Effect of Pressure on Improving COP of a Refrigeration Installation Solar To Absorption

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Abstract—theuse of renewable energycreatesmoreof interestin the world.Disruptionsin oil prices, gas and environmental problemshave led manycountriesto focuson renewableand clean energysuch as solarenergy.
Howevercold productionfromthis energy ispossiblefor our country.
Among the varioussystems used tothis, theabsorption refrigerationmachineseems to beverypromisingin this respect. A request forincreasedcomfort andparticularly hightemperatures wereled tostrong growthof air conditioning indeveloped countries.
The objective of thiswork istostudya solarrefrigerationdoubleeffect(twoboilers, condenser, evaporator and twoabsorbers) operating with simpleabsorptionpanels, to seethe influence ofthermodynamic parametersin particularpressure on theimprovementcoefficientsof performance (COP) and showthat itproduces thecoldwithinteresting performances.Itdoes not requiredistillation column andhas the following advantages: possible operationfroma temperature in theboilers70°C, possibility of usingsolar collectorscheaperwhich affectsthe overall costof the installation.

Keywords-
ColdSolar,absorptionrefrigeration,COP,improvement.

I. INTRODUCTION

The thermal energyrequired to operatethesetechnologiescomes fromsolar radiation. the solar system Converted the solar thermal radiation through a capture by greenhouse effect is called solar radiation sensor.

Solar systemsfor refrigerationreceive energyand convert it intocoldstoreandthe rest isforuse duringthe nights and the bad weather periods.

Generally, theyare threenecessary elements: solar, energy storagesystem, absorption machine[1].
In order toseethe influenceofthepressureontheimprovementofcoefficientofper formance. Theconfigurationconsideredisthat of anabsorption refrigeration machinewith doubleeffect. It works withthreepressures.

High-pressureofbouilleur2andcondenser
low pressureofthe evaporatorandtheabsorbeur1.
The use of asecondbouilleur1powered bybinarysolutioncan raisethe thirdpressureabsorber2itthen beingan intermediate pressurebetween the low pressureevaporatorand the high pressuregenerator2(Figure 2).

II. ABSORPTIONSYSTEM

Refrigerationabsorptionphenomenathatusebinarysolutionsabsorbingvaporsandothersolutionswithinthisdefinedtemperatures.Itthen callsabsorption, thefixationof a substancebythe entire volume ofthe absorbent material.

We knowthatthevaporof a pure substancecanbeabsorbedbythe bodyin the liquid statethan in the casewherethe liquid temperatureis lower than thatofthevapor, unlike thepure substances, solutions havingtheremarkableproperty of absorbingthevaporoftheliquidsolutionofa differentcompositionevenifthe liquid temperatureishigher thanthevapor [2].

Liquidabsorptionmachinesoperate usingthe powerof certain liquidabsorbing(exothermicreaction)anddesorption(endothermic reaction)vapor.
Anabsorption systemcomprises afirstassembly (condenser, expansion valve, evaporator)whereinisthepurerefrigerant.
Thesemachines useabinaryworking fluidmixture, whichisacomponentof themuch more volatilethan the other,isthe refrigerant [1].
Two couplesare mainly used:
•Water /LithiumBromide(H_2O/LiBr)
•Ammonia/ Water (NH_3/H_2O)
We can expressthe composition of thesolutioninthe most volatile component(refrigerant), ortheless volatilecomponent(absorbent).
•Diluted solution(absorbing) or richsolution(refrigerant).
•Concentrated(absorbing) or poorsolution(refrigerant).

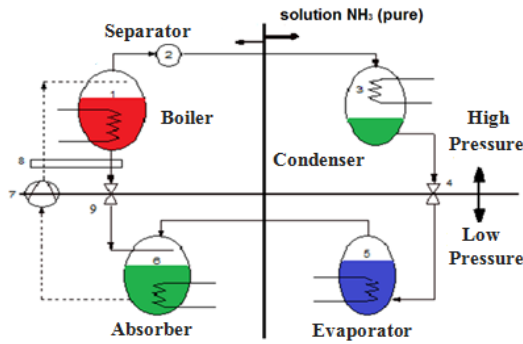Fig1. Absorption refrigeration machineNH₃/H₂O.

III. INSTALLATIONABSORPTIONDOUBLE EFFECT

The absorption machine is, for us, the heart of the whole system of solar cooling. It therefore seemed interesting to see how these machines are modeled. In this regard, there are generally two approaches [3]: The first is based on a phenomenological description of each component of the machine. This approach is based on different energy balances to couple the four components of the machine. The second which is probably the most usedperformance evaluation by an empirical modelwhich isgenerally smoothing curves based on manufacturer's data. Indeed, these are tests that establish a simple correlation customary for the COP and the cooling capacity of the machine. Correlations are thus valid for the range of tests and tested the machine model studied.
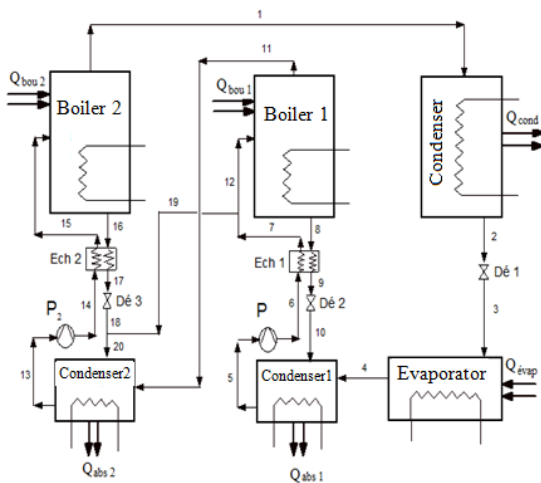

Fig 2: Installing doubleeffect absorption(studied)

To establishthe heat balanceof any componentandtosize theheat transfer surfacesthat are associated with, it is necessary to know preciselytheenthalpyofworking fluidin the liquidandgaseous stateas a function oftemperatureand concentration.

Among theexistingmodels, we have chosenthe one

proposedbyM.Feidt[4]which combinestheGibbs free energyforthe thermal properties and the equations that calculatethebubble pointandthedew point of themixture. This method combinesthe advantages of bothandeliminatesthe need foriterationsin order to haveconditionsofequilibriumphases.

The knowledge ofclimatic and geographicalcharacteristics, of the studied region, is averyimportant factor inthe study.These elementsare variable,butwe can estimatea monthly average[5].

Afterdeterminationsofdifferentmass and energyequationsof the different elementsoftheinstallation,table belowpresents thedifferentequationsformass flow ratesatdifferentpoints of theinstallation.

TABLE1.DETERMINATIONSFLOW RATESOF THE EQUATIONSOF DIFFERENT ELEMENTSOF THE INSTALLATION.

elements	mass flow(kg/s)
1	$\dot{m}_1$
2	$\dot{m}_2 = \dot{m}_1$
3	$\dot{m}_3 = \dot{m}_1$
4	$\dot{m}_4 = \dot{m}_1$
5	$\dot{m}_5 = \dot{m}_1\left[1 + \dfrac{(x_4 - x_5)}{(x_5 - x_{10})}\right]$
6	$\dot{m}_6 = \dot{m}_5$
7	$\dot{m}_7 = \dot{m}_5$
8	$\dot{m}_8 = \dot{m}_{10}$
9	$\dot{m}_9 = \dot{m}_{10}$
10	$\dot{m}_{10} = \dot{m}_1\left[\dfrac{(x_4 - x_5)}{(x_5 - x_{10})}\right]$
11	$\dot{m}_{11} = \dfrac{\dot{m}_1}{2}\left[1 - \dfrac{(x_4 - x_5)}{(x_5 - x_{10})}\right]$
12	$\dot{m}_{12} = \dfrac{\dot{m}_7}{2} = \dfrac{\dot{m}_1}{2}\left[1 + \dfrac{(x_4 - x_5)}{(x_5 - x_{10})}\right]$
13	$\dot{m}_{13} = \dot{m}_1\left[1 + \dfrac{(x_1 - x_{15})}{(x_{15} - x_{16})}\right]$
14	$\dot{m}_{14} = \dot{m}_{13}$
15	$\dot{m}_{15} = \dot{m}_{13}$
16	$\dot{m}_{16} = \dot{m}_1\left[\dfrac{(x_1 - x_{15})}{(x_{15} - x_{16})}\right]$
17	$\dot{m}_{17} = \dot{m}_{16}$
18	$\dot{m}_{18} = \dot{m}_{16}$
19	$\dot{m}_{19} = \dfrac{\dot{m}_7}{2} = \dfrac{\dot{m}_1}{2}\left[1 + \dfrac{(x_4 - x_5)}{(x_5 - x_{10})}\right]$

20	$\dot{m}_{20} = \dfrac{\dot{m}_1}{2}\left[1 + \dfrac{(x_4 - x_5)}{(x_5 - x_{10})} + 2\left[\dfrac{(x_1 - x_{15})}{(x_{15} - x_{16})}\right]\right]$

IV. THERMODYNAMIC STUDY OF THE SYSTEM STUDIED

For the application of the principles of thermodynamics on a real cycle, conditions and assumptions were used:

1. Temperatures in parts of the plant (boiler, condenser, absorber and evaporator) are assumed uniform throughout the volume in question.

2. Solution rich in refrigerant at the outlet of the absorber is saturated at the temperature and the concentration in the absorber liquid. Likewise, the weak solution leaving the refrigerant generator is connected by a balance of pressure and temperature relationship of the concentration generator.

3. The coolant leaving the boiler is taken as saturated vapor at the temperature and corresponding pressure.

4. The coolant leaving the condenser is taken as the saturated liquid at the same temperature and pressure.

The refrigerant at the outlet of the evaporator is in the form of saturated steam at the temperature and low pressure of the evaporator. The isenthalpic expansions are assumed.

Heat exchange with the environment and losses are assumed negligible.

Calculations are based on the determination of the respective enthalpies of the liquid phase and vapor phase from the analytical expressions of the Gibbs energy[6], knowing the pressure, temperature and the concentration of the solution. We must also determine the quantities of vapor-liquid equilibrium of binary pair ammonia water from the Peng-Robinson equation[7] and the interaction coefficient K_{ij} characterizing the mixing torque.

For the determination of various system parameters enthalpies (Hi) and titles (Xi), there are two methods: either from empirical equations or from the diagram of Meckel and Oldham.

In our case, we used diagrams for the determination of these parameters.

Our work is spent on a pressure range that varies from 2 to 20 bars.

The choice of a high pressure of 20 bars, for this case, leads us to a Title X tends to 1.

V. COEFFICIENT OF PERFORMANCE

$$COP = \frac{Q_{\text{évap}}}{Q_{\text{bou 1}} + Q_{\text{bou 2}} + W_{p1} + W_{p2}} \quad (1)$$

$$COP = \frac{\dot{m}_3(h_4 - h_3)}{\dot{m}_8 h_8 + \dot{m}_{11} h_{11} - \dot{m}_{12} h_{12} + \dot{m}_1 h_1 + \dot{m}_{16} h_{16} - \dot{m}_{15} h_{15} + \dot{m}_6(h_6 - h_5) + \dot{m}_{13}(h_{14} - h_{13})} \quad (2)$$

VI. SIMULATION

The simulation is based on the heat balance for the different phases of the absorption cycle[6]: Our simulation is made with the aim of seeing the variation of COP depending on the different temperature levels to elements of the installation with constant pressure absorption.
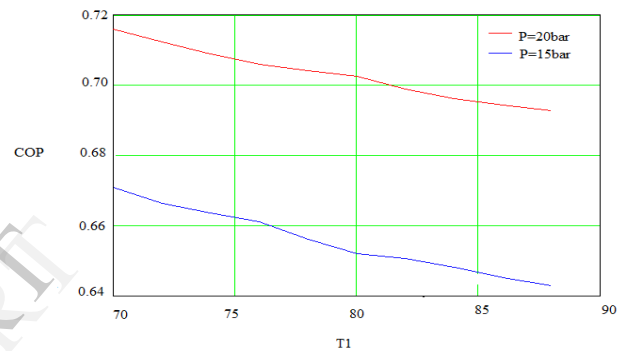
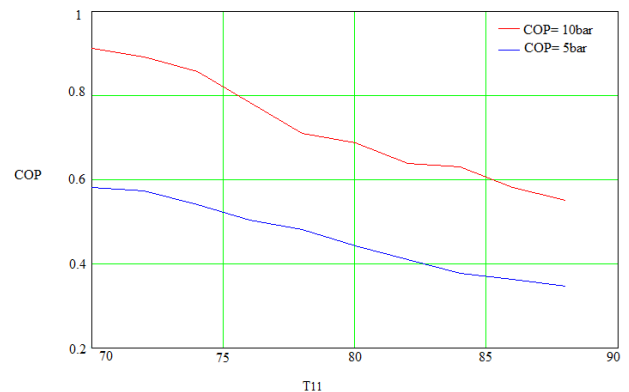Fig 3: Evolution of COP on the temperature variation of boiler 2 (T1).

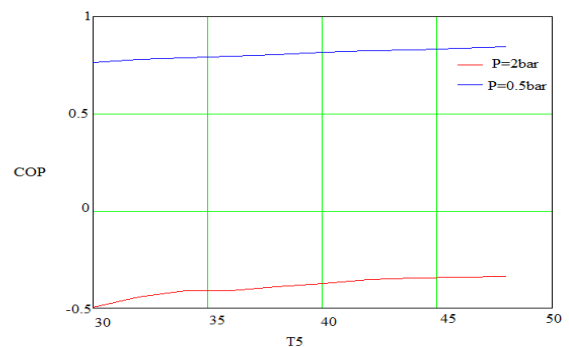Fig 4: Evolution of COP on the temperature variation of boiler1 (T11).

Fig 5: Evolution of COP on the temperature variation of the absorber1 (T5).
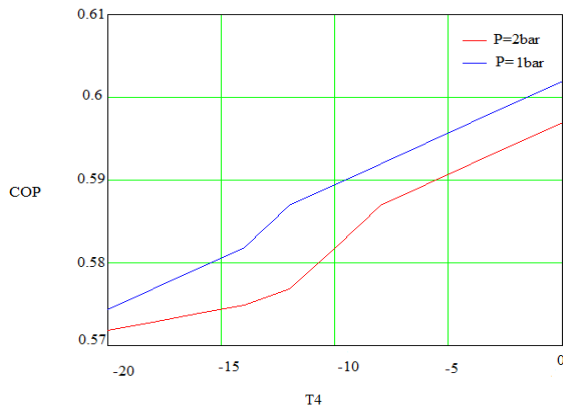
Fig 6: Evolution of COP on the temperature variation of the evaporator (T4)

VII. INTERPRETATION OF RESULTS

It is important to choose the right temperatures and pressures of running an absorption machine (NH_3/H_2O) at boiler, evaporator and absorber.

Figure [3]

It is noted that the variation of COP is inversely proportional with the temperature at the boiler 2: The COP decreases with increasing temperatures of the boiler 2 (T1) in the temperature range $70\ °C \leq T1 \leq 88\ °C$. In this interval, it is noted that decreases with the pressure decrease.

Figure [4]

It is noted that the variation of COP is inversely proportional with the temperature at the boiler 1: The COP decreases with increasing temperatures boiler 1 (T11) in the same temperature range We note also that the COP decreases with the pressure decrease.

Figure [5]

It is noted that the variation of the COP is proportional with the temperature at the absorber 1: the COP increases with increasing temperatures absorbeur1 (T5) in the temperature range $30°C \leq T5 \leq 48°C$ and in this interval it is noted that the variation of the COP is inversely proportional with pressure: COP decreases with increasing pressure.

Figure [6]

It is noted that the variation of COP is proportional with the temperature at the evaporator: the COP increases with increasing temperatures of the evaporator (T4) in the temperature range $-20°C \leq T4 \leq 0°C$ and decreases with increasing pressure.

VIII. GENERAL CONCLUSION

In order to improve COP and reduced cost for solar absorption refrigeration, this work is devoted to geometric changes to an absorption chiller (dual engine effects constitute two boilers, condenser, evaporator and two absorbers) as the latter can operate with solar panels plans.

This study allowed us to analyze the absorption refrigeration systems coupled with solar energy. Thermodynamic analysis of the refrigeration cycle with double effect absorption running torque (NH_3/H_2O) showed that the boiler temperature is inversely proportional to the coefficient of performance of the installation the latter more(COP) with the proportional pressure level of boilers. On the other hand, the temperature of evaporator and absorber are in proportion with the COP of the installation. The COP is very important for low pressure at the evaporator.

The geometric modifications we made on this system (number of boilers)does not require distillation column.

Their operationis possible fromtemperatureswhich varygoshawks70°C.The possibility of using solar collectors, less expensive and available in the market,allows us to reduces the overall cost of this type of facilities.

NOMOCLATURE

Nomenclatures	Désignations	Unité
T	Température	K
P	Pression	Pa
X	Titre du frigorigène ou de l'absorbant dans la solution	%
Q	Puissance	Kw
Qe	Puissance frigorifique de l'évaporateur	Kw
Qb	Puissance frigorifique de bouilleur	Kw
Qc	Puissance frigorifique de condenseur	Kw
Wp	Puissance de la pompe	Kw
H	Enthalpie	KJ/kg
$\dot{m}$	Débit massique	Kg/s
COP	Coefficient de performance	-

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