18èmes Journées Internationales de Thermique (JITH 2017)
Monastir (Tunisie), 25 - 27 Octobre 2017

An improved cycle for low grade heat cooling application in double-lift absorption/compression system using organic fluid mixtures

Rabeb TOUJANI[a], Dorra LOUNISSI[a*], Nahla BOUAZIZ[a,b]

[a] *Unite de Recherche Energetique et Environnement, Ecole Nationale des Ingenieurs de Tunis, Universite Tunis EL Manar, BP, 37, Le BELVEDERE, 1002 Tunis, Tunisia*
[b] *Universite Tunis EL Manar, Institut Preparatoire aux etudes d'Ingenieurs d'EL Manar, 2092 Tunis El Manar,Tunisia*

dorra.lounissi@gmail.com
nahlabouaziz@yahoo.fr
rabeb.toujani@gmail.com

Abstract : At present, much interest is being shown in hybrid absorption refrigeration cycles driven by low temperature heat sources, such as solar energy or low-grade waste-heat. In the present work, an exergetic investigation of a novel configuration double-lift absorption/compression refrigeration system is developed. The potential of the organic fluid mixtures 1,1,1,3,3,3-Hexafluoropropane/Dimethylacetamide (R236fa/DMAC) in the low grade heat refrigeration field is discussed. In this study, modeling and simulation of the proposed configuration is attempted. The ammonia-water system was used for comparison purposes. The results show that the performances of the proposed configuration are improved significantly when using the above mentioned organic mixture. Coefficient of performance of the cycle with R236fa/DMAC is 15% higher than with ammonia/water. Exergetic performances of the system are also improved by about 10% with the use of the proposed new mixture. The most important result given in this study is that the above system operates at low generator temperatures between 50°C and 80°C which makes it a good alternative of the ammonia-water systems in the low grade heat refrigeration field.
Keywords : Compression/Absorption, exergy analysis, organic absorbent, hybrid heat pump

1. 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 compression refrigerators (Lee SF and Sherif SA, 2001) [4]. However, combined cycles of vapor compressioneabsorption 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 singleeffect 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 bromideewater absorption chillers and computed the COP and the exergetic efficiency as well. It is obvious from literature that exergy investigation as regards compressioneabsorption 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.

2. Heat pump presentation

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] (Figure1).

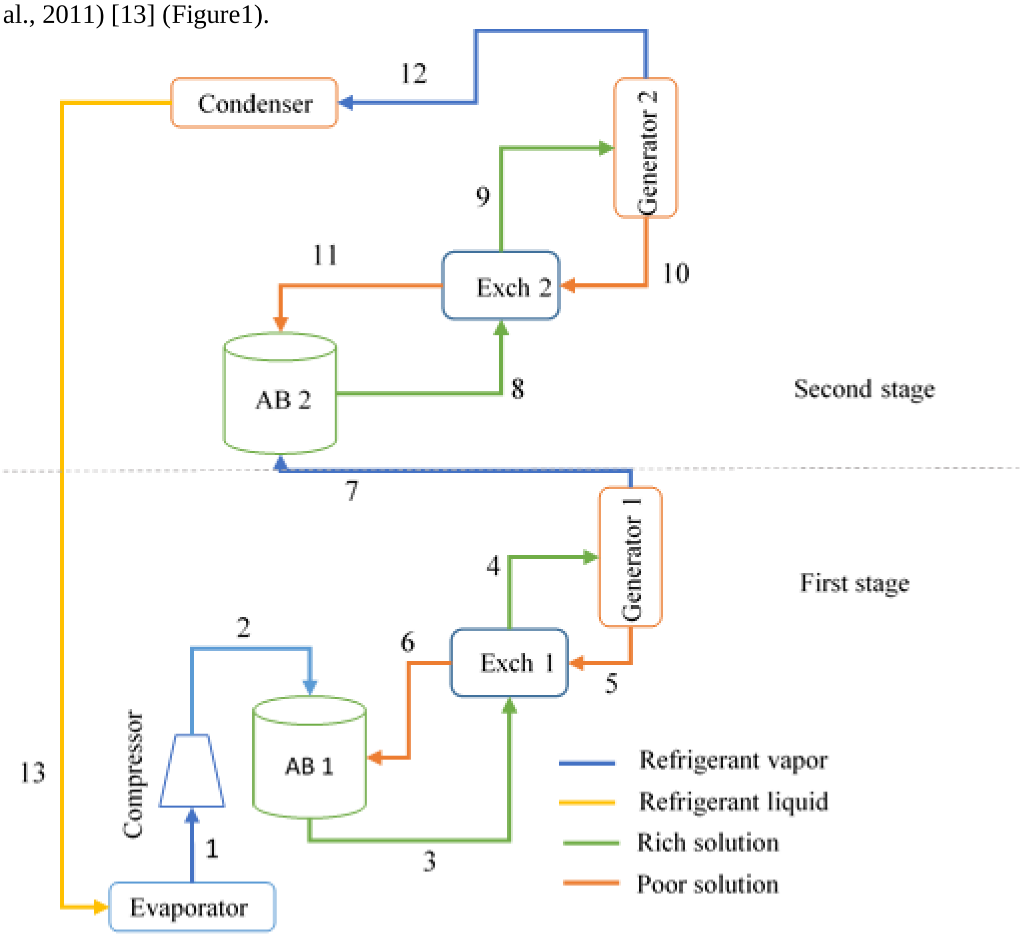

Figure 1: New hybrid double stage absorption heat pump

The system works above three pressure levels. The vapor refrigerant coming from the evaporator with low pressure (1) is compressed by an isentropic transformation (2) to an intermediate pressure (P_2) and then, it is inserted into the absorber. The rich solutions from absorbers (3) and (8) are heated by the poor solution originating from the generators (5) and (10) 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 intersolutions heat exchangers.
The evaporator operates at the low pressure (P_{EV}) which is increased by the compressor to the absorber (1) pressure (P_{comp}), the first generator (GE1) operates at higher pressure (P1), so the second absorber operates at the same pressure (P1). Finally, the second generator and the condenser are operating at the highest pressure (P_{CD}).

3. 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 component of the system is presented by equations (4) to (9):

$$\dot{Q}_{CD} = \dot{m}_{NH3}.(h_{13} - h_{12}) \tag{4}$$

$$\dot{Q}_{EV} = \dot{m}_{NH3}.(h_1 - h_{13}) \tag{5}$$

$$\dot{Q}_{GE1} = (f-1).\dot{m}_{NH3}.h_5 + \dot{m}_{NH3}.h_7 - f.h_4 \tag{6}$$

$$\dot{Q}_{GE2} = (f-1).\dot{m}_{NH3}.h_{10} + \dot{m}_{NH3}.h_{12} - f.h_9 \tag{7}$$

$$\dot{Q}_{AB1} = f.h_3 - (f-1).\dot{m}_{NH3}.h_6 - \dot{m}_{NH3}.h_2 \tag{8}$$

$$\dot{Q}_{AB2} = f.h_8 - (f-1).\dot{m}_{NH3}.h_{11} - \dot{m}_{NH3}.h_7 \tag{9}$$

For an isentropic process, Laplace relation gives:

$$T_{comp-in}.P_{comp-in}^{(1-k)/k} = T_{comp-out}.P_{comp-out}^{(1-k)/k} \tag{10}$$

where: T_{compin}, P_{compin} and, $T_{compout}$, $P_{compout}$ 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:

$$\dot{Q_{\iota s}} = \dot{m}_{NH3}.cp_{NH3}.\left(T_{comp-out} - T_{comp-in}\right) \tag{11}$$

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

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

$$\dot{Q}_{Real} = \dot{m}_{NH3}\ (h_2 - h_1) \tag{13}$$

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 h_{is} is given by [13-15]:

$$\eta_{is} = 0.874 - 0.0135.\tau \quad (14)$$

With:

$$\tau = {P_{comp-out}}/{P_{comp-in}} \quad (15)$$

Coefficient of performance (COP) is given by the following expression [14]:

$$COP = {\dot{Q}_{EV}}/{(\dot{Q}_{GE1} + \dot{Q}_{GE2} + \dot{Q}_{comp})} \quad (16)$$

4. 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) [3,16].
The exergy balance for a control volume undergoing steady-state process is expressed as (Lee SF, Sherif SA, 2001) [4]:

$$Ex_{Di} = \sum(\dot{m}Ex)_{in} - \sum(\dot{m}Ex)_{out} \pm \sum(Ex_{\dot{Q}}) \pm \sum\dot{W} \quad (15)$$

Where $Ex_{\dot{Q}}$ is the thermal exergy and expressed as follow [33, 34]:

$$Ex_{\dot{Q}} = \dot{Q}(1 - {T_0}/{T}) \quad (16)$$

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

The first two terms on the right-hand side represent the exergy of streams entering and leaving the control volume. Both third and fourth terms are the exergy associated with heat transfer $\dot{Q}$from the source maintained at a constant temperature T and is equal to the work obtained by the Carnot engine operating between T and T_0, and is therefore equal to 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.

$$Ex_{DT} = \sum Ex_{Di} \quad (17)$$

We can also express the exergy loss in terms of exergetic efficiency; it is the rate between the inlet exergy and the outlet one [17,18]:

$$\eta_{ex} = \frac{outlet\ system\ exergy}{intletsystèmeexergy} \quad (18)$$

The exergy destruction in each component of the hybrid cycle is given by [22e24]:

$$Ex_{D.CD} = \dot{m}_{NH3}(h_{12} - Ts_{12}) - \dot{m}_{NH3}(h_{13} - Ts_{13}) + \dot{Q}_{CD}(1 - T_0/T_{CD}) \quad (19)$$

$$Ex_{D.EV} = \dot{m}_{NH3}(h_{13} - Ts_{13}) - \dot{m}_{NH3}(h_1 - Ts_1) + \dot{Q}_{EV}(1 - T_0/T_{ev}) \quad (20)$$

$$Ex_{D.AB1} = \dot{m}_{NH3}[(h_2 - Ts_2) + (f - 1)(h_6 - Ts_6) - f(h_3 - Ts_3)] - \dot{Q}_{AB1}(1 - 1 - \frac{T_0}{T_{cd}}) \quad (21)$$

$$Ex_{D.AB2} = \dot{m}_{NH3}[(h_7 - Ts_7) + (f - 1)(h_{11} - Ts_{11}) - f(h_8 - Ts_8)] - \dot{Q}_{AB2}(1 - 1 - \frac{T_0}{T_{cd}}) \quad (22)$$

$$Ex_{D.GE1} = \dot{m}_{NH3}[f(h_4 - Ts_4) - (f - 1)(h_7 - Ts_7) - (h_5 - Ts_5)] + \dot{Q}_{GE1}.(1 - \frac{T_0}{T_{GE}}) \quad (23)$$

$$Ex_{D.GE2} = \dot{m}_{NH3}[f(h_9 - Ts_9) - (f - 1)(h_{10} - Ts_{10}) - (h_{12} - Ts_{12})] + \dot{Q}_{GE2}.(1 - \frac{T_0}{T_{GE}}) \quad (24)$$

$$Ex_{D.comp} = \dot{m}_{NH3}(h_1 - Ts_1) - \dot{m}_{NH3}(h_2 - Ts_2) + \dot{W}_{COMP} \quad (25)$$

Exergetic efficiency of this machine is so given by equation (26) [14, 17,18]:

$$\eta_{exT} = \frac{\dot{Q}_{EV}(1 - T_0/T_{ev})}{\dot{W}_{COMP} + (\dot{Q}_{GE1} + \dot{Q}_{GE2})(1 - T_0/T_{GE})} \quad (26)$$

<u>Assumptions</u> :

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.

We can also highlight the irreversibility percentage of each component of the system from the total exergy loss by the relation below:

$$\dot{I}(\%) = {}^{Ex_{D.i}}/_{Ex_{DT}} \quad (27)$$

5. Results and discussion

In this study, numerical analysis of the system was performed using the Aspen plus software. Thermodynamic properties of the binary mixtures were determined using the NRTL model based on experimental results presented by Zhang et al [19]

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.

In figure 2 is shown the COP evolution versus the generator temperature. Different condensation temperatures are used for the R236fa/DMAC system, while the COP of the NH_3/H_2O system is presented only for a temperature of 40°C. The COP of the hybrid double stage system (HDS) is given in solid line while that of the simple absorption double stage system (DS) is given by the dotted line. The results are given for the same compression ratio of about 2.

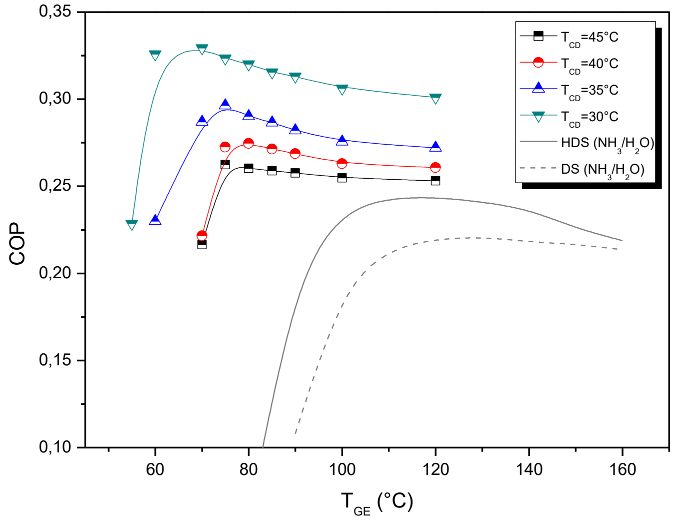

Figure 2: COP evolution versus T_{GE} for various T_{CD} for the two mixtures and both studied systems

The best energetic performances are obtained with R236fa/DMAC mixture used in the hybrid double stage system, it is about 0.33 for a condensation temperature of 30°C and 70°C as generator temperature.
COP of this hybrid double stage system is about 0.23 when working with NH3/H2O. In fact, the maximal COP for the classic mixture is achieved for a generator temperature of 120°C when that of the proposed couple is achieved for only 70°C to 80°C.

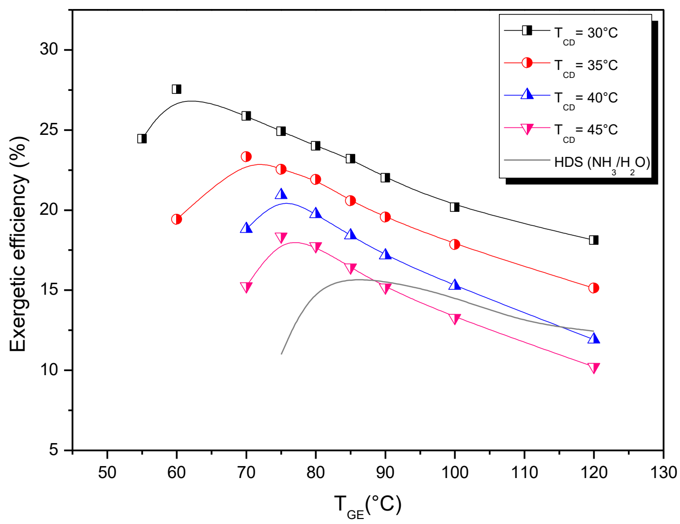

Figure 3: Exergetic efficiency evolution versus T_{GE} for various T_{CD} for the two mixtures and both studied systems

In figure 3 is presented the exergetic efficiency of the whole hybrid system working with the proposed mixture at the previous same working conditions.
It is observed from figure 3 that exergetic efficicency of the R236fa/DMAC system is higher than NH_3/H_2O one.
Results show that optimal exergetic efficiency value of the system is achieved for a condensation temperature of 30°C and at 60°C only as generator temperature. For the worse working conditions (TCD=45°C) the optimum value is obtained at 80°C. From the figure 3, it is clear that the exergetic efficiency of the system decreases for the high generator temperatures unlike the COP behavior.

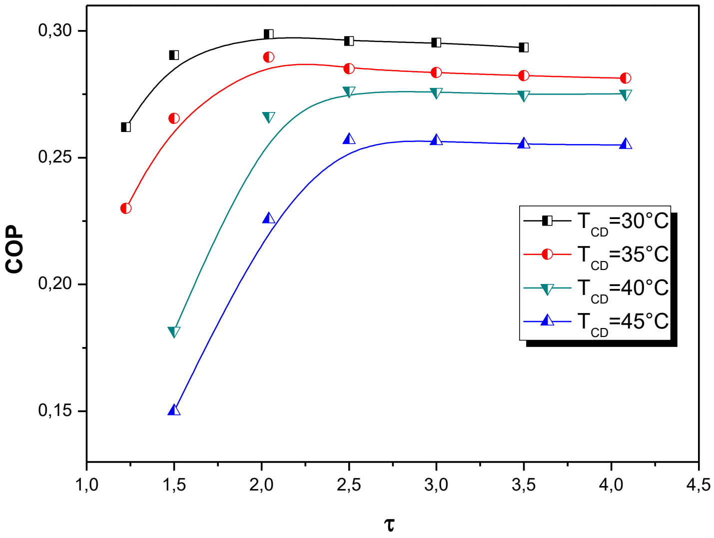

Figure 4: COP evolution versus the compression ratio for various T_{CD} for the two mixtures and both studied systems at T_{GE} = 80°C

In order to further evaluate the hybrid double stage cycle performances, the effect of the compression ration "τ" given by the equation (15) is investigated.

In figure 4, the COP evolution versus the compression ratio is presented for a generator temperature of 80°C. Optimal values of COP are obtained for τ varying from 2 to 2,5, and further increasing the compressor outlet pressure doesn't ameliorate the energetic performance of the system any more. The COP of the DHS can achieve 0,30 for a condensation temperature of 30°C.

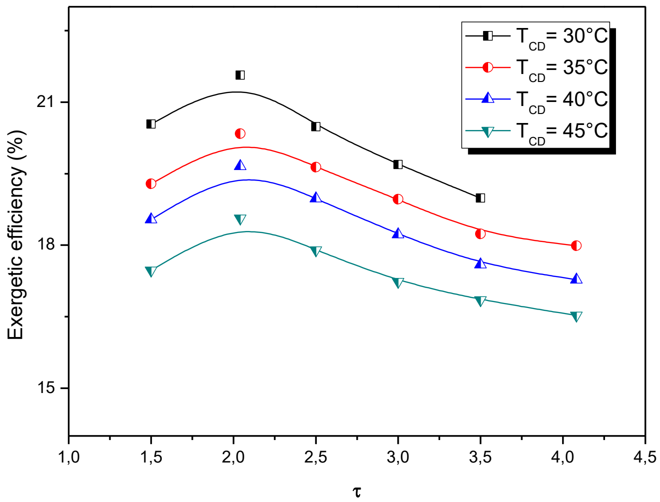

Figure 5: Exergetic efficiency evolution versus the compression ratio for various T_{CD} for the two mixtures and both studied systems at T_{GE} = 80°C

Figure 5 shows the variation of exergetic efficiency of the system beyond the compression ratio in the same previous operating conditions. It is clear from this figure that the exergetic efficiency decreases for the compression ratios exceeding 2. This result proves that running the system at low pressures preserves the energy quality.

In previous works, we evaluated the performances of several mixtures on different hybrid absorption systems such as R124/DMAC mixture [15] and the R245fa/DMAC and even the R236fa/DMAC on a one stage hybrid system [20]. In figure 6, we present a comparison study of the pressure gaps of all studied. There is also the results relative to the Ammonia/water system as a reference for the hybrid absorption/compression refrigeration system.

From this comparative study, we retain that the R236fa/DMAC mixture uses the lowest pressure gap in addition of the lowest threshold temperatures. This mixture is very suitable for low grade energy use.

For average pressure use, both of R124/DMAC and R245fa/DMAC are useful. R124/DMAC system has better energetic and exergetic performances than the R245fa/DMAC system.
Ammonia/water system has the higher pressure gap (between 4kPa to 18kPa), it is requires necessarily more specific materials and more electric consumption in the compression process.

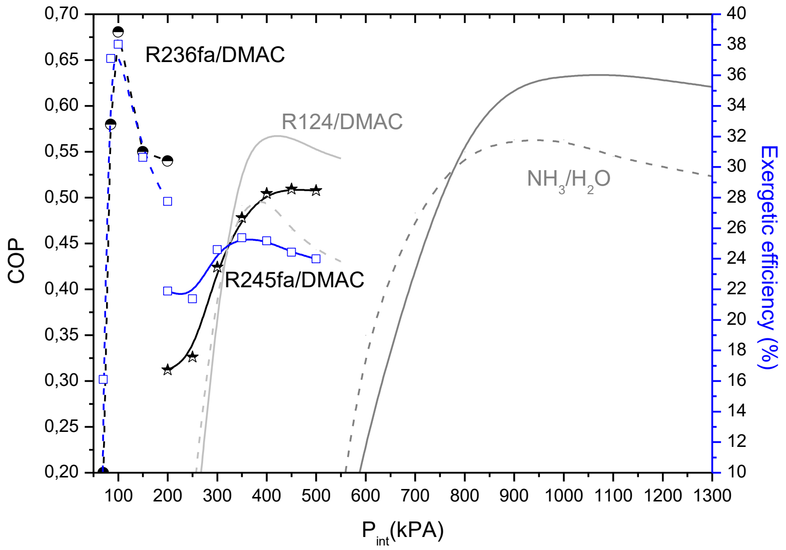

Figure 9 Comparison of the intermediate pressure gaps for
the two proposed mixtures with old studied one (T_{GE}=80°C and T_{CD}=30°C)

Conclusion

In this work, simulation of a hybrid compression/absorption double stage heat pump has been elaborated with a novel proposed couple by Aspen Tech software: R236fa/DMAC (1,1,1,3,3,3-Hexafluoropropane/ N,N'dimethylacetamide). Exergetic analysis was developed to compare the system performances using the two mixtures: The classic one NH_3/H_2O and the proposed one. Conclusions of this study are drawn below:

- The best energetic performances of the heat pump are obtained with R236fa/DMAC mixture. COP reaches 0.33 for a condensation temperature of 30°C and for low generator temperature (70°C) and even if T_{CD} is about 45°C, the COP is about 0.27 for only 80°C.
- Energetic performances of the system are ameliorated with the new couple while providing lower generator temperatures for the system (from 50°C to 80°C) which makes this couple a good option for solar source use.
- Working with the proposed mixture allows working in lower pressures (from 300kPA to 1800kPA for Ammonia/water and from 67Pa to 500kPa for R236fa/DMAC). This can make the engine safer and easily realizable.
- The new proposed couple is reducing irreversibility of the system, exergetic efficiency is much better than that off all previous studied mixtures in the hybrid absorption refrigeration field.
- The proposed new working mixture, not only can solve the toxicity problem of the ammonia use, but also adapt the system to low temperatures sources thanks to its low generator temperatures.

Nomenclature

COP	Coefficient of Performance
P	Pressure, *bar, Pa*
T	Temperature, *K, °C*
x	mass fraction
Ex_D	Exergy destruction, *W*
$\dot{m}$	Mass flow rate, $kg.s^{-1}$
$\dot{W}$	Work transfer rate, *W*
$\dot{Q}$	Heat transfer rate, *W*
Ex	Specific exergy of a stream, $kJ.kg^{-1}$
h	Specific enthalpy of a stream, $kJ.kg^{-1}$
s	Specific entropy of a stream, $kJ.kg^{-1}.K^{-1}$
f	Specific solution circulation factor
η_{ex}	Exergetic efficiency
η_{is}	Isentropic efficiency

Subscribes

i	= Component or stage i	EV	= Evaporator
T	= Total	COMP	= Compressor
0	= Reference	GE	= Generator
2	= intermediate	ECH	= Solution exchanger
v	= Vapor	AB	= Absorber.
		SR	= Rich solution
		SP	= Weak solution

Références (Times New Roman,10 pts, Gras)

[1] Auteurs (Initiales du prénom Nom), Titre publication, *Revue (Congrès…),* Volume, Page, Date.
Exemple

[1] Bouaziz N, Ben Iffa R, Kairouani L. Performance of a water ammonia absorption system operating at three pressure levels. J Mech Eng Res;3(4):120-7, 2011.

[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;25:450-9, 2002

[4] Lee SF, Sherif SA. Second law analysis of various double effect lithium bromide/water absorption chillers. . ASHRAE Transactions; AT-01-9-5:664-673, 2001

[5] Arora A, Kaushik SC. Theoretical analysis of LiBr/H_2O absorption refrigeration systems. Int J Energy Res ;33:1321-40, 2009

[6] Dincer I, Cengel YA. Energy, entropy and exergy concepts and their roles in thermal engineering. Entropy ;3:116-49, 2001

[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;14(Nos. 1-4):55-9, 1998

[12] Anand DK, Kumar B. Absorption machine irreversibility using new entropy calculations. Sol Energy ;39(3):243-56, 1987

[13] L. Kairouani and E. Nehdi, (2005), Thermodynamic Analysis of an Absorption/Compression Refrigeration System Using Geothermal Energy. American Journal of Applied Sciences 2 (5), 914-919.

[14] Nahla Bouaziz, D. Lounissi, Energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling. International journal of hydrogen energy, 40 (2015) 13849-13856.

[15] Lounissi, D., Bouaziz, N. and Kairouani, L. (2015) 'An exergetic study of a new hybrid heat pump based on R124/DMAC mixture for solar cooling', Int. J. Exergy, Vol. 17, No. 4, pp.513–531.

[16] 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.

[17] Lounissi D, Bouaziz N, Exergetic analysis of an absorption/compression refrigeration unit based on R124/DMAC mixture for solar cooling, International Journal of Hydrogen Energy (2016), http://dx.doi.org/10.1016/j.ijhydene.2016.11.082

[18] Bouaziz N, Lounissi D, Experimental analysis, hybridization and exergetic investigation of an absorption refrigeration unit, International Journal of Hydrogen Energy (2016), http://dx.doi.org/10.1016/j.ijhydene.2016.11.056

[19] Min Zhang, Danxing Zheng, Wenzhi Yang, and Xiaoxiao Li. Measurement and Correlation of Isothermal Vapor−Liquid Equilibrium of 1,1,1,3,3-Pentafluoropropane + N,N-Dimethylacetamide and 1,1,1,3,3,3-Hexafluoropropane + N,N-Dimethylacetamide/Diethylene Glycol Dimethyl Ether Systems. J. Chem. Eng. Data, 2014, 59, 3912−3919.

[20] Rabeb Toujani, Dorra Lounissi and Nahla Bouaziz, Exergetic comparison of two novel mixtures for an upgrated absorption/compression heat pump : R245fa/DMAC and R236fa/DMAC, 4ème Conférence Internationale des Energies Renouvelables (CIER-2016) Proceedings of Engineering and Technology – PET Pol.16, pp.120 – 125.