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Energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling

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ABSTRACT

The objective of this work is to present an energy and exergy analysis of a novel configuration of absorption cooling system operating at low enthalpy sources. The double stage cycle developed in the present work is operating with water-ammonia. In this investigation, modeling and simulation of the proposed configuration is attempted. Also, a thermodynamic model based on the energy and exergy balances is developed. The obtained numerical results obtained have been compared with those corresponding to the conventional machine. Great emphasis is given to the estimation of the refrigeration systems' performance, the exergy efficiency, the global exergy destruction in the system and the exergy destruction in each of the main components. The analyzed parameters are the coefficient of performance (COP), the irreversibility and the exergetic efficiency. The results of the study reveal that the performance of the novel configuration is better than that of the two stage conventional configuration. Besides, it allows a lower operating temperature, about 60–120 °C instead of 100–160 °C for the conventional cycle.

Introduction

The cooling and refrigeration cycles are mostly based on mechanically driven vapor compression. The cooling demand in countries with a hot climate leads to a peak in electricity consumption; consequently, the use of alternative technologies should be encouraged. One possibility consists in the modification of absorption cycles [1]. Their principal advantages compared to mechanically driven compression cycles are summarized to the following: a) no contribution to the destruction of the ozone layer and to the global warming effect because of the natural refrigerants use, b) little energy consumption, because the compression cycles are thermally driven (Herold et al., 1996; Ziegler, 2002) [2,3] and c) absence of moving parts in, some circulating pumps. Absorption cycles use a working couple consisting of a refrigerant and an absorbent. In generally being water-lithium bromide, (LiBr), or ammonia-water. The basic absorption cycle structure is the single effect, having four basic components: absorber, generator, evaporator and condenser. Absorption refrigerators are commercially available and perform stable operation under part-load conditions, but their coefficient of performance (COP) values are relatively low compared to vapor

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compression refrigerators (Lee SF and Sherif SA, 2001) [4]. However, combined cycles of vapor compression–absorption refrigeration system can provide high COP. Several works on combined cooling system or absorption refrigerator (mainly on the cooling performance analysis and optimization) have been carried out (Lee SF and Sherif SA, 2001; Arora and S.C.Kaushik, 2009) [4,5]. In general, performance analysis of these systems is investigated using energy analysis method, based only on the first law of thermodynamics (energy balance) by means of the coefficient of performance (COP). Unfortunately, this approach is of limited use in view of the fact that it fails to make out the real energetic losses in a refrigerating system. For example, it does not identify any energetic losses occurring during the throttling process though there is a potential pressure drop and this can be predicted only through entropy or exergy analysis. Distinction between reversible and irreversible processes was first introduced in thermodynamics through the concept of 'entropy' (Dincer and Cengel, 2001) [6]. Thus, in contrast to energetic approach, the exergy analysis, which takes into account both the first and the second thermodynamics laws, assists the evaluation of the magnitude of the available energy losses in each component of the refrigeration system and the worth of energy from a thermodynamic point of view. In thermal design decisions, utilisation of the second law of thermodynamics is very well referenced (Bejan, 1994, 1995, 1996) [7–9]. In addition, the exergy analysis allows explicit presentation and improved comprehension of thermodynamic processes by quantifying the effect of irreversibility occurring in the system along with its location. Some studies have carried out exergy analysis (Lee SF and Sherif, 1999; .Ravikumar et al., 1998) [10,11] pertaining to single, double and multiple-effect absorption refrigerating systems that usie $LiBr/H_2O$ or NH_3/H_2O (Anand and Kumar, 1987) [12], in these three last references was carried out irreversibility analysis of single and double-effect systems under the following conditions: condenser and absorber temperatures 37.81 °C, evaporator temperature 7.21 °C and generator temperature 87.81 °C for the single-effect and 140.61 °C for the double-effect system. In these studies, there was neither computed the optimum generator temperature nor calculated the exergetic efficiency for the operation of series flow double-effect system. (Lee and Sherif, 1999) [10], have presented the second law analysis of various double-effect lithium bromide–water absorption chillers and computed the COP and the exergetic efficiency as well. It is obvious from literature that exergy investigation as regards compression–absorption heat pumps has not been carried out. This motivates the present investigation.

In the present study, energy and exergy analysis of a novel two stages hybrid heat pump based on NH3/H2O, has been carried out. All energetic and exergetic results are compared to those of the two stages absorption heat pump. The analysis also brings out the effects of generators, absorbers, evaporator, condenser, compressor and solution heat exchangers on the various performance parameters. The effects of the compressor discharge pressure and the generator temperature on system performances are examined. Exergy loss of each component of the heat pump was evaluated for several working conditions.

Heat pump cycle description

The heat pump, subject of this study, is a combination between the two conventional absorption stages (two absorbers, two generators, condenser and evaporator) and the compression one. A compressor is injected into the cycle, upstream the absorption part, in order to ameliorate the absorption process as was brought by Bouaziz et al. (Bouaziz et al., 2011) [13] (Fig. 1).

The system works above three pressure levels. The vapor refrigerant coming from the first generator (6) with intermediate pressure (P_1) is compressed by an isentropic transformation (6C) to an intermediate pressure (P_2) and then, it is reinserted into the second absorber. The rich solutions from absorbers (2) and (7) are heated by the poor solution originating from the generators (4) and (9) via heat exchangers inter-solution. The condenser and the second generator operate at the third pressure level (P_{CD}).

This installation has two generators operating at the same temperature (T_{GE}), two absorbers and a condenser working at the same temperature (T_{CD}) besides an evaporator and inter-solutions heat exchangers.

The evaporator and the first absorber (AB1) operate at the same pressure (P_{EV}), the first generator (GE1) operates at higher pressure (P1) which is increased by the compressor, so the second absorber operates at a second intermediate pressure (P2). Finally, the second generator and the condenser are operating at the highest pressure (P_{CD}).

Energy and mass balances

The mass balance for the two stages, governing the three present substances: weak solution, rich solution and refrigerant gas gives:

$$\dot{m}_{NH3} = \dot{m}_{NH3i} \tag{1}$$

The rich and poor solution flow rates are given by Equations (2) And (3):

$$\dot{m}_{SRi} = f_i \dot{m}_{NH3i} \tag{2}$$

$$\dot{m}_{SPi} = (f_i - 1)\dot{m}_{NH3i} \tag{3}$$

Energy balance for each installation component is presented by Equations (4)–(9):

$$\dot{Q}_{CD} = \dot{m}_{NH3}(h_{12} - h_{11}) \tag{4}$$

$$\dot{Q}_{EV} = \dot{m}_{NH3}(h_1 - h_{12}) \tag{5}$$

$$\dot{Q}_{GE1} = (f-1)\dot{m}_{NH3}h_4 + \dot{m}_{NH3}h_6 - fh_3 \tag{6}$$

$$\dot{Q}_{GE2} = (f-1)\dot{m}_{NH3}h_9 + \dot{m}_{NH3}h_{11} - fh_9 \tag{7}$$

$$\dot{Q}_{AB1} = fh_2 - (f-1)\dot{m}_{NH3}h_5 - \dot{m}_{NH3}h_1 \tag{8}$$

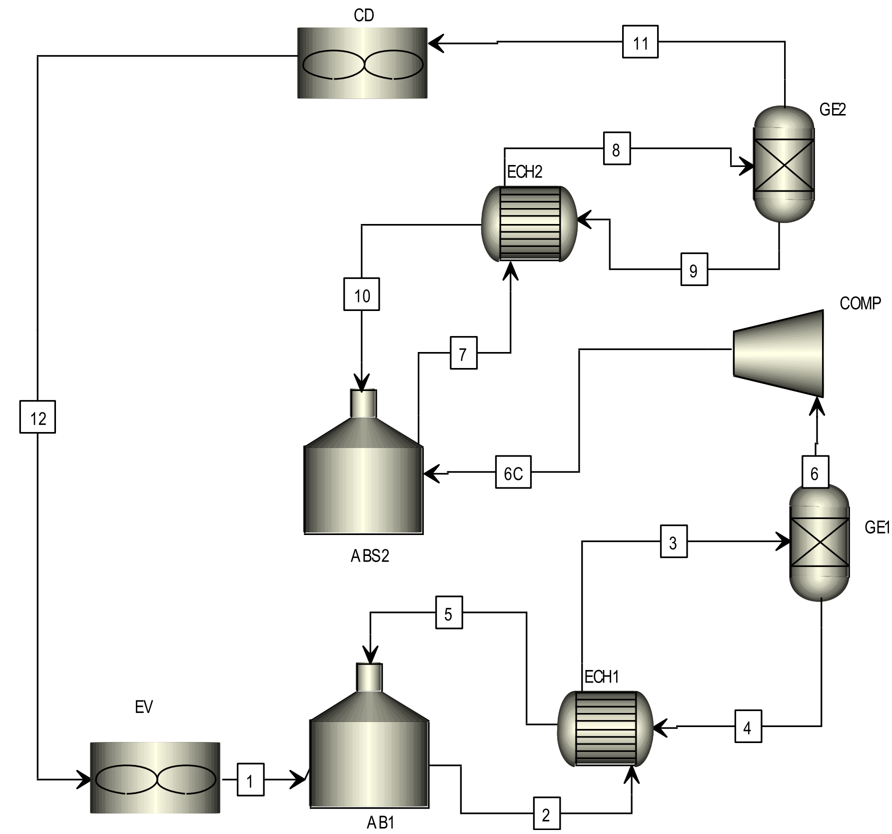

Fig. 1 — Two-stage hybrid cycle.

$$\dot{Q}_{AB2} = fh_7 - (f-1)\dot{m}_{NH3}h_{6C} - \dot{m}_{NH3}h_{10} \tag{9}$$

For an isentropic process, Laplace relation gives:

$$T_{comp-in}P_{comp-in}^{\frac{1-k}{k}} = T_{comp-out}P_{comp-out}^{\frac{1-k}{k}} \tag{10}$$

where:

$T_{comp-in}$, $P_{comp-in}$ and, $T_{comp-out}$, $P_{comp-out}$ are the compressor temperature and pressure at the inlet and outlet, respectively.

Under the assumption of isentropic processes (ideal case), the consumed power is given by:

$$\dot{Q}_{is} = \dot{m}_{NH3}Cp_{NH3}\left(T_{comp-out} - T_{comp-in}\right) \tag{11}$$

Taking into account the isentropic efficiency η_{is}, the real power is given by:

$$\dot{Q}_{real} = \frac{\dot{Q}_{is}}{\eta_{is}} \tag{12}$$

$$\dot{Q}_{real} = \dot{m}_{NH3}(h_{6C} - h_6) \tag{13}$$

By combining Equations 12 and 13 with Equations (4) and (5), the value of the steam enthalpy at the compressor outlet is deduced:

Were the isentropic efficiency η_{is} is given by (Bouaziz et al., 2011; Brunin et al., 1997) [13,14]:

$$\eta_{is} = 0.874 - 0.0135.\tau \tag{14}$$

with

$$\tau = \frac{P_{comp-out}}{P_{comp-in}} \tag{15}$$

Coefficient of performance (COP) is given by the following expression [15,16]:

$$COP = \frac{\dot{Q}_{EV}}{\left(\dot{Q}_{GE1} + \dot{Q}_{GE2} + \dot{Q}_{comp}\right)} \tag{16}$$

Exergy balance

As is well known exergy is the measure of useful work or potential of a stream to cause change. Besides it is an effective measure of the potential of a substance to impact the environment (Ziegler, 2002; Gungor et al., 2013) [3,17].

The exergy balance for a control volume undergoing steady-state process is expressed as (Lee SF, Sherif SA, 2001) [4]:

$$Ex_{Di} = \sum(\dot{m}Ex)_{in} - \sum(\dot{m}Ex)_{out} \pm \sum\left(Ex_{\dot{Q}}\right) \pm \sum\dot{W} \tag{17}$$

Where $Ex_{\dot{Q}}$ is the thermal exergy and expressed as follow (Chen and Z.H, 2007; Chen and, Chen, 2006) [18,19]:

$$Ex_{\dot{Q}} = \dot{Q}\left(1 - \frac{T_0}{T}\right) \quad (18)$$

Ex_{Di} represents the rate of exergy destruction, also called irreversibility, occurring in the process in the component i.

The first and second terms on the right-hand side of Equation (17) represent the exergy of streams entering and leaving the control volume. The third term represents the exergy associated with heat transfer $\dot{Q}$ from the source maintained at a constant temperature T which is equal to the work obtained by the Carnot engine operating between T and T_0, and is therefore equal to the maximum reversible work that can be obtained from heat energy $\dot{Q}$. The last term is the mechanical work transferred to or from the control volume.

$$Ex_{DT} = \sum Ex_{Di} \quad (19)$$

Also, the exergy loss can be expressed in terms of exergetic efficiency; which is the rate between the inlet exergy and the outlet exergy (Herold et al., 1996; P.Kumar Satapathy, 2008; Koroneos and Rovas, 2013) [2,20,21].

$$\eta_{ex} = \frac{outlet.system.exergy}{inlet.system.exergy} \quad (20)$$

The exergy destruction in each component of the hybrid cycle is given by Refs. [22–24]:

$$Ex_{D.CD} = \dot{m}_{NH3}(h_{11} - Ts_{11}) - \dot{m}_{NH3}(h_{12} - Ts_{12}) + \dot{Q}_{CD}\left(1 - \frac{T_0}{T_{CD}}\right) \quad (21)$$

$$Ex_{D.EV} = \dot{m}_{NH3}(h_{12} - Ts_{12}) - \dot{m}_{NH3}(h_1 - Ts_1) + \dot{Q}_{EV}\left(1 - \frac{T_0}{T_{ev}}\right) \quad (22)$$

$$Ex_{D.AB1} = \dot{m}_{NH3}[h_{6C} - Ts_{6C} + (f-1)(h_5 - Ts_5) - f(h_2 - Ts_2)] - \dot{Q}_{AB1}\left(1 - \frac{T_0}{T_{CD}}\right) \quad (23)$$

$$Ex_{D.AB1} = \dot{m}_{NH3}[h_{6C} - Ts_{6C} + (f-1)(h_{10} - Ts_{10}) - f(h_7 - Ts_7)] - \dot{Q}_{AB2}\left(1 - \frac{T_0}{T_{CD}}\right) \quad (24)$$

$$Ex_{D.GE1} = \dot{m}_{NH3}[f(h_3 - Ts_3) - (f-1)(h_6 - Ts_6) - (h_4 - Ts_4)] + \dot{W}_{GE1} \quad (25)$$

$$Ex_{D.GE2} = \dot{m}_{NH3}[f(h_8 - Ts_8) - (f-1)(h_9 - Ts_9) - (h_{11} - Ts_{11})] + \dot{W}_{GE2} \quad (26)$$

$$Ex_{D.comp} = \dot{m}_{NH3}(h_6 - Ts_6) - \dot{m}_{NH3}(h_{6C} - Ts_{6C}) + \dot{W}_{COMP} \quad (27)$$

Exergetic efficiency of this machine is so given by Refs. [22–24]:

$$\eta_{ext} = \frac{\dot{Q}_{EV}\left(1 - \frac{T_0}{T_{ev}}\right)}{\dot{W}_{comp} + \dot{W}_{GE1} + \dot{W}_{GE2} + \dot{Q}_{CD}\left(1 - \frac{T_0}{T_{CD}}\right)} \quad (28)$$

$$\dot{I}(\%) = \frac{Ex_{Di}}{Ex_{DT}} \quad (29)$$

Assumptions

Several assumptions were taken into account in the exergetic study:

- Kinetic and Potential exergy are neglected.
- All transformations are in a steady state.
- Pressure and heat losses in the system component are neglected.
- The exchange temperature is the input and the output logarithmic mean temperature. - The reference temperature and pressure P_0 and T_0 are 1 atm and 25 °C, respectively.

Results and discussion

In this study, simulation was done by Aspen Tech software, which is an integrated flow sheeting environment for sequential-modular and equation-oriented simulation and optimization with a widespread database available for the physical and chemical properties of substances (AES, version 11.1, 2001) [25]. For the determination of thermodynamic properties of water/Ammonia, the flow sheet simulator Aspen Properties was used. Mixture binary parameters were determined by ELECTNRTL method existing in Aspen. In the following investigation, a comparative study of the COP and the exergy destruction of the conventional and the novel configuration has been carried out.

Comparison between the two-configurations

Fig. 2 reveals a coefficient of performance (COP) comparison between the double effect novel configuration and the

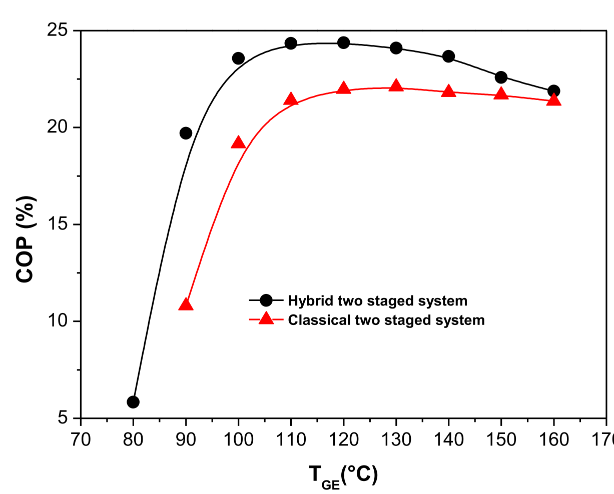

Fig. 2 – COP evolution versus TGE for the conventional cycle and the two novel configurations.

conventional one. The COP of novel configuration is about 24% and it is better than that of the conventional one (21%). Furthermore, the proposed modification allows lower threshold temperatures, it is about 90 °C for the first one and the compressor addition lets it down to 80 °C. The hybrid cycle is more suitable to low energy sources.

In Fig. 3, there is an irreversibility comparison between the known and commercialized system and the modified proposed system. Irreversibility in the novel configuration seems to be limited by about 5 kW.

First results show that such cycle is promising, comparing with the classical one, and should be further analyzed. In the rest of this section all exergetic and energetic performances of the hybrid two staged system are investigated.

Novel configuration analysis results

According to Fig. 4, COP of the proposed machine increases when the condensation temperature decrease and it achieves an optimum point for each value of T_{CD}. This optimum corresponds to $T_{GE} = 100$ °C for a condensation temperature about 45 °C and $T_{GE} = 70$ °C for $T_{CD} = 30$ °C. The increase of the condensation temperature increases the threshold temperature of the system. The increase of COP values with the decrease of the condensation temperature is not linear, the increase narrows for highest condensation temperatures. The system can reach a COP of about 32% at a condensation temperature of 30 °C and a generator temperature of 70 °C, this is promising for the use of low energy sources like solar energy.

Fig. 5 details exergy destruction evolution versus condensation temperature conditions. This parameter falls when T_{CD} declines and it achieves an optimum that coincides with the optimum coefficient of performance (COP). For low generator temperatures, exergy loss is greater than that at high temperatures, it can achieve 1.8 kW at the threshold temperature, this is due to the high circulation factor at these temperatures which is responsible of the rising of rich and poor solutions flow rates in the cycle.

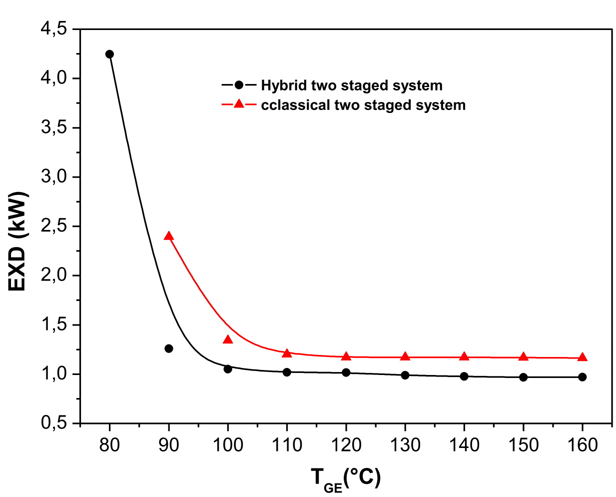

Fig. 3 – Exergy destruction evolution versus T_{GE} for the conventional cycle and the novel configuration.

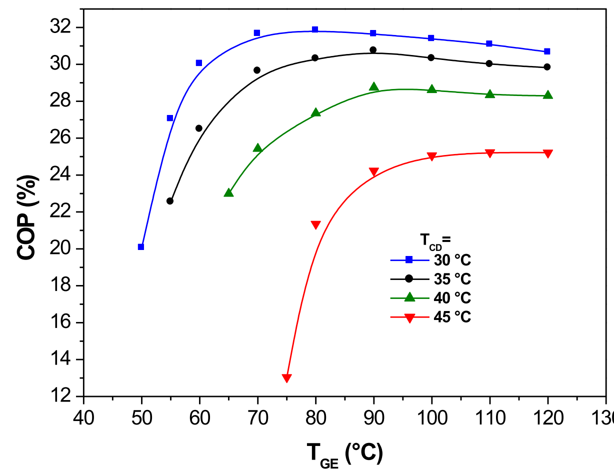

Fig. 4 – COP, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C and $P_2 = 900$ kPa.

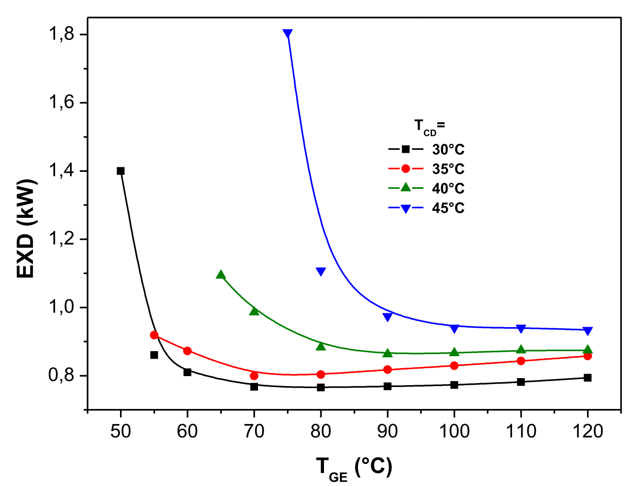

Fig. 5 – Irreversibility, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C and $P_2 = 900$ kP.

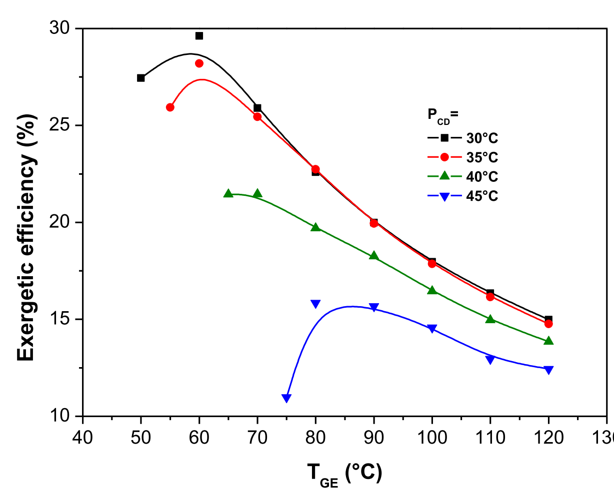

Fig. 6 – Exergy efficiency, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C for different T_{CD}.

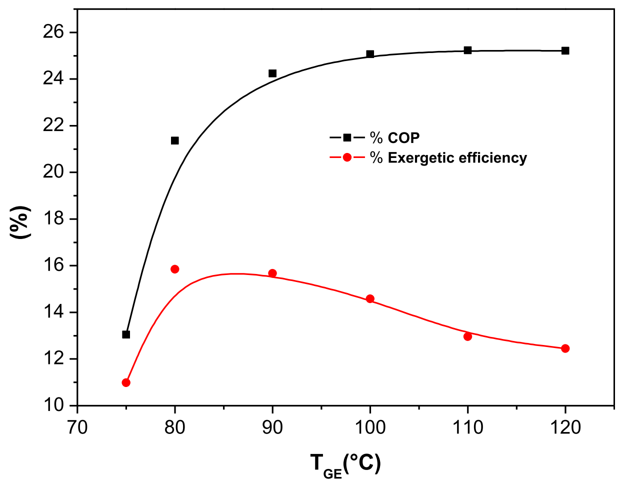

Fig. 7 – COP and exergetic efficiency, of the hybrid two staged system, behavior versus T_{GE} with $T_{ev} = -10$ °C and $T_{CD} = 45$ °C.

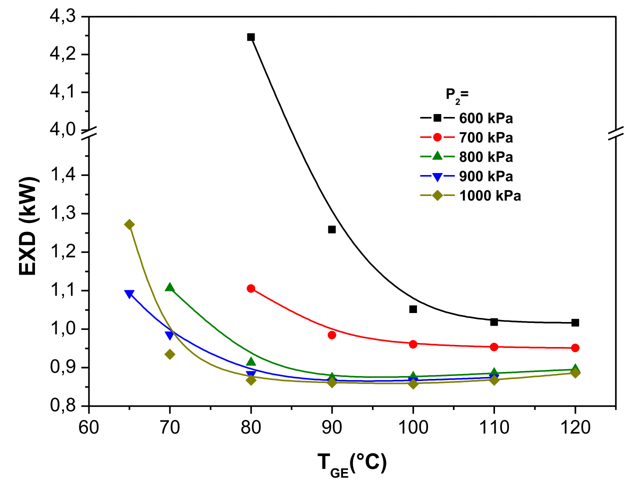

Fig. 9 – Exergy destruction, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C and $P_{CD} = 40$ °C.

Fig. 6 depicts the variation of exergetic efficiency for various temperature conditions. The observed first important result is that exergetic efficiency is growing when the condensation temperature is reduced, this was expected if we consider the two precedent results. Second significant feature is that exergetic efficiency is lower for smaller or higher generator temperature. Those curves have an increasing first part until an optimum point then it begins to decrease when T_{GE} increases. These curves let to realize that exergetic performance of the system are down at high generator temperatures even if the COP is right. In Fig. 7 are illustrated, simultaneously, both of the COP and exergetic efficiency allures for the worse condensation temperature condition. It comes out from Fig. 7 that while the COP continues to rise, exergetic efficiency falls when achieving a generator temperature about 80 °C. By this fact, choosing a low operating temperature will not only offer a gain at the generator electric work but also a minimum exergy loss.

Figs. 8–10 illustrate the evolution of the system performances along the progression of the intermediate pressure P_2 (compressor outlet pressure).

As shown in Fig. 8, the COP is improved when the intermediate pressure increases and this progression narrows in high pressures (progression between 600 and 700 KPa is better than progression between 900 and 1000 KPa). Moreover, increasing intermediate pressure offers a decrease in the threshold temperatures, most desired results. As well as COP, irreversibility is reduced when rising P_2. All curves present an optimum point which falls also with the increase of P_2. Threshold temperature can be reduced by 25 °C by increasing the intermediate pressure P_2 from 600 kPa to 1000 kPa.

Fig. 9 expand exergetic efficiency evolution versus the pressure P_2. It increases with the growth of the compression marge until a pressure P_2 of 900 kPa, further increasing this pressure does not increase the exergetic efficiency of the system. Same behavior of the exergetic efficiency, deduced in

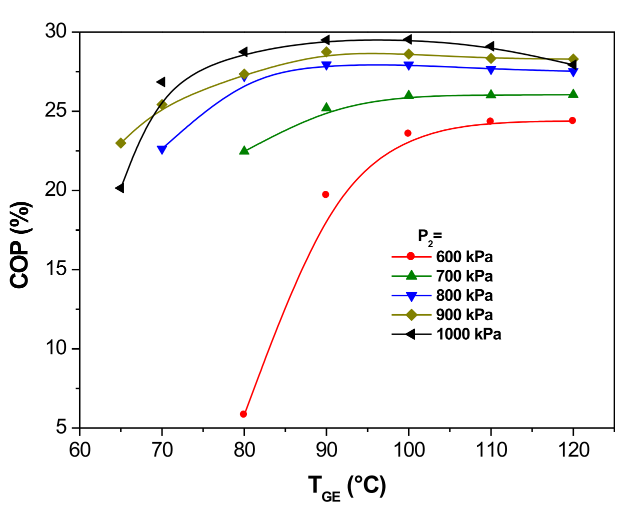

Fig. 8 – COP, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C and $P_{CD} = 40$ °C.

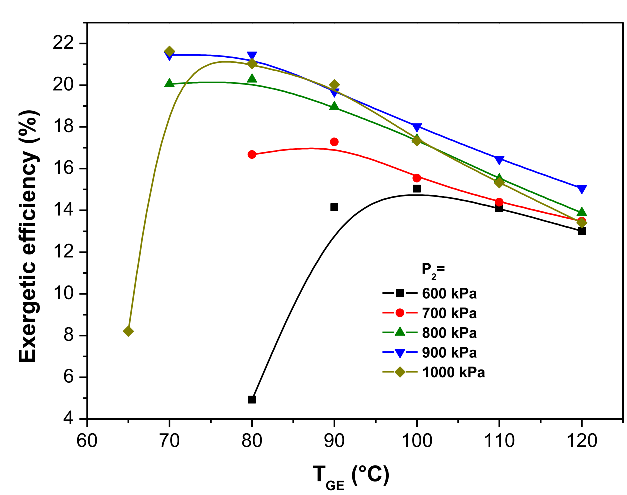

Fig. 10 – Exergy efficiency, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C for different P_2.

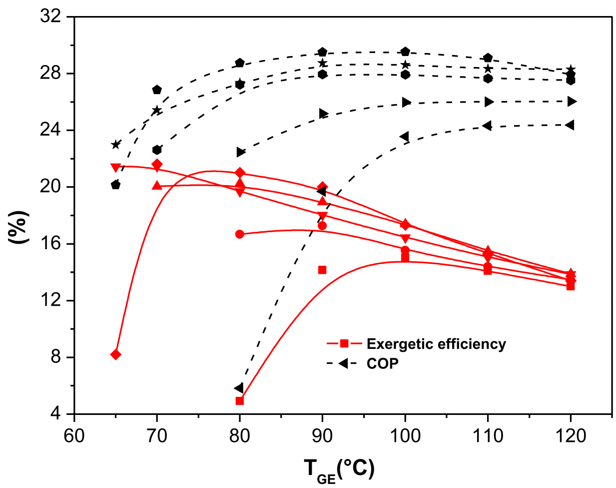

Fig. 11 – Exergy efficiency and COP, of the hybrid two staged system, evolution versus T_{GE} with $T_{ev} = -10$ °C for different P_2.

Fig. 6, is drawn from this figure: This parameters drops at high generator temperatures even if changing intermediate pressure. Fig. 11 illustrates what was said before about exergetic efficiency behavior via the COP one. But the new relevant results that is revealed by this figure is that exergetic efficiency optima occurs before the coefficient of performance ones, there is a shift of about 10 °C between them for example: for an intermediate pressure of 700 kPa: COP optima is achieved at 100 °C while the exergetic efficiency one is achieved at only 90 °C. Such result is difficult to pick out or to predict without such comparison.

Conclusions

A numerical study has been achieved to predict the performance of a double staged absorption/compression hybrid system. An energetic and exergetic investigation has been developed on the classical cycle and on a new hybrid proposed cycle of the system. The main results are:

- The increase in the generator temperature causes an elevation of the COP in both hybrid and real systems. However, at higher generator temperatures the COP curve stabilizes. The COP of the hybrid cycle is nearly 25–32% greater than that for the classical cycle. The maximum values of the COP are reached at lower generator temperature for the hybrid proposed cycle between 70 and 110 °C.
- The increase of condensation temperature decreases the COP of the hybrid system and increases the threshold temperature. The influence of increasing compressor discharge pressure is to increase the COP values and to decrease the threshold temperature which can reach 70 °C and then it becomes possible to use low energy sources.
- The exergy loss in the proposed system is relatively lower than that in the present system. Optimum value of exergy loss for the hybrid system are between 0,8 and 1 kW, while it is about 1.3 kW for the classical system at the same condensation temperature.
- Exergetic efficiency decreases when increasing condensation temperature. It also decreases with the decrease of compressor discharge pressure until 900 kPa then it falls down significantly. The difference between COP and exergetic efficiency behavior is that the later decreases significantly after reaching its maximum value while COP stabilizes. This means that better exergetic performances are achieved at low temperature and pressure and further increasing these parameters after optimum COP has a negative influence on energy quality.

Nomenclature

COP	Coefficient of performance
P	Pressure, Pa
T	Temperature, K, °C
x	mass fraction
Ex_D	Exergy destruction
$\dot{m}$	Mass flow rate, kgs^{-1}
$\dot{W}$	Work transfer rate, W
$\dot{Q}$	Heat transfer rate, W
Ex	Specific exergy of a stream, $kJkg^{-1}$
h	Specific enthalpy of a stream, $kJkg^{-1}$
s	Specific entropy of a stream, $kJkg^{-1}\ K^{-1}$
f	Entrainment factor
η_{ex}	Exergetic efficiency
η_{is}	isentropic efficiency
$\dot{I}$(%)	Irreversibility (percentage of exergy destruction of a component)

Subscripts

i	Component i
T	Total
0	Reference
2	intermediate
EV	Evaporator
COMP	Compressor
GE	Generator
ECH	Solution exchanger
AB	Absorber.

REFERENCES

[1] Bouaziz N, Ben Iffa R, Kairouani L. Performance of a water ammonia absorption system operating at three pressure levels. J Mech Eng Res 2011;3(4):120–7.
[2] Herold K, Radermacher R, SA K. Absorption chillers and heat pumps. Boca Raton, Florida: CRC Press; 1996.
[3] Ziegler F. State of the art in sorption heat pumping and cooling technologies. Int J Refrig 2002;25:450–9.
[4] Lee SF, Sherif SA. Second law analysis of various double effect lithium bromide/water absorption chillers. 2001. ASHRAE Transactions; AT-01-9-5:664–673.
[5] Arora A, Kaushik SC. Theoretical analysis of LiBr/H_2O absorption refrigeration systems. Int J Energy Res 2009;33:1321–40.
[6] Dincer I, Cengel YA. Energy, entropy and exergy concepts and their roles in thermal engineering. Entropy 2001;3:116–49.

[7] Bejan A. Entropy generation through heat and fluid flow. 2nd ed. New York: Wiley; 1994.
[8] Bejan A. Convection heat transfer. New York: Wiley; 1995.
[9] Bejan A. Entropy generation minimization. Boca Raton, FL: CRC Press; 1996.
[10] Lee SF, Sherif SA. Second law analysis of multi effect lithium bromide/water absorption chillers. 1999. ASHRAE Transactions; Ch-99-23-3:1256–1266.
[11] Ravikumar TS, Suganthi L, Samuel AA. Exergy analysis of solar assisted double effect absorption refrigeration system. Renew Energy 1998;14(Nos. 1–4):55–9.
[12] Anand DK, Kumar B. Absorption machine irreversibility using new entropy calculations. Sol Energy 1987;39(3):243–56.
[13] Bouaziz N, BenIffa R, Nehdi E, Kairouani L. Conception of an absorption refrigerating system operating at low enthalpy sources. In: Moreno-Piraján Juan Carlos, editor. Thermodynamics – systems in equilibrium and non-equilibrium; 2011. ISBN: 978-953-307-283-8, InTech.
[14] Brunin O, Feidt M, Hivet B. Comparison of the working domains of some compression heat pumps and a compression-absorption heat pump. Int J Refrig 1997;20:308–18. 417.
[15] Figueredo GR, Bourouis M, Coronas A. Thermodynamic modelling of a two-stage absorption chiller driven at two-temperature levels. Appl Therm Eng 2008;28:211–7.
[16] Venegas M, Izquierdo M, de Vega M, Lecuona A. Thermodynamic study of multistage absorption cycles using low-temperature heat. Int J Energy Res 2002;26:775–91.
[17] Gungor Afsin, Bayrak Mustafa, Beylergil Bertan. In view of sustainable future energetic-exergetic and economic analysis of a natural gas cogeneration plant. Int J Exergy 2013;12(No. 1):109–18. 436.
[18] Chen GQ, Qi ZH. Systems account of societal exergy utilization: China 2003. Ecol Model 2007;208:102–18. 438.
[19] Chen B, Chen GQ. Exergy analysis for resource conversion of the Chinese Society 1993 under the material product system. Energy 2006;31:1115–50. 440.
[20] Kumar Satapathy P. Exergy analysis of a compression–absorption system for heating and cooling applications. Int J Energy Res 2008;32:1266–442. 1278.
[21] Koroneos Christopher, Rovas Dimitrios. Exergy analysis of geothermal electricity using the Kalina cycle. Int J Exergy 2013;12(No. 1):54–69.
[22] Arora A, Kaushik SC. Energy and exergy analyses of a two-stage vapor compression refrigeration system. Int J Energy Res 2009;34:907–23.
[23] Gong G, Zeng W, Chang S, He J, Li K. Scheme-selection optimization of cooling and heating sources based on exergy analysis. Appl Therm Eng 2007;27:942–50.
[24] Pons M, Kodama A. Entropic analysis of adsorption open cycles for air conditioning. Part 1: first and second law analyses. Int J Energy Res 2000;24:251–62.
[25] AES, version 11.1, 2001. Aspen Aspen Technology, Inc.,Ten Canal Park Cambridge, MA 02141–2201,USA. www.aspentech.com.