gen} - T_{11}}{T_{Gen} - T_{10}} \quad (25)$$

$$\dot{Q}_{Gen} = \dot{m}_{14} h_{14} + \dot{m}_{11} h_{11} - \dot{m}_{13} h_{13} - \dot{m}_{10} h_{10} \quad (26)$$

$$(\dot{m}_{10} e_{10} + \dot{m}_{13} e_{13}) - (\dot{m}_{11} e_{11} + \dot{m}_{14} e_{14}) + (1 - \frac{T_0}{T_{Gen}}) \dot{Q}_{Gen} = \dot{E}_{d,Gen} \quad (27)$$

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