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EXERGETIC COMPARISON OF TWO NOVEL MIXTURES FOR AN UPGRADED ABSORPTION/COMPRESSION HEAT PUMP: R245fa/DMAC AND R236fa/DMAC

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***Abstract*— In this work we present an energetic and exergetic comparison study of two novel mixtures for an upgraded absorption/compression heat pump : R245fa/DMAC (1,1,1,3,3-Pentafluoropropane/N,N'dimethylacetamide) and R236fa/DMAC (1,1,1,3,3,3-Hexafluoropropane/ N,N'dimethylacetamide). These working fluids are selected due to their low GWP. Results relative to the new couples are compared with those relative to the classical water/ammonia working fluids. Investigated parameters are the COP, the exergy loss, the exergetic efficiency and the irreversibility of each cycles' component. Results show that the best performance coefficient of the system is obtained with R236fa/DMAC. Its optimum value is about 0.62. COP of R245fa/DMAC system has similar rates as the classical water/ammonia one. Furthermore, the system using the new couples uses lower threshold temperatures, between 50°C and 80°C for optimum COP, which allows the use of low temperature energy sources. Results of the exergetic analysis indicate that exergy loss of the R236fa/DMAC system is the lowest. For 3kW as refrigeration power optimum exergy loss is about 2.5 kW. Best exergetic efficiency is also noted for the R236fa/DMAC system. It is noted from this study that the major gain brought by these new pairs is the diminution of the threshold temperatures which enables the solar energy use.**

***Keywords*— Compression/Absorption, exergy analysis, organic absorbent, hybrid heat pump.**

I. 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 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

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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/H2O or NH3/H2O (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.

The aim of this work is to present an energetic and exergetic comparison study of two novel mixtures for an upgraded absorption/compression heat pump : R245fa/DMAC (1,1,1,3,3-Pentafluoropropane/N,N'dimethylacetamide) and R236fa/DMAC (1,1,1,3,3,3-Hexafluoropropane/ N,N'dimethylacetamide. All energetic and exergetic results are compared to NH3/H2O hybrid machine. The effects of the condensation temperature and the generator temperature on system performances are examined. Exergy loss of the heat pump was evaluated for several working conditions. It is hoped that these results could serve as a source of reference for designing and selecting new absorption refrigeration systems, developing new working fluid pairs and optimizing suitable operating conditions.

II. Heat pump presentation

The heat pump, subject of this study, is a combination between the conventional absorption one and the compression one. A compressor is injected into the cycle, upstream the absorption part, in order to ameliorate the absorption process (figure1).

The system operates at three pressure levels; the refrigerant vapor leaving the evaporator (1) is at the lower pressure (PEV). It is compressed by an isentropic transformation (2) to an intermediate pressure (P2) and finally, injected into the absorber. The rich solution going from the absorber (3) is heated by the poor one coming from the bottom of the generator (5) by a solution heat exchanger. The condenser and the generator operate at the third level of pressure (PCD) which is the condensation pressure.

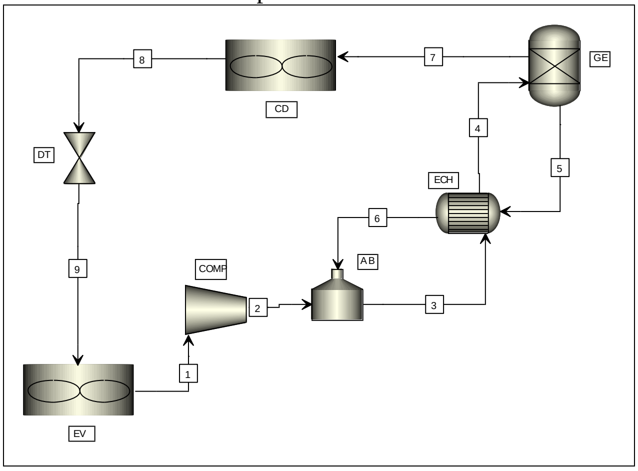


Fig.1 One stage hybrid cycle

III. Analysis and modelling

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 [13].

A. *Energy and mass balance*

The mass balances in this cycle governing the three present substances: weak solution, rich solution and refrigerant gas are given by:

$$\dot{m}_{SR} = f.\dot{m}_{NH3} \quad (1)$$

$$\dot{m}_{Sp} = (f-1).\dot{m}_{NH3} \quad (2)$$

Where f is the specific solution circulation factor for the cycle:

$$f = \frac{x_V - x_{SP}}{x_{SR} - x_{SP}} \quad (3)$$

$\dot{m}_{SR}$, $\dot{m}_{Sp}$ and $\dot{m}_{NH3}$ are consecutively the rich solution, the weak solution and the gas mass flow rates.

Installation contains five principle components: Condenser, evaporator, generator, absorber and compressor.

Energy balance for each installation component is presented by the following equation:

$$\dot{Q}_i = \sum(\dot{m}.h)_{output} - \sum(\dot{m}.h)_{input} \quad (4)$$

ISSN 2356-5608

The compressor duty is determined by the equations below:

$$\dot{W}_{real} = \dot{W}_{is}/\eta_{is} \qquad (5)$$

$$\dot{W}_{real} = \dot{m}_{NH3}\,(h_3 - h_2) \qquad (6)$$

So, we can conclude from "Eq (4)" and "Eq (5)", the value of the steam enthalpy at the compressor outlet.

Were the isentropic efficiency η_{is} is given by [14, 15]:

$$\eta_{is} = 0.874 - 0.0135.\tau \qquad (7)$$

$$\tau = P_{comp-out}/P_{comp-in} \qquad (8)$$

Coefficient of performance is deduced from the following expression, [16-19]:

$$COP = \dot{Q}_{EV}/(\dot{Q}_{GE} + \dot{W}_{real}) \qquad (9)$$

B. Exergy model

The exergy balance is based on exergy destruction method, were it is calculated for each component as follow [20]:

$$ExDi = \sum(\dot{m}Ex)_{in} - \sum(\dot{m}Ex)_{out} \pm \sum(Ex_{\dot{Q}}) \pm \sum\dot{W} \qquad (10)$$

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.

$Ex_{\dot{Q}}$ is the thermal exergy and expressed as follow [21, 22]:

$$Ex_{\dot{Q}} = \dot{Q}(1 - T_0/T) \qquad (11)$$

We can also express the exergy loss in terms of exergetic efficiency; it is the rate between the inlet exergy and the outlet one [23, 24]:

$$\eta_{ex} = \frac{outlet\ system\ exergy}{intletsystèmeexergy} \qquad (12)$$

Third level heading: 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 P0 and T0 are 1atm and 25°C, respectively.

IV. Results and Discussions

In the present study simulation of the absorption/compression cycle was done by Aspen Plus flow sheet simulator like shown before in Figure1.

A comparative study between three working mixtures is carried out; simulation was done for a refrigeration capacity of about 3kW for both systems.

In figure 2 is shown the COP evolution versus the generator temperature. Different condensation temperatures are used for the R245fa/DMAC and R236fa/DMAC systems, while the COP of the NH3/H2O system is presented only for a temperature of 30°C. The results are given for the same compression ratio of about 3.

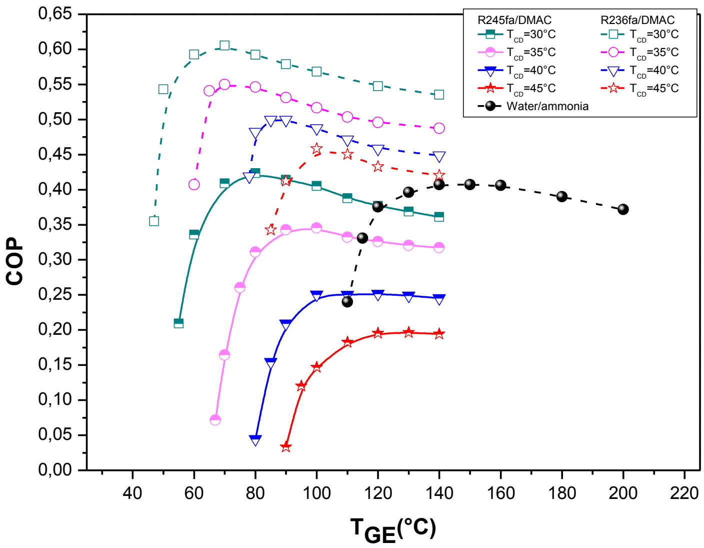


Fig.2 COP evolution versus T_{GE} for various T_{CD} for the three mixtures

The best energetic performances are obtained with R236fa/DMAC mixture, it is about 0.62 for a condensation temperature of 30°C and 70°C as generator temperature.

COP of this hybrid one stage system is about 42% when working with NH3/H2O and it can achieve the same value with the new proposed mixture R245fa/DMAC for a lower generator temperature. In fact, the maximal COP for the classic couple is achieved for a generator temperature of 140°C when that of the proposed couple is achieved for only 80°C.

Fig. 3 presents the exergy loss of the system for the two novel working mixtures and for the same conditions as fig. 2. The first remarked result, is that irreversibility of the system working with the R236fa/DMAC is lower than that caused by the classic one of about 12kW. Irreversibility is reduced when the condensation temperature decreases and when increasing the generator temperature. Its optimum values are obtained between 60°C and 100°C as generator temperature for the

ISSN 2356-5608

R236fa system and between 70°C to 110°C for the R245fa system. It is noted that less exergetic losses are obtained with the first mixture.

In fig. 4 is presented the exergetic efficiency of the whole hybrid system working with the proposed mixtures at the previous same working conditions.

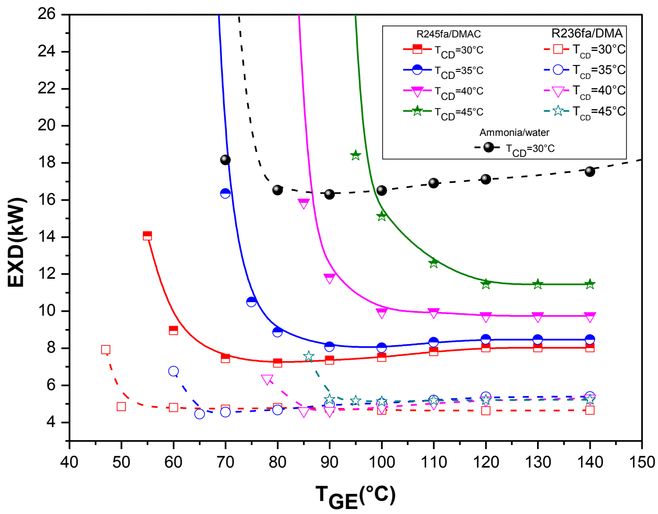


Fig. 3 Irreversibility evolution versus T_{GE} for various T_{CD} for the three mixtures

It is observed from fig. 4 that exergetic efficicency of the R236fa/DMAC system is higher than R245fa/DMAC 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 (T_{CD}=45°C) the optimum value is obtained at 90°C. From the fig. 4, it is clear that the exergetic efficiency of the system decreases for the high generator temperatures unlike the COP behavior.

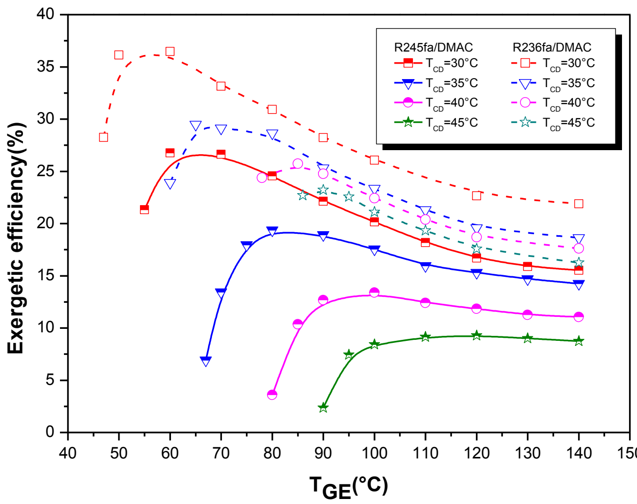
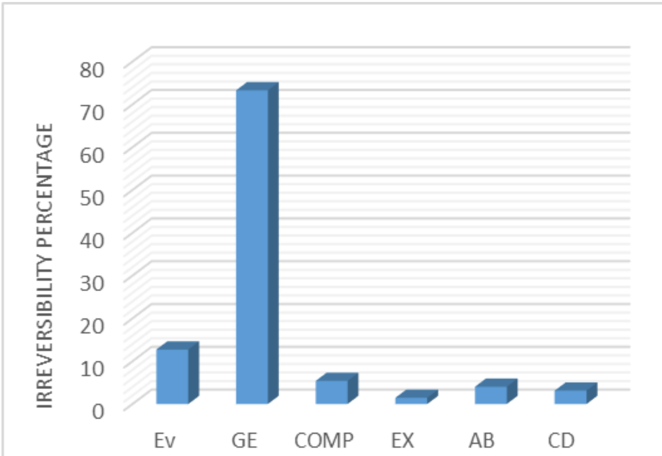


Fig. 4 Exergetic efficiency evolution versus T_{GE} for various T_{CD} for the two novel mixtures

Fig. 5 presents the part of each component in the total irreversibility of the system. Results are generated for a generator temperature of 90°C and a condensation temperature of 35°C.

From fig. 5, the great part of the system irreversibility is caused by the generator, about 72%. This result is predicted because of the high working pressures and temperatures in this component.

Some efforts have to be provided to decrease the irreversibility of the generator.



V. Conclusions

In this work, simulation of a hybrid compression/absorption one stage heat pump has been elaborated with two novel proposed couples by Aspen Tech software: water/ammonia R236fa/DMAC (1,1,1,3,3,3-Hexafluoropropane/ N,N'dimethylacetamide) and DMAC/R245fa (1,1,1,3,3-Pentafluoropropane /N, N' dimethylacetamide). Exergetic analysis was developed to compare the system performances using these two mixtures. Conclusions of this study are drawn below:

- The best energetic performances of the heat pump are obtained with R236fa/DMAC mixture. COP reaches 0.62 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 0.45 for only 100°C. This enables the solar energy use.
- Energetic performances of the system are conserved with the new couple R245fa/DMAC while providing lower generator temperatures for the system (from 50°C to 75°C), optimum functioning states are obtained for generator temperatures varying from 80°C to 110°C with given COP of 52%, which makes this couple a good option for solar source use.

ISSN 2356-5608

- Working with the proposed mixtures allows working in lower pressures (from 300kPA to 1800kPA for Ammonia/water, from 100kPA to 600kPA for R245fa/DMAC) and from 67Pa to 500kPa for R236fa/DMAC.This can make the engine safer and easily realizable.
- The new proposed couples are reducing irreversibility of the system, exery loss is reduced from 16kW with ammonia/water system to 4kW with the R236fa/DMAC one for a refrigeration power of 3kW. R245fa/DMAC still in the middle.
- The proposed new working mixtures, 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.
$\dot{m}$	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
T	= Total
0	= Reference
2	= intermediate
v	= Vapor
EV	= Evaporator
COMP	= Compressor
GE	= Generator
ECH	= Solution exchanger
AB	= Absorber.
SR	= Rich solution
SP	= Weak solution

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