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Conception of an Absorption Refrigerating System Operating at Low Enthalpy Sources

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1. Introduction

Since the work of Duhem in 1899, relating to binary mixtures, that the absorption chillers have been growing significantly.
In 1930, Borzig had developed a machine using the couple water-ammonia. This system is interesting in that it works by supplying heat energy regardless of its origin (waste heat, geothermal water, solar energy...).

2. Operating principle

The operating principle of such a machine is briefly describe below (Figure 1). The absorption system differs from the vapor compression machine by providing a third heat source which is the generator (Qg). The absorption machine uses a binary mixture; one fluid is more volatile than the other and constitutes the refrigerant. Couples most commonly used are: Water-Ammonia (H2O/NH3) Ammonia is the refrigerant. Lithium Bromide-Water (LiBr/H2O), water is the refrigerant. The elements of an absorption machine are shown in Figure 1. They are: Boiler or generator: the refrigerant rich solution is heated to the temperature Tg, which is higher than the temperature of vaporization of refrigerant to the pressure considered. Condenser: similar to that of a vapor compression machine.
Evaporator: similar to that of a vapor compression machine.

- Absorber: steam from the evaporator is absorbed by the existing solution which will be enriched in refrigerant. The absorber is also connected to the generator. The weak solution coming from the boiler enters to the absorber.
- Inter-exchange solution: all current machines include a heat exchanger (sometimes called internal transmitter) between the rich solution leaving the absorber to Tab and the weak solution leaving the boiler at Tg. This exchanger is used to preheat the rich solution before entering at the generator.

A pump is used to lead the rich solution to the generator, and an expander is used to return the weak solution to the absorber. In general, the coefficient of performance (COP) of such a machine is around 0.7. To improve the COP or adapt the machine to any source of energy, some purpose can be considered such a multiple effects machines, combined machines (absorption-compression, integration of ejectors...) [1-7]. The COP is defined as:

$$\text{COP} = \frac{\dot{Q}_F}{\dot{Q}_g + \dot{W}_P} \tag{1}$$

$\dot{W}_P$ is low so we can write:

$$\text{COP} = \frac{\dot{Q}_F}{\dot{Q}_g} \tag{2}$$

Where $\dot{Q}_F$ is the amount of cold produced and $\dot{Q}_g$, the heat energy supplied to the generator.
The mass balance at the generator provides for 1kg of refrigerant vapor, (*f*) kg of solution as:

$$f = \frac{x_v - x_p}{x_r - x_p} \tag{3}$$

x_v is the mass fraction of steam (x_v =1 for LiBr/H_2O and is close to 1 for the couple NH_3/H_2O). Où x_p and x_r are respectively the titles of the weak and rich solutions, determined from the diagrams of Merkel and Oldham. *(f)* is called entrainment ratio, he must have reasonable values in order to reduce the energy consumption of the pump. In what follows, we present the performance of absorption chillers using couples NH_3/H_2O or LiBr/H_2O.

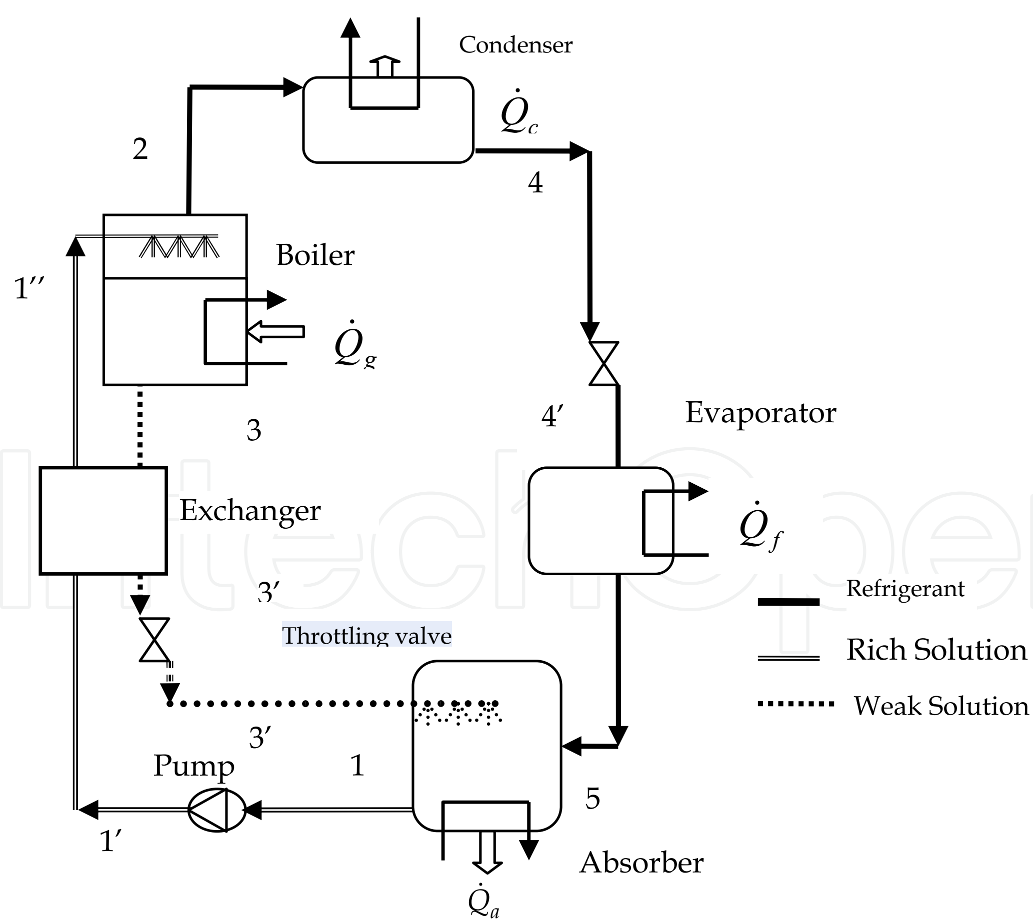

Fig. 1. Operating principle of an absorption machine

2.1 Oldham diagram

The refrigeration cycle is shown in the Oldham diagram (Log P, $\frac{1}{T}$) on which we can trace the iso-titles of the solution. By choosing the pair of pressure evaporation and condensation (P_e, P_c), it follows the pair of corresponding temperatures (T_e, T_c). From the saturation line (x=100%), we draw a vertical line to determine the rich solution (x_r). The intersection of the line of the rich solution and isobaric P_c indicates the threshold temperature (T_s). The threshold temperature (T_s) is the minimum temperature of the generator, below which the installation does not work. The generator temperature determines the line of the weak solution and hence its title (xp) (Figure 2).

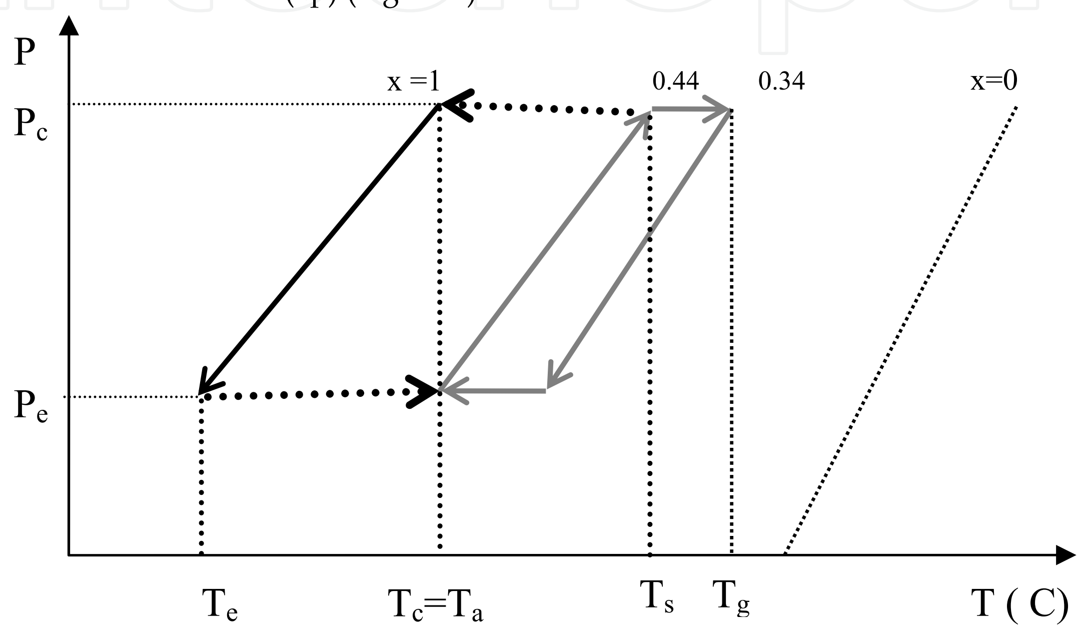

Fig. 2. Oldham Diagram

NH_3/H_2O installations must be equipped with a rectification column to remove water entrained with the refrigerant to prevent it from solidifying in the pipes of the evaporator.
The cooling capacity is:

$$\dot{Q}_e = \dot{m}_{ff}\Delta h_{év} \tag{4}$$

$$\dot{Q}_e = COP\dot{Q}_g \tag{5}$$

$\dot{Q}_g$ is the heating power:

$$\dot{Q}_g = \dot{m}_{ff}[-f\,h_{abs} + (f-1)h_g + h_v] \tag{6}$$

(*f*) is the driving factor, it is the mass of rich solution is likely to emit one kg of refrigerant vapor.
h_v is the heat of vaporization of refrigerant in the solution.
h_{abs} is the enthalpy of the rich solution leaving the absorber.
h_g is the enthalpy of the weak solution leaving the generator.

These enthalpies are taken from the diagram of Merkel or can be obtained from empirical correlations [8-9]

3. Combined and multi-stage installations

We have presented a single absorption plant; it is conceivable to study the combined or hybrid systems.
For the combined system, the absorption cycle, serve to ensure the condensation of the refrigerant for the vapor compression cycle. The latter can operate between temperatures of condensation and evaporation desired.
For the hybrid system, a compressor acts as a liaison between two stages of absorption.

3.1 Combined installations

The system presented as an example, Figure 3, uses for the installation of R134a vapor compression and the couple water-ammonia absorption for installation.
Condensing temperature is 30 C and evaporating temperature of R134a is -10 C. The COP of the plant absorption is only 0.64 [1].
This system can be profitable if you have a free source of energy or recovery such as solar, thermal discharges of gas power plants or geothermal energy.

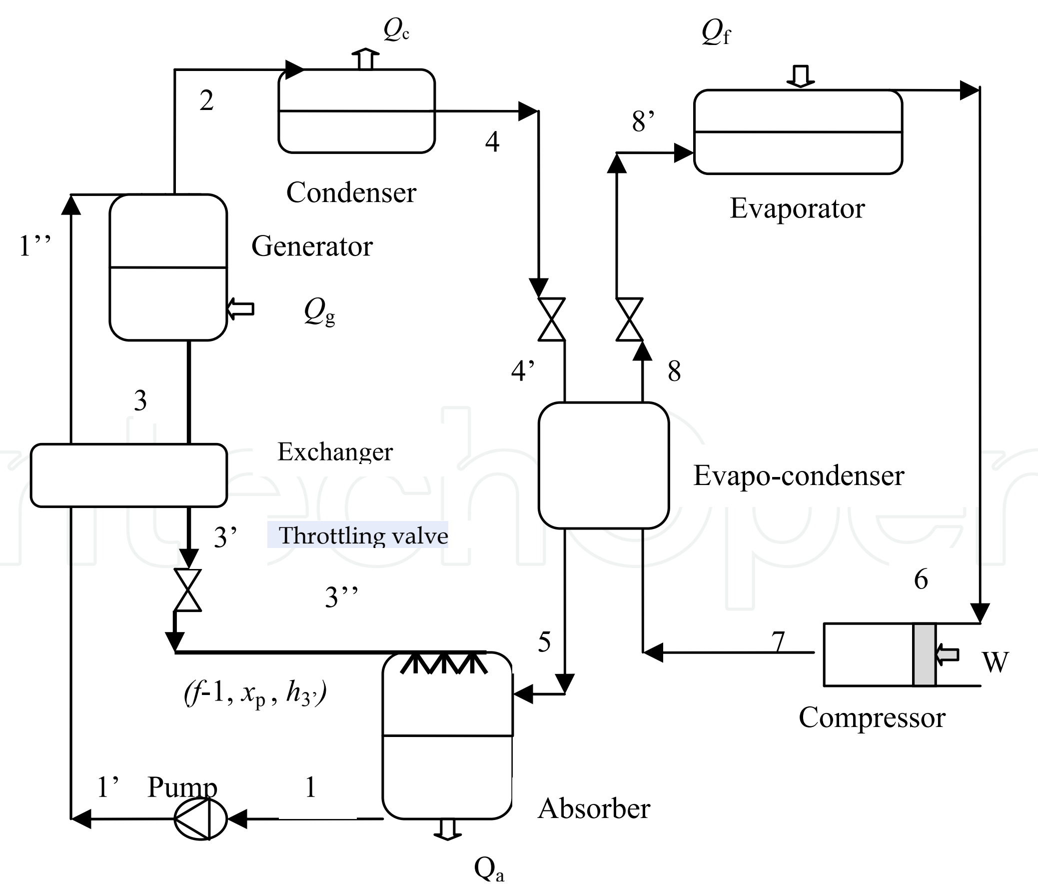

Fig. 3. Combined Installation

This absorption/compression refrigeration system is proposed to improve the overall cycle efficiency. The COP excluding the pump work and the generator energy required is as high as 5.4–6.2, which is higher than that of the single vapor compression cycle and absorption cycle, under the same operating conditions (evaporation temperature at 263 K and condensation temperature at 308 K).
This system presents an opportunity to reduce the continuously increasing electrical energy consumption.
Further investigations are needed to optimize the combined system design and operating parameters and to assess the efficiency and the feasibility of the system. A pilot installation can be built near geothermal, solar or waste energy sources [1].

3.2 Double stage system

In the absorption system at double stage, shown in Figure 4, the displacement of the refrigerant from low pressure to high pressure by means of two thermo-compressors 1 and 2 combined in series. To analyze the cycle of transformations, we consider the following assumptions:

- Temperatures of output rich solutions from absorbers Ab_1 and Ab_2 are equal and identical to the condensation temperature T_c.
- Temperatures of output weak solutions from generators Ge_1 and Ge_2 are equal.

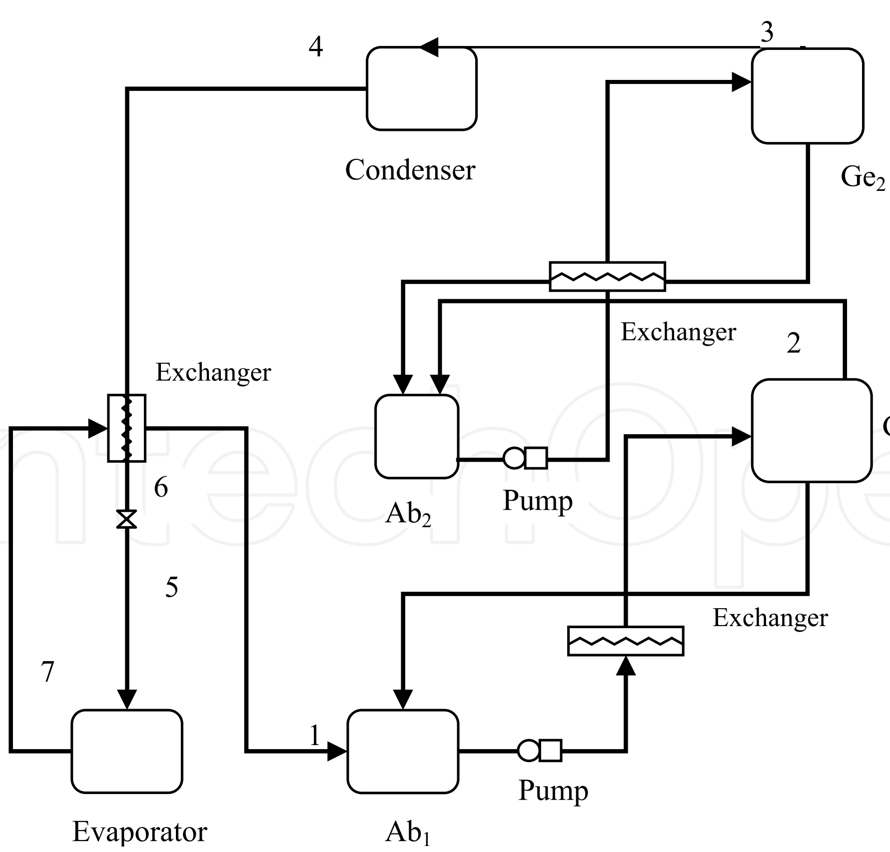

Fig. 4. Absorption machine with double-stage

There is no wall friction in the circuit.
In this installation, the first thermo-compressor transports the refrigerant from low pressure P_F to an intermediate pressure Pi corresponding to a saturation temperature of the refrigerant, T_i. Mass titles are respectively x_{r1} and x_{p1} for rich and poor solutions.
The second thermo-compressor transports the refrigerant of intermediate pressure Pi to the condenser pressure Pc. Mass titles are respectively xr2 and xp2 for rich and poor solutions.
The addition of one or more intermediate stages has a direct influence on lowering the generator temperature. But if the number of stages increases, the coefficient of performance decreases. Multi-stage systems have been studied by several authors; the results show that the COP is about 0.37 but a generator temperature is less than that of a single stage. The generator temperature can reach 65 C when T_c is 40 C and the COP of the plant is 0.26 which is relatively higher than that of a single stage that does not exceed 0.25 for an evaporation temperature of -10 C. These operating conditions can be profitable for valorization of energy sources at low enthalpy.

3.3 Other systems with multi-stages

We develop other configurations double-stage, we detail the calculation of energy and mass balance for some of them.
Several authors have considered different absorption machine configurations. Some systems are composed with simple stage machine [10-15] and others are formed by a succession of stages with various component associations and sometimes inserting other new components [16-20]. In the following section, we present three different configurations of the multi-stage refrigeration system and we develop a novel absorption hybrid configuration. We will explain and quantify it's adaptability to low-enthalpy sources. All configurations object of this work are followed by the representation of the corresponding cycle on the Oldham diagram.

3.3.1 System AGEcAG

The cascade system is composed of two elementary cycles, each one is considered as a single stage with the main difference, that the second stage is operating at a higher evaporative and condenser temperatures (see Figure 5). Figure 6 shows the Oldham diagram of the cycle.
In the following, we develop the energy and mass balance for the system AGEcAG. The entrainment factor f_i is the necessary rich solution flow able to move 1 kg of NH_3 from generator.

$$f_i = \frac{x_{vi} - x_{pi}}{x_{ri} - x_{pi}} \tag{7}$$

The proportion of the hot and cold flows at the mixer (evapo-condenser) is noted y and defined as follows:

$$y = \frac{h_{v1} - h_{sor\_mé\_liq}}{\left(h_{sor\_mé\_va} - h_{sor\_cd}\right)} \tag{8}$$

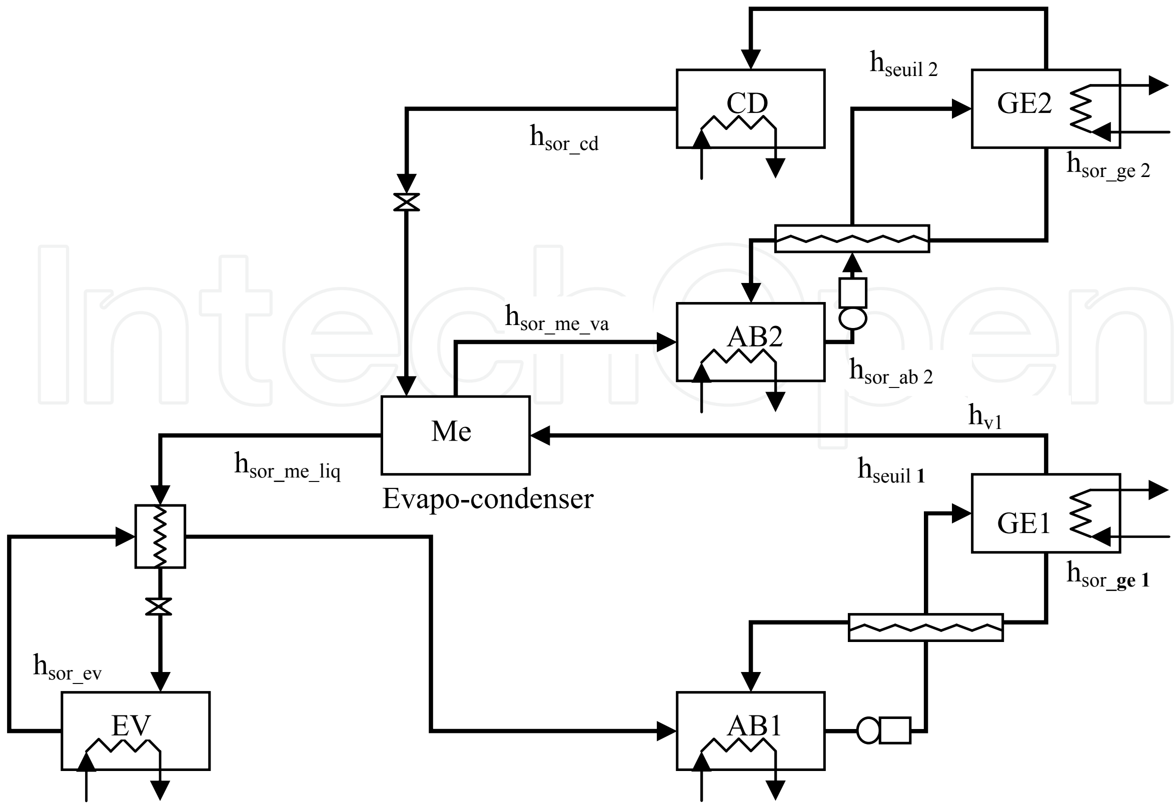

Fig. 5. System AGEcAG

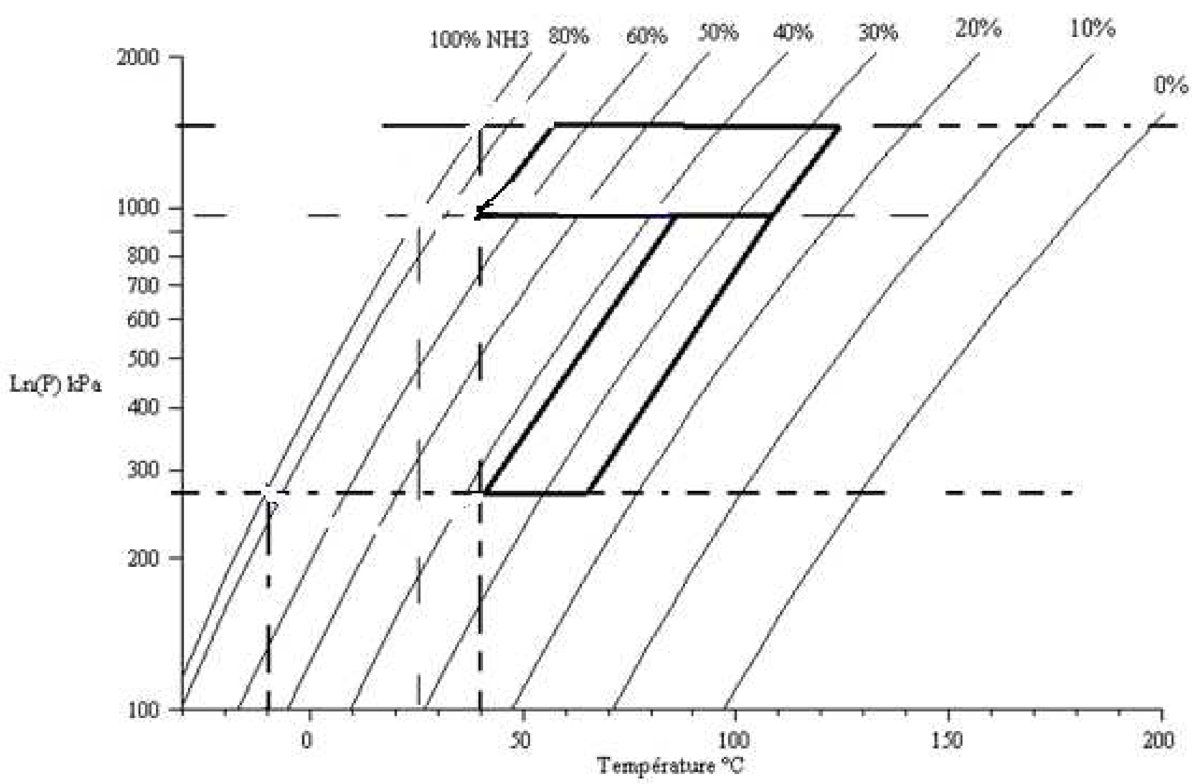

Fig. 6. Oldham Diagram of system AGEcAG

Energy balance for each installation component is presented. By neglecting the rectifier, we get for, Condenser, Evaporator, Generator and Mixer respectively:

$$\dot{Q}_{CD} = \dot{m}_{NH3} y \cdot (h_{v2} - h_{sor\_cd}) \tag{9}$$

$$\dot{Q}_{EV} = \dot{m}_{NH3}\,(h_{sor\_ev} - h_{sor\_mé\_liq}) \tag{10}$$

$$\dot{Q}_{GE1} = \dot{m}_{NH3}(h_{v1} + (f_1 - 1)\cdot h_{sor\_ge1} - f_1 \cdot h_{seuil1}) \tag{11}$$

$$\dot{Q}_{GE2} = \dot{m}_{NH3} y \cdot \left(h_{v2} + (f_2 - 1)\cdot h_{sor\_ge2} - f_2 \cdot h_{seuil2}\right) \tag{12}$$

$$\dot{Q}_{AB1} = \dot{m}_{NH3}(h_{sor\_ev} + (f_1 - 1)\cdot h_{ent\_ab1} - f_1 \cdot h_{sor\_ab1}) \tag{13}$$

$$\dot{Q}_{AB2} = \dot{m}_{NH3} y \cdot \left(h_{sor\_mé\_liq} + (f_2 - 1)\cdot h_{ent\_ab2} - f_2 \cdot h_{sor\_ab2}\right) \tag{14}$$

$$\dot{Q}_{Mé} = \dot{m}_{NH3}(h_{v1} - h_{sor\_mé\_liq}) \tag{15}$$

After developing the energy and mass balance for the cascade system of AGEcAG,
The COP's system is defined as follows:

$$COP = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE1} + \dot{Q}_{GE2}} \tag{16}$$

Where $\dot{Q}_{GE1}, \dot{Q}_{GE2}$ and $\dot{Q}_{EV}$ are the generators and evaporator power, respectively.
We note that for the considered absorption refrigerating system, the second stage is used to lower the operating temperature of the first stage and not to increase the COP. It is evident that the amount of the system required energy is higher than that required for a single stage, because of the two generator components.

3.3.2 System AGAG

This machine is composed by two absorbers, a condenser working at the same temperature TAB, two generators operating at the same temperatures (T_{GE1}=T_{GE2}) and an evaporator. Besides the absorber $AB_1$and the evaporator EV, the absorber AB_2 and the generator GE1, the condenser CD and the generator GE_2 operate respectively at the pressures P_{EV}, P_{moy} and P_{CD}. The connection between the two stages is provided between the generator G_{E1} and the absorber AB_2 (see Figure 7). Figure 8 shows the Oldham diagram of the cycle.
We develop bellow, the energy balance and mass for the cascade system AGAG
To calculate the entrainment factors, we use equation (6) and after determining x_{ri}, x_{pi} for (i = 1 or 2) that are the titles of the rich solutions and the poor solution for the first stage (i = 1) and the second stage (i = 2).

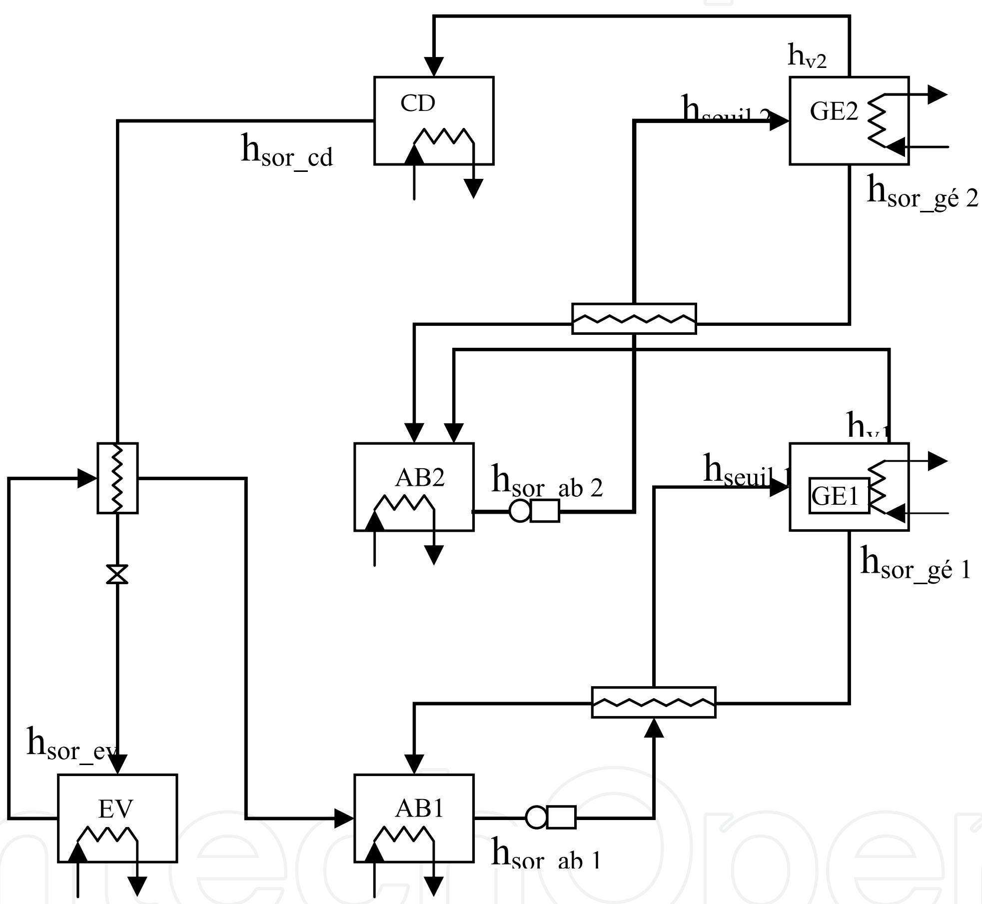

Fig. 7. System AGAG (connection generator - absorber)

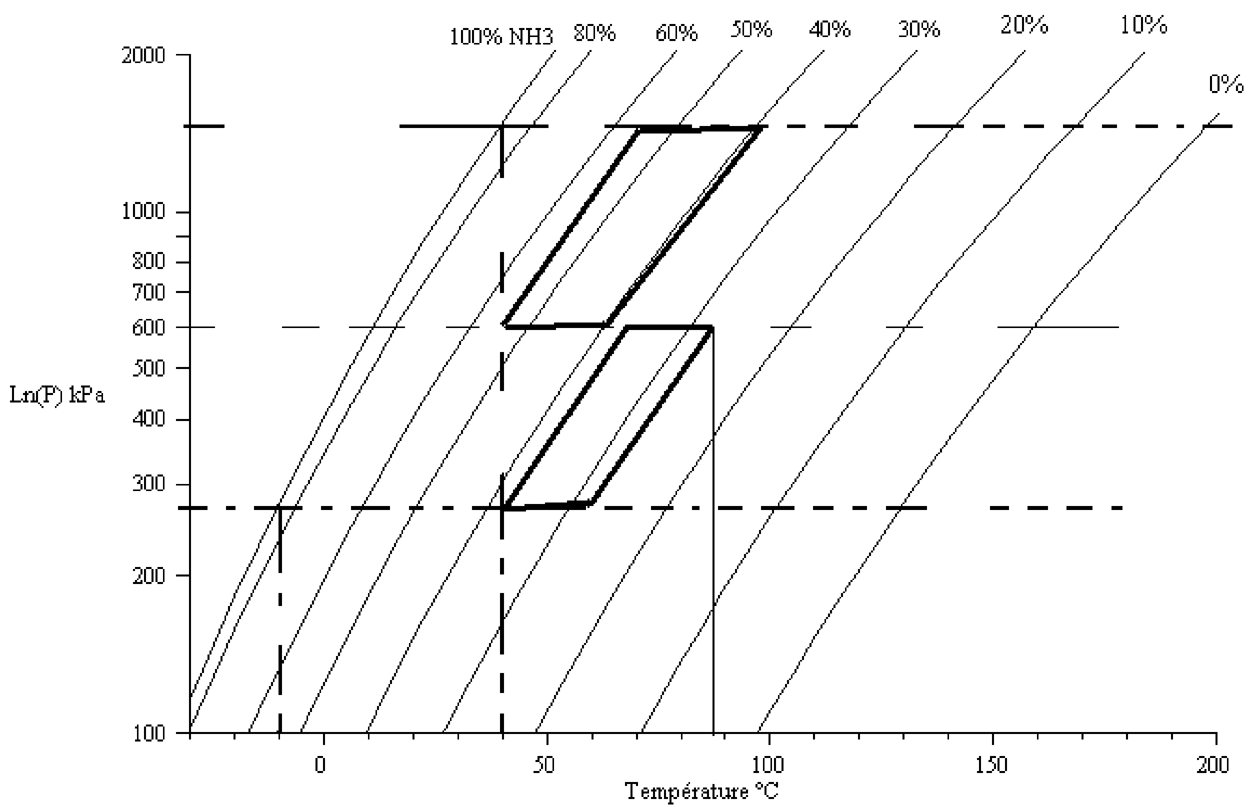

Fig. 8. Oldham diagram of system AGAG

$\dot{m}_{NH31}$ and $\dot{m}_{NH32}$ are respectively the mass flow of the refrigerant at the 1st and the 2nd stage. The mass balance for the two stages, gives:

$$\dot{m}_{NH3} = \dot{m}_{NH3i} \tag{17}$$

Two equations can be deduced from (17):

$$\dot{m}_{SRi} = f_i \cdot \dot{m}_{NH3i} \tag{18}$$

$$\dot{m}_{Spi} = (f_i - 1) \cdot \dot{m}_{NH3i} \tag{19}$$

For i=1 or 2

In order to establish the energy balance, we consider the same assumptions and we neglect the work of the pumps and the thermal power of the two rectification columns.

Heat released from the condenser is:

$$\dot{Q}_{CD} = \dot{m}_{NH3}(h_{v2} - h_{sor\_cd}) \tag{20}$$

Heat added to the evaporator is:

$$\dot{Q}_{EV} = \dot{m}_{NH3}(h_{sor\_ev} - h_{sor\_cd}) \tag{21}$$

Heat added to the generators is:

$$\dot{Q}_{GE1} = \dot{m}_{NH3}(h_{v1} + (f_1 - 1)\cdot h_{sor\_ge1} - f_1 \cdot h_{seuil1}) \tag{22}$$

$$\dot{Q}_{GE2} = \dot{m}_{NH3}(h_{v2} + (f_2 - 1)\cdot h_{sor\_ge2} - f_2 \cdot h_{seuil2}) \tag{23}$$

Heat released from the absorbers is:

$$\dot{Q}_{AB1} = \dot{m}_{NH3}(h_{sor\_ev} + (f_1 - 1)\cdot h_{ent\_ab1} - f_1 \cdot h_{sor\_ab1}) \tag{24}$$

$$\dot{Q}_{AB2} = \dot{m}_{NH3}(h_{v1} + (f_2 - 1)\cdot h_{ent\_ab2} - f_2 \cdot h_{sor\_ab2}) \tag{25}$$

We deduce the coefficient of performance as:

$$COP = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE1} + \dot{Q}_{GE2}} \tag{26}$$

Using the expression of $\dot{Q}_{EV}, \dot{Q}_{GE1}$ and $\dot{Q}_{GE2}$, the explicit formula of the coefficient of performance becomes:

$$COP = \frac{\left(h_{sor\_ev} - h_{sor\_cd}\right)}{\left(h_{v1} + (f_1 - 1)h_{sor\_ge1} - f_1 h_{seuil1}\right) + \left(h_{v2} + (f_2 - 1)h_{sor\_ge2} - f_2 h_{seuil2}\right)} \tag{27}$$

In this system, it is remarkable that there is a single evaporator and two generators so there is higher energy consumption. It is twice the consumption of a single stage, but the added value of this system is to lower the generator temperature. So we can conclude that for this preliminary study, the COP of this system is lower than that of a single stage.

3.3.3 System AAG

The system works as follows; rich solution is pumped from the absorber AB_2 (at the temperature T_{AB} and intermediate pressure Pmoy) and enters to the generator. Ammonia vapor goes to the condenser CD and the poor solution discharges through the absorber AB_1. The connection between the double-stages is insured between the absorbers AB_1 and AB_2 (see Figure 9). Figure 10 shows the Oldham diagram of the cycle. In such case, the COP is defined as.

$$COP = \frac{\dot{Q}_{EV1} + \dot{Q}_{EV2}}{\dot{Q}_{GE}} \tag{28}$$

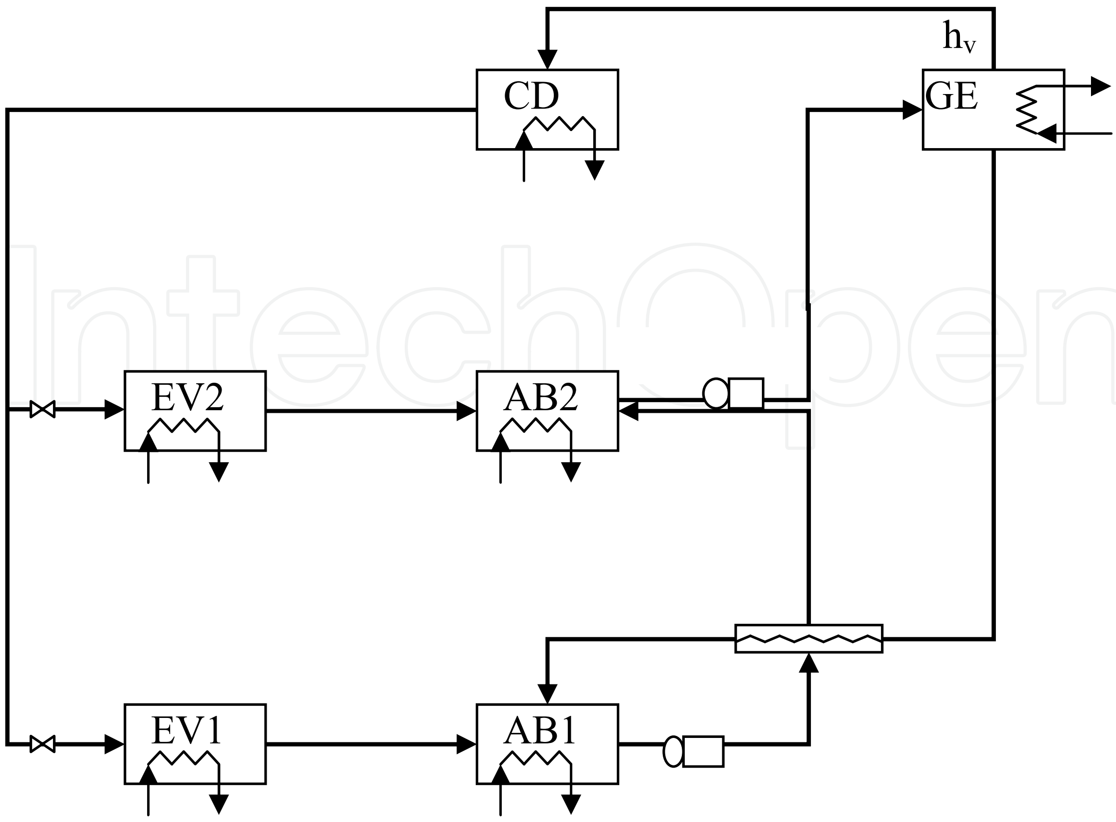

Fig. 9. System AAG (connection absorber-absorber)

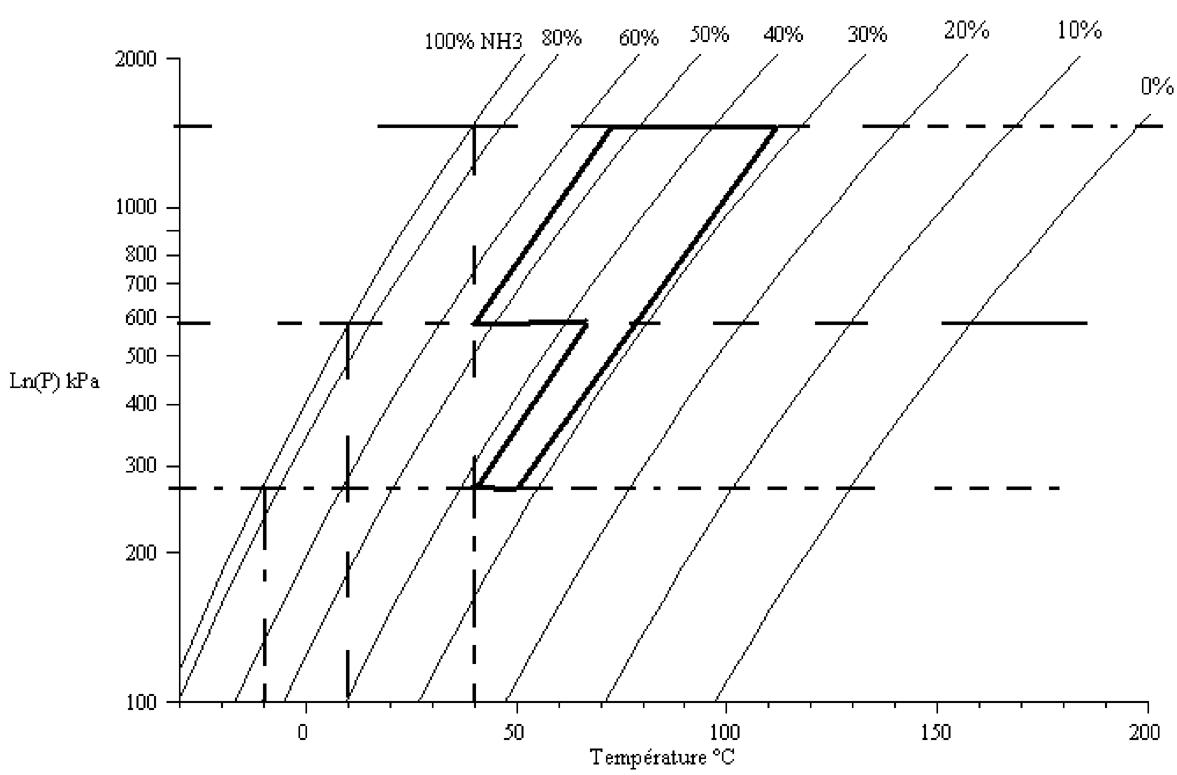

Fig. 10. Oldham diagram of system AAG

3.3.4 New system

The preceding sections, present different possible combinations with connection between the various components of the double-stage absorption refrigeration system; we propose a new designed system. The configuration consists to introduce a compressor between the first and the second stage. The considered new system is composed by two generators, two absorbers; a condenser, an evaporator and a compressor (see Figure 11).

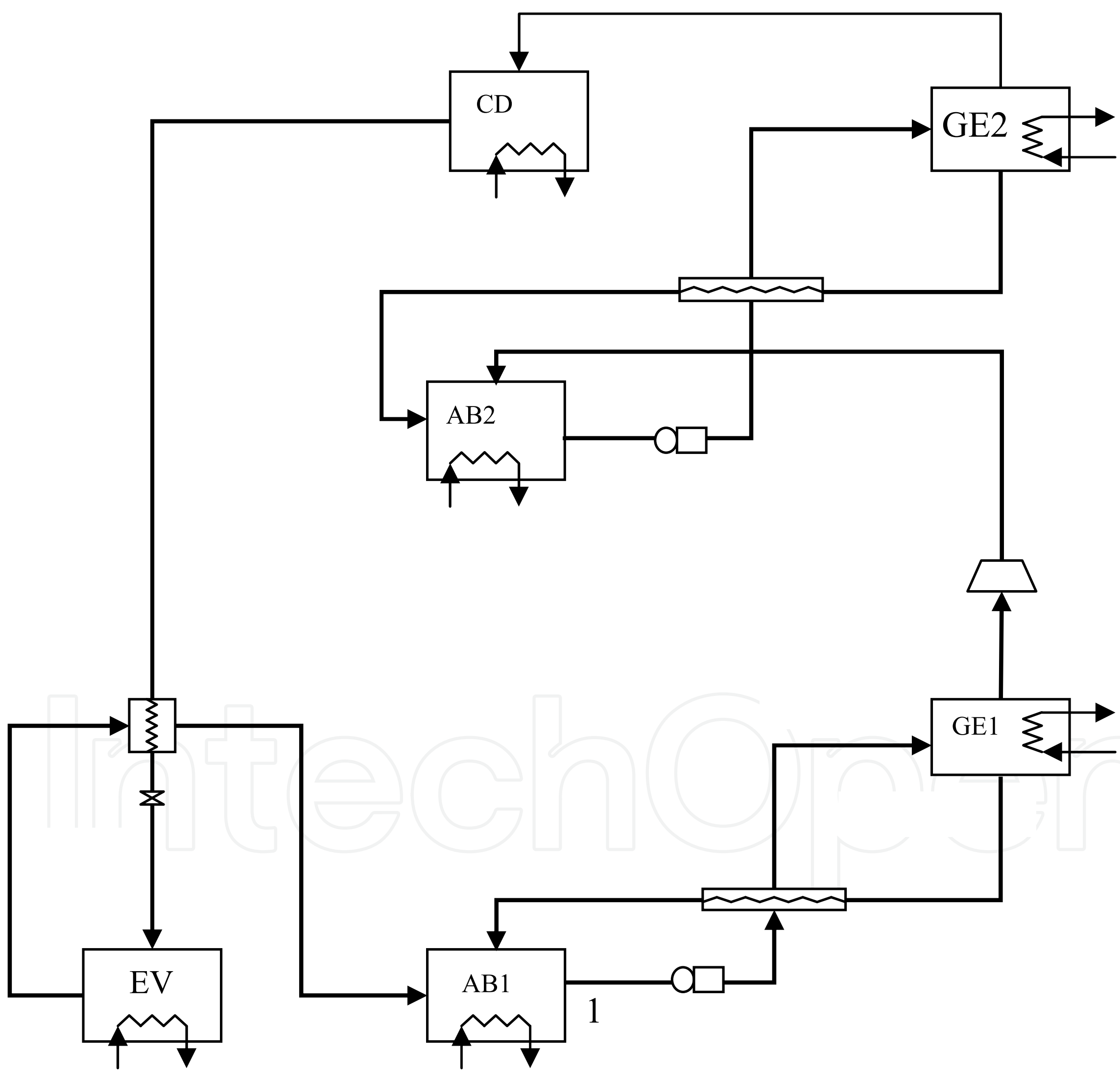

Fig. 11. New system

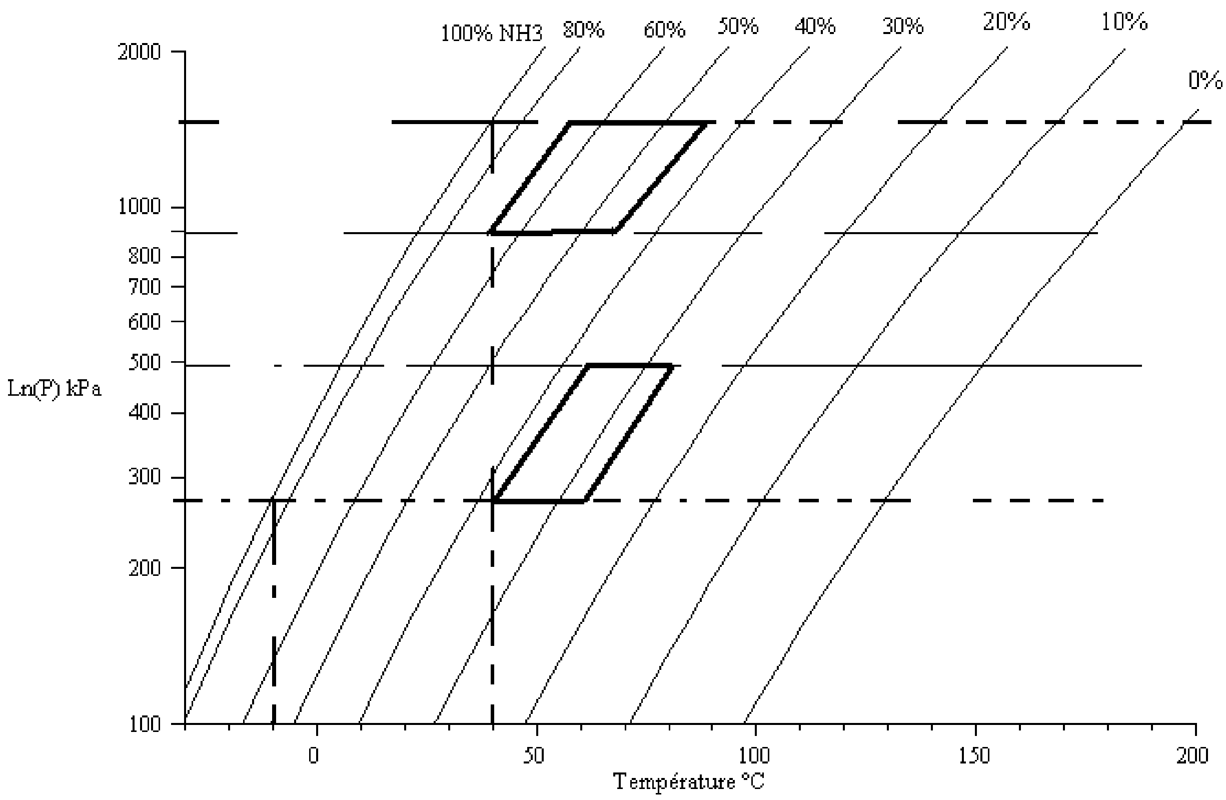

Fig. 12. Oldham diagram of new system

To determine the entrainment factors, mass flow rates and heat of the various components of such machine, we use the same equations as for the cascade AGAG, except the compressor's modeling, which must be studied separately. In fact, to determine the compressor power, we consider the ammonia at the generator exit (GE_1) as an ideal gas. For an isentropic process Laplace relation gives:

$$T_{ent\_comp} \times P_{ent\_comp}^{(1-k)/k} = T_{sor\_comp} \times P_{sor\_comp}^{(1-k)/k} \tag{29}$$

Where: T_{ent_comp}, P_{ent_comp} and, T_{sor_comp}, P_{sor_comp} are the compressor temperature and pressure inlet and outlet respectively.
Under assumption of isentropic processes (ideal case), the consumed power is given by:

$$\dot{Q}_{is} = \dot{m}_{HN3} cp_{NH3} \times (T_{sor\_comp} - T_{ent\_comp}) \tag{30}$$

But we must take into account the isentropic η_{is} where the real power:

$$\dot{Q}_{réel} = \frac{\dot{Q}_{is}}{\eta_{is}} \tag{31}$$

$$\dot{Q}_{réel} = \dot{m}_{HN3} (h_{sor\_comp} - h_{ent\_comp}) \tag{32}$$

So we can deduce from (20) and (21), the value of the steam enthalpy at the compressor outlet:

$$h_{sor\_comp} = h_{ent\_comp} + \frac{\dot{Q}_{is}}{\dot{m}_{HN3}\eta_{is}} \tag{33}$$

With

$$\eta_{is} = 0.874 - 0.0135 \cdot \tau \tag{34}$$

$$\tau = \frac{P_{sortie}}{P_{entrée}} \tag{35}$$

In this case, we note that the COP's formulation is different from other systems, since it depends on the mechanical work that is no longer negligible. Therefore, in addition to the two generators power, the compressor power ($\dot{Q}_{comp}$) is considered. The COP's expression becomes:

$$COP = \frac{\dot{Q}_{EV}}{\dot{Q}_{GE1} + \dot{Q}_{GE2} + \dot{Q}_{comp}} \tag{36}$$

3.3 Results and discussions

Several studies have been devoted to determine the COP and limitations of absorption system operating conditions [21-23]. In order to evaluate the refrigeration absorption system performance, relative to different previously presented configuration, we have developed a numerical program. The calculating procedures of the fluid thermodynamic properties and the performance coefficient were obtained using MAPLE computer tools.

3.3.1 Single-stage machine

By setting the three temperature levels T_{EV}, T_{AB}, T_{GE} and different operating conditions we determine the thermodynamic properties of the studied refrigerating system allowing the evaluation of its performance coefficient.
We note that the COP depends mainly on the evaporating temperature (necessary for the production of desired cold), the condensation temperature (function of cooling temperature of the absorber and condenser components) and finally generator temperature.
For a fixed generator temperature T_{GE} with a condensation temperature data, we analyze numerically the COP's variation of the single stage machine versus the evaporator temperature (see Figure 13).According to figure 13 and 14, we note that the coefficient of performance of a single-stage absorption system increases with the evaporator temperature rising and increases with the condenser temperature decrease.
It is noted from Figure 13, that the COP's system is higher for low values of T_{CD} and high values of T_{EV}. It is apparent that the range of the single stage machine operating conditions is adaptable to different generator temperatures. We note that for a generator temperature of 100 C and a condensing temperature higher than 40 C, the machine can operate at an evaporator temperature above -5 C. Under these conditions, the corresponding COP is approximately 0.45. We can conclude that for a temperature of 100 C at the generator, 40 C or higher for condensation, the single-stage machine is rather favorable to the air conditioning (T_{EV}> 0) than refrigeration. The COP may reach 0.55 for a condensing

temperature of 45 C and an evaporator temperature of 15 C. Besides, by increasing the generator temperature of 10 C, the same system can work at a temperature of -15 C (evaporation) with the same constraint in the condensing temperature (40 C) in order to reach a COP of 0.3. To produce cold, the absorption system loses almost one-third of the COP's machine.

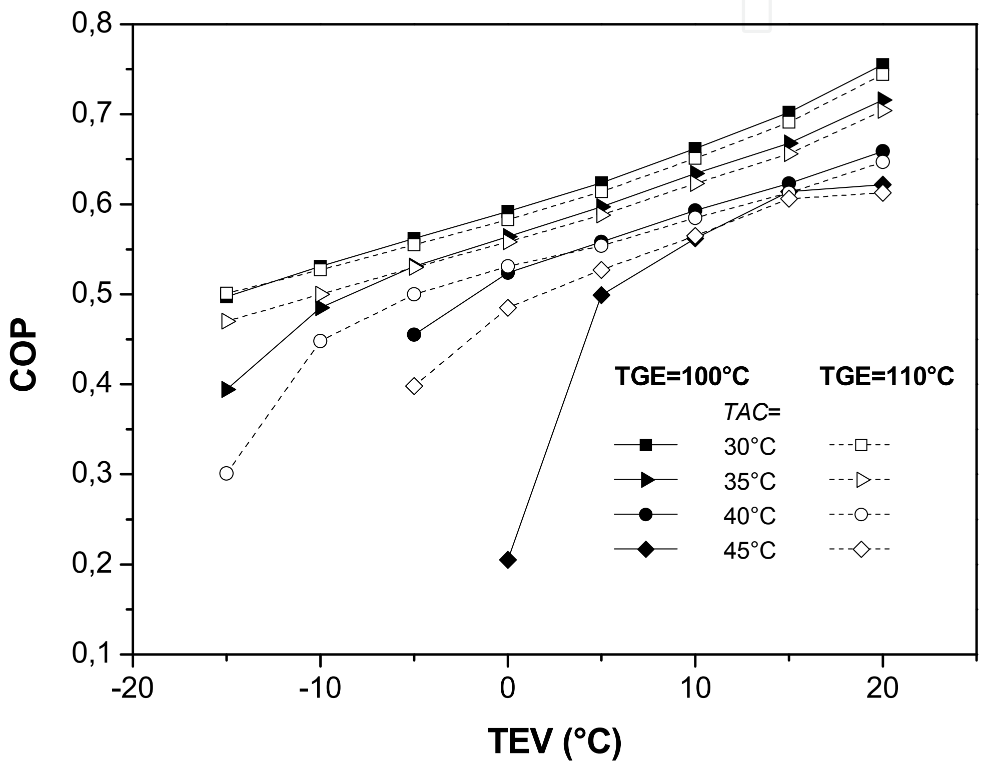

Fig. 13. COP evolution versus T_{EV} for different T_{AC} temperature and T_{GE} =100 C and 110 C

In the following, we fix the evaporator temperature for each family of curve. The numerical results illustrate the evolution of the performance coefficient for different generator temperatures. Each family has different curves for different condensation temperatures chosen between 30 C and 40 C (see Figure 14).
We note that the coefficient of performance increases with the generator and the evaporator temperature increase. While the optimal functioning depends on the condensation temperature, in fact, if it increases, the COP decreases. The cold production begins at a generator temperature greater than 110 C. On the other hand, Figures 13 and 14 show that the single stage absorption system has limited operating evaporation, condensation and generator temperatures.

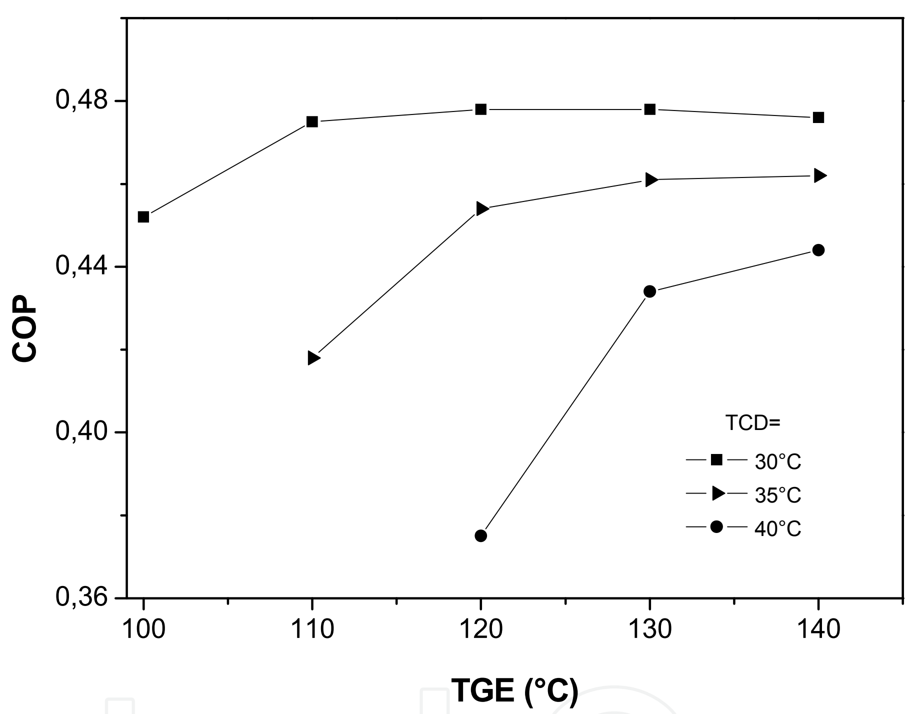

Fig. 14. COP evolution versus T_{GE} for T_{EV}=-20 C

3.3.2 Multi-stage machine

For cascading cycles, we note that there are many configurations, the difference between them is the connection between the various components of the refrigeration installation, Figures 15 and 16 show the evolution of the COP versus the possible generator and evaporation temperatures.

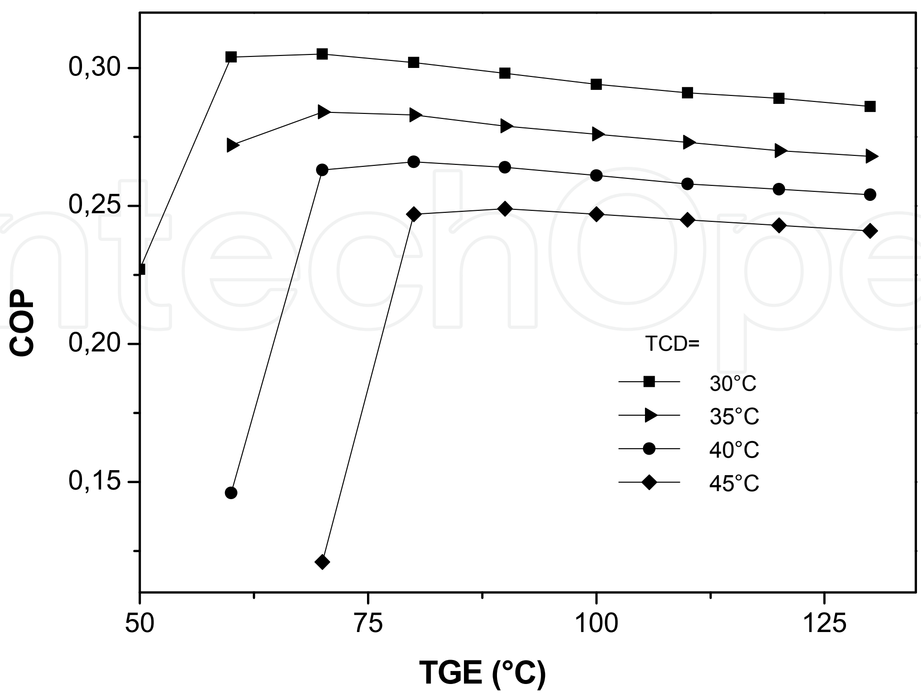

Fig. 15. COP evolution versus T_{GE} for P_1=500 kPa, P_2=900 kPa and T_{EV}=-10 C.

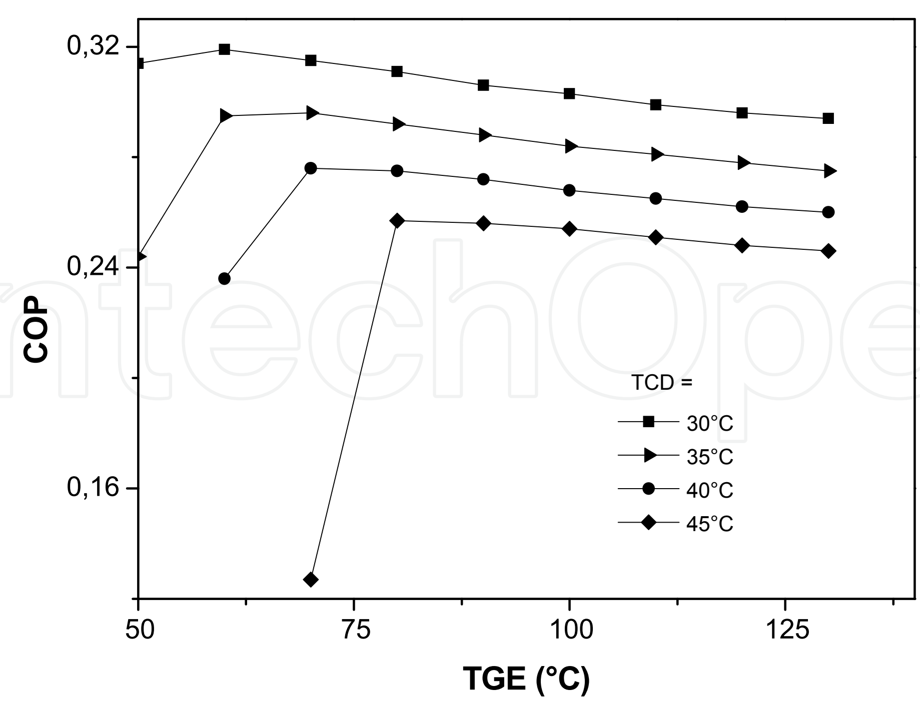

Fig. 16. COP evolution versus T_{GE} for P_1=500 kPa, P_2=900 kPa and T_{EV}=-5 C

The analysis of the COP's evolution versus different temperatures shows that the coefficient of performance increases when the condensation temperature decreases and the temperature evaporation increases. Besides, we note that in the first part of the curve, the COP increases with an increase of the generator temperature until the value of 85 C.
Figures 15 and 16 show that the COP is maximum for the generator temperature range varying between 60 and 85 C.
Figures 17 and 18 represent the COP's evolution versus the intermediate pressure, P_1, P_2. They show that the pressure averages don't have a great influence on the increase of the absorption refrigerating system performance; the advantage of this installation is that it can increase the difference of title between rich solution and weak solution. It is remarkable that this machine can operate at low temperatures. The efficiency of the hybrid absorption system proposed can reach 8.2, while the efficiency, proposed in literature which cannot exceed 5.
In the following section, we present respectively the COP evolution versus the generator temperature and the intermediate pressure (figures 19,20 and 21).

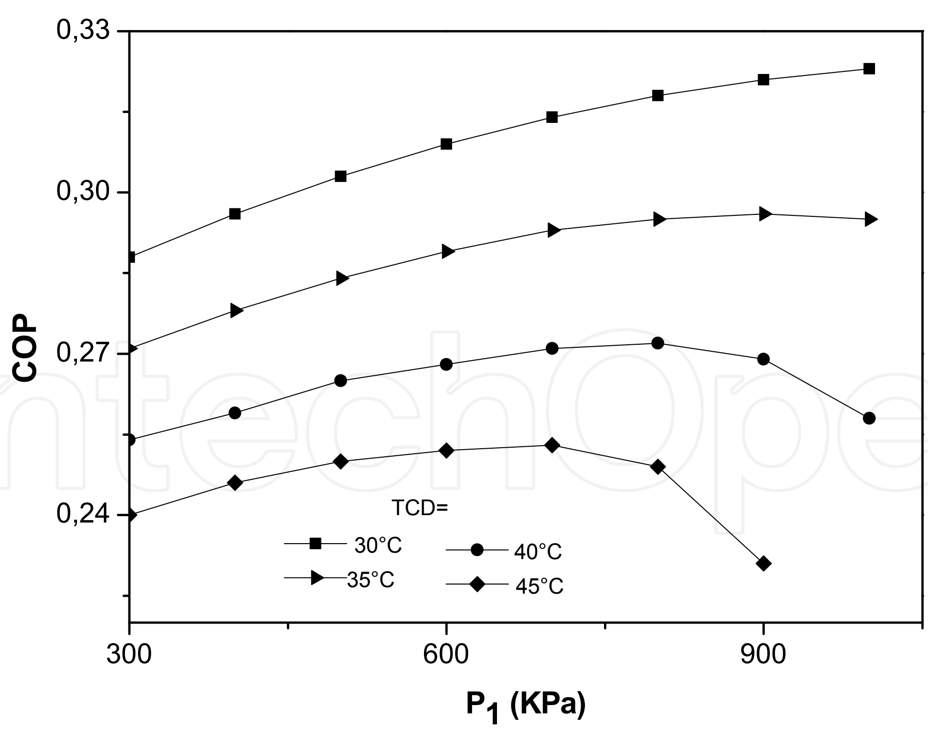

Fig. 17. COP evolution versus P_1 for T_{EV}=-10 C, P_2=1000 kPa and T_{GE}=90 C

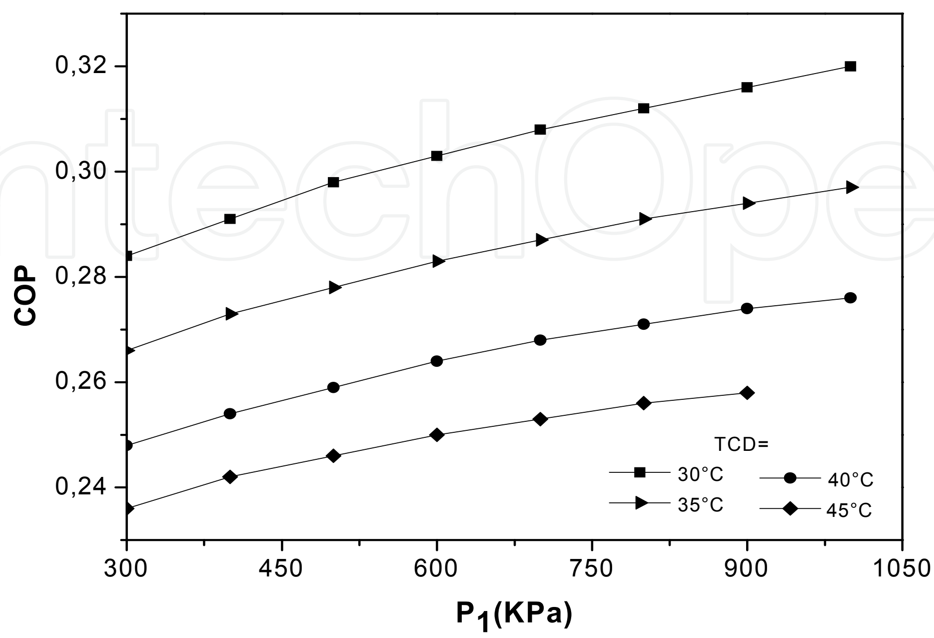

Fig. 18. COP evolution versus P_1 for T_{EV}=-10 C, P_2=1000 kPa and T_{GE}=110 C

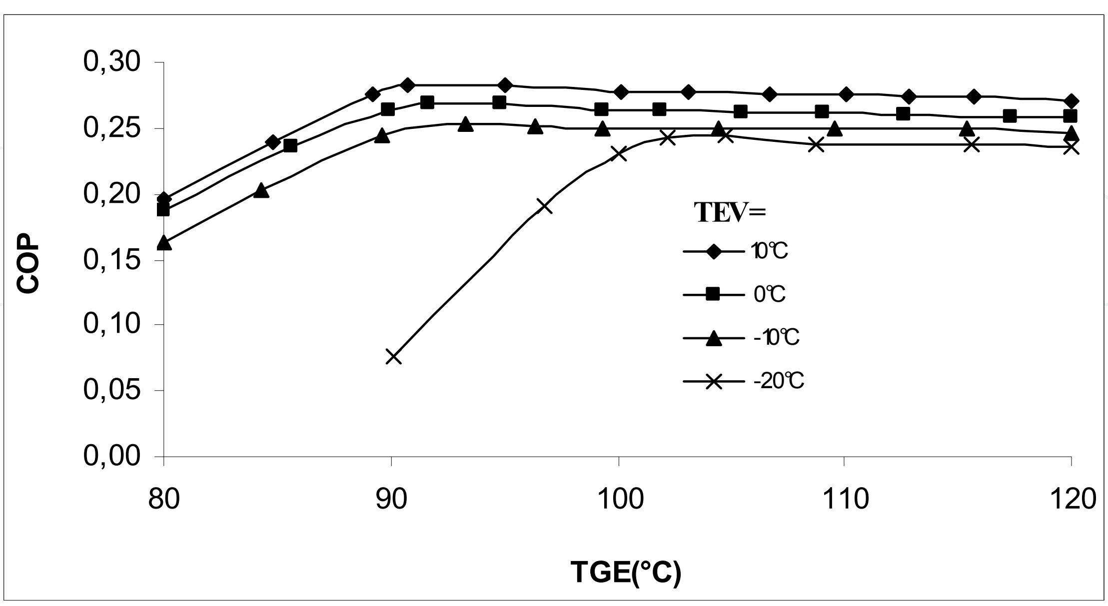

Fig. 19. COP evolution versus T_{GE} with T_{CD}=45 C and P_{moy}= 700 kPa

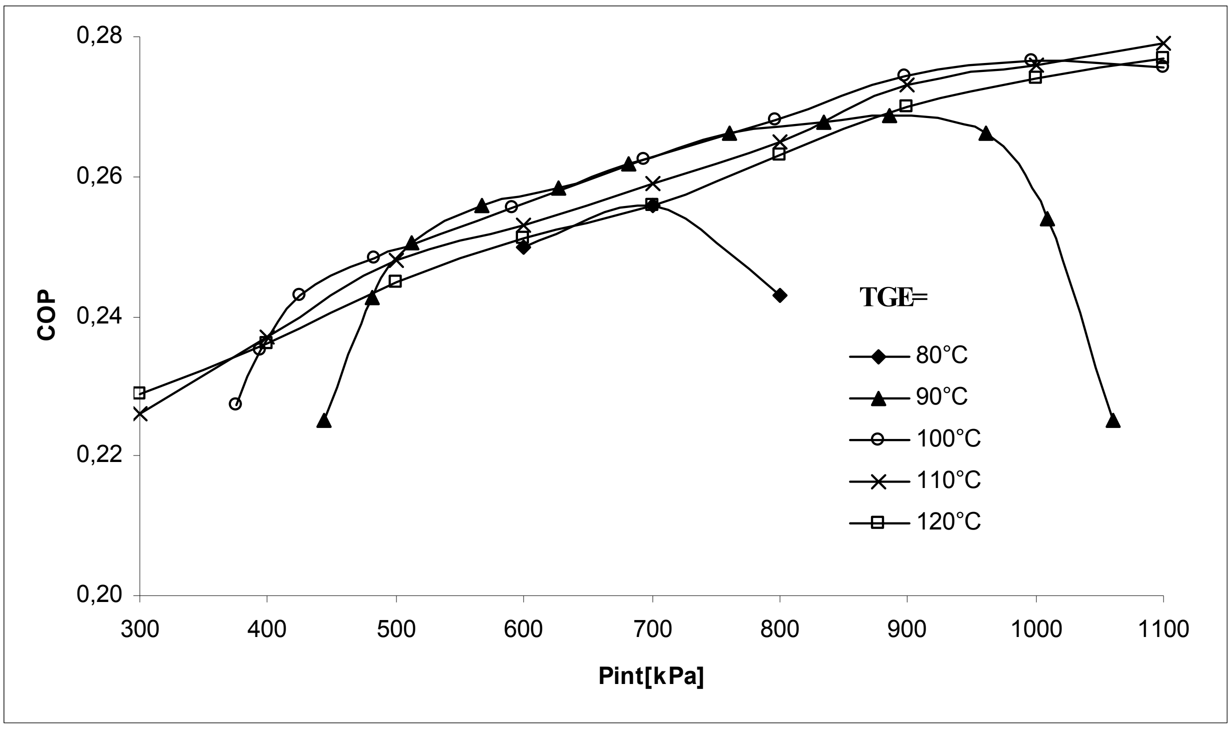

Fig. 20. COP evolution versus P_{int} with T_{CD}=40 C and T_{EV}= -10 C

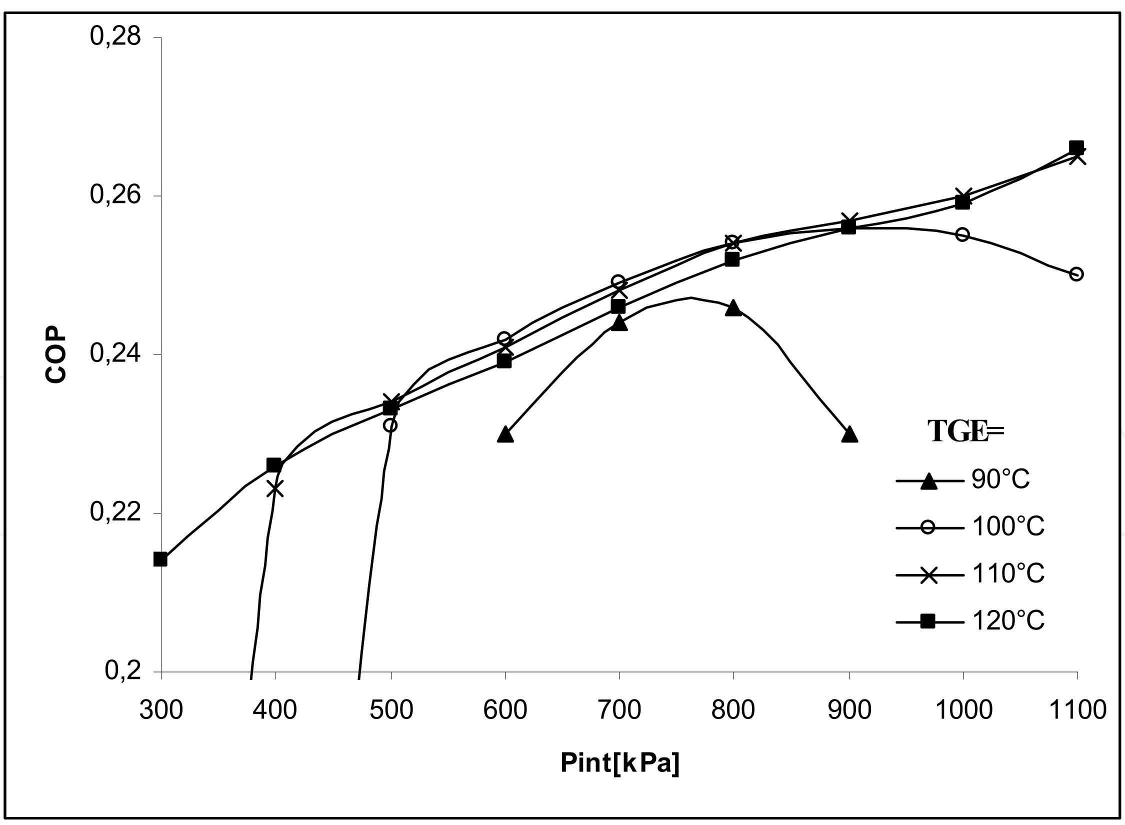

Fig. 21. COP evolution versus P_{int} with T_{CD}=45 C and T_{EV}= -10 C

From figures 19, 20 and 21, we conclude that the COP is always less than 0.28.

4. Conclusion

In this investigation, single and double-stage absorption cycles using water-ammonia are analyzed. We presented different configurations and we proposed a novel hybrid absorption refrigeration cycle. The proposed absorption/compression refrigerating system object of this work is studied in details. In order to evaluate the performance of the invoked machine, a procedure based on the MAPLE software is set up to compute accurately the thermodynamic properties of different states. The comparative study of the performance relative to different absorption cycles is carried out and the numerical results highlight that the single-stage machine has a COP higher than that of the double-stage absorption system; however, it requires a temperature generator relatively high. On the other hand, the average pressure has no influence on the increase of the system performance while it reveals very important on reducing the generator temperature.
Finally, we can conclude through this study that sources at moderate temperatures (solar, geothermal or other) can be used to power refrigeration systems absorption. The COP is acceptable and it is approximately 0.28.

5. Nomenclature

COP = Coefficient of Performance
f = Circulation ratio
h = Specific enthalpy (Jkg^{-1})
$\dot{m}$ = Mass flow rate (kgs^{-1})
P = Pressure (bar, Pa)
$\dot{Q}$ = Heat-transfer rate (W)
T = Temperature (K, C)
x = mass fraction
Greek symbols
Δ = Variation
ξ, η = Efficiency
Subscripts
1 = First stage
2 = Second stage
AB = Absorber
CD = Condenser
EV = Evaporator
GE = Generator
p = Poor
r = Rich
Sor = Outlet
v = Vapour

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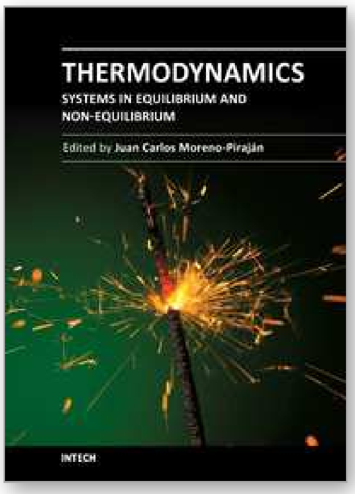

Thermodynamics - Systems in Equilibrium and Non-Equilibrium

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