Thermodynamic Modelling and Performance Assessment of a Solar-driven Ammonia-water Absorption Chiller

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Abstract

A thermodynamic modelling of an ammonia-water absorption chiller has been carried out using the EES (engineering equation solver) environment. The equations governing the thermal and mass transfers in the different chiller components and internal piping have been solved.

The model calculates the properties of the binary solution (ammonia-water) at all the state points of the single-effect absorption chiller. The model evaluates the global performance of the system and calculates the amounts of heat absorbed or released by the chiller's components as well as the total performance coefficient COP.

The model has been coupled to TRNSYS (Transient system simulation program) and simulations have been carried out to evaluate the performance of a solar-driven ammonia-water absorption chiller for an office building. The simulation results have shown that such system can reach a COP as high as 0,45.

The major results of this study have been published in [9].

In this paper, the absorption chiller model will be explained and the effects of the variation of the machine's parameters on its performance will be studied. We will also study the possibility of application of the solar-driven absorption chiller in building air conditioning by introducing the results found by our model in a series of dynamic simulations on TRNSYS.

Key-words: Absorption chiller, solar cooling, single-effect chiller, ammonia-water, modelling, simulation.

1. Introduction

Absorption systems represent an ecological technology that can be employed in several applications. Based on the cooling temperature demand, these applications can be broadly classified into three categories: buildings air-conditioning (8 to 15°C), refrigeration for food and vaccine storage (0 to 8°C) and freezing for ice making or other purposes(<0°C). In addition, the high correlation between the availability of solar energy and the need for cooling in buildings provides an inherent advantage to solar driven cooling systems. For this reason, numerous numerical and experimental studies have been performed to improve absorption systems performance especially in space air-conditioning.

The objective of the current work is to write a numerical model of an ammonia-water absorption chiller in order to introduce it in a simulation of a complete solar assisted system made up of a solar collector, a storage tank and an absorption chiller. We will study the performance of such system in covering a building load during summer season. The aim of this work is to reach the main advantages of system numerical modelling which are the elimination of expense of building prototypes, the optimisation of the system components, the estimation of the amount of energy delivered from the system and the prediction of temperature variations of the system.

2. Literature Review

Many researches were developed in the solar assisted absorption refrigeration systems field and led to the production of numerous experimental units as well as numerical codes simulating their functioning. Some of these studies are presented here.

G.A. Florides and al.[2] studied the performance of a solar assisted system used in a 196m^2 building located in Cyprus for air-conditioning in summer and hot water supply in winter. The modelling of the system on an annual basis has shown that the application of the solar system during the whole year for air-conditioning and domestic hot water production guarantees important life cycle savings.

K. Sumathy and al. [7] experimented in China a low temperature driven solar cooling and heating system employing a new model of two-stage lithium bromide absorption chiller, covering a 24-storey building load during the whole year. Experiments have shown that this system is efficient and cost effective since it could be driven by low temperature hot water ranging from 60 to 75°C, which can be easily provided by conventional solar hot water systems. Thus, the proposed system with a two-stage chiller achieved roughly the same total COP as of the conventional system (employing single-stage chiller) with a cost reduction of about 50%.

Z.F. Li and al. [14] analysed the use of a stratified storage tank in a solar system using a simple effect LiBr–H_2O absorption chiller in Hong Kong. Experimental studies were carried out to compare such system to an ordinary system employing a traditional storage tank. Results have shown that the studied system is 15% more efficient in air conditioning than the ordinary system especially in cloudy days.

3. Ammonia-water absorption cycle description

An absorption cycle consists mainly of an evaporator, a condenser, a desorber and an absorber. The configuration studied in this article is shown in figure 1, it contains in addition to the principal components a rectifier and two heat exchangers. The first heat exchanger (liquid-vapour: Ech-1) sub-cools the liquid leaving the condenser by overheating the vapour leaving the evaporator. The second heat exchanger (rich solution-weak solution: Ech-2) optimizes the functioning of the system by preheating the rich solution leaving the absorber and cooling the weak solution extracted from the desorber. This leads to decrease the energy consumption within the desorber and facilitates the exothermic operation of absorption by lowering the temperature of the weak solution entering the absorber. The rectifier is added to prevent water from entering the evaporator, it is a vital organ in the absorption cycles operating with ammonia-water given that the vapour of ammonia leaving the desorber is not pure; it contains some tracks of steam (≈3%) [5,6]. The rectifier ensures the condensation of these tracks and the water obtained is then returned back to the desorber.

The system operates between two pressure levels: the condenser and the desorber on the higher pressure P_{high}, and the evaporator and the absorber on the lower pressure P_{low}. The evolution of the binary solution between the two pressure levels is guaranteed by a solution pump and two expansion valves (D_1 and D_2). In addition, the absorption system interacts with heat sources/sinks at three temperature levels: the high temperature (solar) heat supply in the desorber T_{Des}, the intermediate temperature heat rejection in the absorber T_{Abs} and the condenser and the evaporator temperature T_{Evap}.

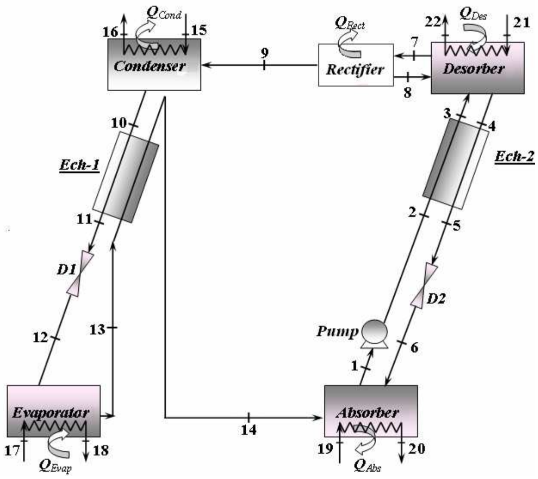

Fig. 1. Single-effect Ammonia-water absorption chiller

The quantity of heat $\dot{Q}_{Des}$ supplied in the generator section is added to the working fluid solution at a relatively high temperature T_{Des}. This heat causes the ammonia (refrigerant) to be boiled out of the solution in a distillation process. The vapour (state point 7) transfers to the rectifier where the tracks of water are condensed and returned in a liquid state to the desorber (state point 8). The ammonia vapour that results (state point 9) passes into the condenser section where the vapour is condensed back to a liquid state, delivering an amount of heat $\dot{Q}_{Cond}$ at an intermediate temperature T_C. The liquid (state point10) passes through the liquid-vapour heat exchanger where it is sub-cooled by overheating the vapour leaving the evaporator, then, it flows down through the expansion valve (D_1) to the evaporator section where it passes over tubes containing the fluid to be cooled, while the weak solution leaves the generator (state point 4) and passes through the solution heat exchanger to the absorber section where it absorbs the vaporised ammonia (leaving the evaporator and the liquid-vapour heat exchanger: state point 14). Finally, the strong refrigerant solution leaving the absorber is pumped through a heat exchanger where it is pre-heated and passes to the desorber section where a high pressure and temperature are maintained.

4. Mathematical Model

4.1. EES Model

The ammonia-water absorption cycle described above is modelled using the Engineering Equation Solver "EES" which is a simulation environment providing numerical solution of non linear algebraic and differential equations. In addition, EES provides built-in thermodynamic and transport property functions for many fluids including water, dry and moist air, most CFC and HCFC refrigerants …etc [1]. Thus, EES allows, by its extensive library of thermo-physical property functions, solving the system of non linear equations governing the absorption cycle functioning and calculating thermodynamic properties (temperature, pressure, enthalpy, entropy, mass fraction) of the binary mixture ammonia-water in all its states (liquid, vapour, sub-cooled liquid…) within the cycle components and piping.

The working fluids in the absorption cycle are binary mixtures (ammonia and water), this means that for each component two mass balances are necessary in addition to an energy balance, excepting the case of the heat exchangers which are modelled each by four mass balances and two energy balances given that they are crossed by two independent solution streams.

In every component of the absorption chiller, the amount of heat acquired or released is given by the energy balance:

$$\dot{m}_{in} h_{in} + \dot{Q}_{in} = \dot{m}_{out} h_{out} + \dot{Q}_{out} \quad (1)$$

The mass balances of the binary solution and the ammonia mass fraction in every section of the absorption chiller are given by:

$$\dot{m}_{in} = \dot{m}_{out} \quad (2); \qquad \sum \dot{m}_{in} x_{in} = \sum \dot{m}_{out} x_{out} \quad (3)$$

The mass and heat transfer in the desorber for example are modelled by the following equations:

$$\dot{m}_8 + \dot{m}_3 = \dot{m}_7 + \dot{m}_4 \quad (4); \qquad \dot{m}_8 x_8 + \dot{m}_3 x_3 = \dot{m}_7 x_7 + \dot{m}_4 x_4 \quad (5)$$

$$\dot{m}_8 h_8 + \dot{m}_3 h_3 + \dot{Q}_{Des} = \dot{m}_7 h_7 + \dot{m}_4 h_4 \quad (6); \qquad \dot{Q}_{Des} = \dot{m}_{21}(h_{21} - h_{22}) \quad (7)$$

Finally, the coefficient of performance (COP) is determined from the ratio of the cooling capacity of the chiller $\dot{Q}_{Evap}$ and the amount of heat consumed by the desorber $\dot{Q}_{Des}$, given that the pump work is negligible compared with the heat transfer rates associated with the other components.

$$COP = \frac{\dot{Q}_{Evap}}{\dot{Q}_{Des}} \quad (8)$$

The variance of the absorption cycle or the number of independent parameters defining completely the operating state is 14. The fundamental data adopted for this study were collected from the experimental study realized by S.A.Klein [11]; the following temperature and efficiency values are considered:

- Evaporator outlet temperature: T_{13}=11°C,
- Condenser outlet temperature: T_{10}=47,85°C,
- Absorber outlet temperature: T_1=48,85°C,
- Efficiency of the solution heat exchanger: Eff_1=0,5,
- Efficiency of the liquid-vapour heat exchanger: Eff_2=0,5,
- Pump efficiency: η=0,5.

However, the modelling of the absorption cycle requires additional assumptions about the thermodynamic state at various points within the cycle. The explanation of these assumptions is detailed below.

In the case of ideal conditions, the vapour leaving the evaporator would be of an optimum quality: q[13]=1. In our simulation, we assumed that q[13]=0,94. In addition, the desorption leads to separating the binary mixture into two flows:

- the vapour leaves the desorber at state point 7 (saturated vapour): q[7]=1,
- the weak solution leaving the desorber at state point 4 is a saturated liquid water: q[4]=0.

The rectifier condenses the last existing tracks of steam in the vapour leaving the desorber, as a result, at state point 9 the refrigerant is a pure saturated vapour : q[9]=1. An ideal rectification would lead to a fraction of ammonia x_9=1, we assumed in our simulation that x_9=0,98. Finally, the solution leaving the absorber is a rich solution, its thermodynamic state is a saturated liquid q[1]=0.

The data used in the simulations as well as thermodynamic assumptions are used first of all to determine the two pressure levels:

- the high pressure : pressure within the condenser P_C calculated thanks to the characteristics of the solution at state point 10: $P_{10} = P_{Cond}$,
- the low pressure : pressure within the evaporator P_{Evap} estimated in the same way from the characteristics at state point 13: $P_{13} = P_{Evap}$.

The final model wrote on EES is composed of all the operating conditions, the thermodynamic assumptions as well as the system of equations. This system of non linear equations governing the absorption cycle is constituted by the mass and energy balances, the additional relations characterising the heat transfer within heat-exchangers and the properties of the mixture at different state points of the cycle.

4.2. TRNSYS Model

The solar driven absorption system considered is modelled with the TRNSYS [12] simulation program. The solar driven system studied in this work is made up of an evacuated tube collector connected to a thermal storage tank, with an internal auxiliary heater, feeding an ammonia-water absorption chiller (Fig.2). The absorption system provides chilled water to the cooling coils of the building. The TRNSYS program is used to model the system during the summer season (15th May- 15th September). To simulate the solar cooling system, an accurate climatic data base (TMY2 file) of a typical meteorological year is used to get the solar radiation, the dry bulb temperature and the relative humidity of the outside air in Tunis.

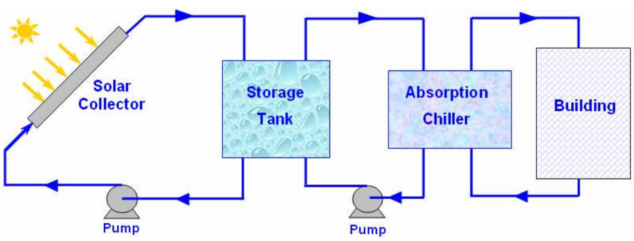

Fig. 2. The solar system

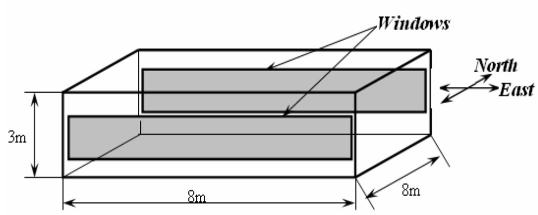

Fig. 3. Simplified Floor Description

The solar collector employed is an evacuated tube collector modelled by type 71 with technical data supplied by (SPF, 1999) [13]. The storage tank is modelled by type 60, it is a cylindrical vertical tank with an internal auxiliary heater operated when the average internal temperature of the fluid falls below a temperature threshold we fixed at 100°C. The single speed pump is modelled by type 3. The absorption chiller is modelled by its model wrote on EES described in the previous section, TRNSYS calls that model in every iteration via type 66. Finally, we studied a three-storey office building, every storey is 8x8x3m with north and south oriented windows ($12m^2$), as shown in Fig.3. The building is modelled by type 56 that calls in every iteration of the simulation a building file (extension: *.bld) containing the detailed description of the building.

5. Simulation Results

We used the EES absorption chiller model described in section 4.1 to carry out a simulation considering the case of heating the desorber by a circulation of a hot water-glycol solution characterized by a temperature T_h=100°C and a mass flow rate: $\dot{m}_h$=0,2kg/s. The temperature of the water-glycol solution circulating within the evaporator was set at 13°C [3,4,8,10]. The resulting coefficient of performance calculated by equation (8) is: *COP=0,442*. The absorption chiller consumes $\dot{Q}_{Des} = 23{,}86\ kW$ and produces a cooling capacity $\dot{Q}_{Evap} = 10{,}55\ kW$. In addition, the model calculates all the thermodynamic properties of the binary solution (ammonia-water) at all the absorption cycle points and at its different states (liquid, vapour, saturated, subcooled or superheated) [9]. The most important procedures provided by EES library we used in our model are:

- the *NH_3H_2O* procedure that let the model calculate the following thermodynamic properties: temperature (K), pressure (bar), ammonia mass fraction, enthalpy (kJ/kg), entropy (kJ/kg.K), internal energy (kJ/kg), specific volume (m^3/kg) and vapour quality.
- the *CPEG* and *DENSITYEG* functions that help us identifying the thermodynamic characteristics of the hot water-glycol solution circulating in the desorber.

A series of simulations were then carried out to study the effect of the desorber inlet hot solution characteristics variations on the performance of the whole system while holding all other inputs constant. In addition, we considered a test on the value of ammonia mass fraction of the weak solution as a simulation stop criterion. Once this value is null, increasing the temperature of the hot solution becomes useless.

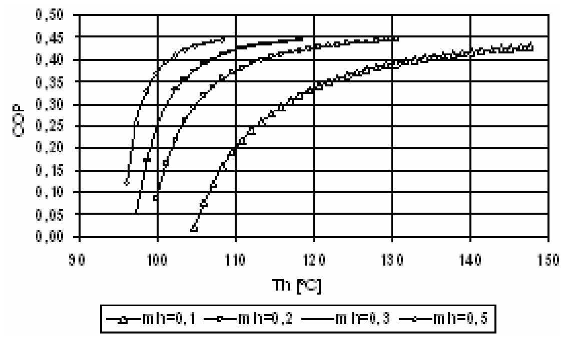

Fig.4. Effect of mass flow rate on COP

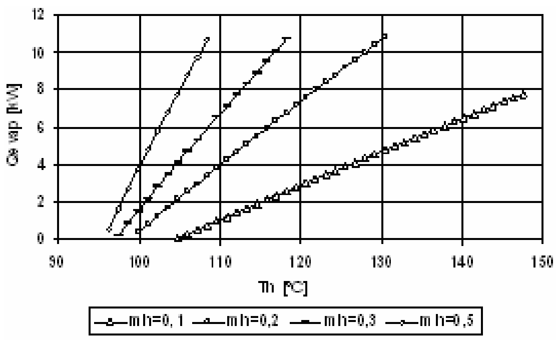

Fig.5. Effect of mass flow rate on cooling capacity

The effect of varying the mass flow rate of the hot solution ($\dot{m}_h$) from 0,1 to 0,5kg/s and the temperature (T_h) from 90 to 200°C on COP and cooling capacity is shown in Fig. 4 and 5. This temperature range spans the practical range for a single effect ammonia water chiller. The capacity plot is almost linear for all flow rate values, as the temperature increases, the cooling capacity increases too, that means that in order to increase the cooling effect of the machine we have to increase the temperature of the hot solution until an extreme value beyond which, heating becomes useless.

The COP plot shows that as temperature increases, COP increases too and exhibits constancy from a certain value of T_h, a value that depends on the considered flow rate $\dot{m}_h$. This can be explained by a compensation of the simultaneous increases of cooling capacity $\dot{Q}_{Evap}$ and the amount of heat supplied to the desorber $\dot{Q}_{Des}$. Besides, the simulations span the temperature range detailed above (from 90 to 200°C), but the curves are represented in a more restricted temperature interval (between 96 and 108°C for $\dot{m}_h$ =0,5 kg/s and between 97 and 118°C for $\dot{m}_h$ =0,3 kg/s). Except this interval, the hot solution temperature is either lower than the temperature threshold, it is thus unable to satisfy good operating conditions, or it is superior to the maximal temperature and the heat supplied to the desorber is a wasted heat since it is no more used to ensure the separation/desorption of the binary mixture (ammonia-water).

In the second part of our numerical study, we introduced the absorption chiller EES model in a system simulation using the TRNSYS simulation program. We are interested to carry out a system optimisation in order to select the right equipment of a solar absorption air conditioning system covering the load of the office building described in section 4.2. We selected an absorption chiller fed by a hot solution at T_h=130°C ($\dot{m}_h$ =0,2kg/s), such machine produces a cooling capacity $\dot{Q}_{Evap} = 10{,}796\ kW$, as a result *COP=0,449.* The first series of simulations let us calculate the building internal temperature and load fluctuations during summer season from the 15^{th} May (3241 h) until the 15^{th} September (5741h) of the building temperature. The temperature exceeds 40°C during the heat waves and the load reaches 17,42 kW. A preliminary estimation of the dimensions of the absorption system that covers the building load during 95% of the simulation period led us to choose a $16m^3$ storage tank and a hot solution mass flow rate ($\dot{m}_{coll}$ =20kg/h.m^2) circulating within the collector piping. A series of simulations were then carried out to optimise the various factors affecting the performance of the absorption system while the collector area is varying from 40 to $400m^2$.

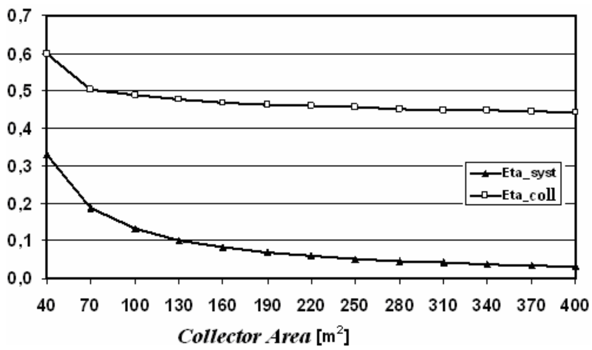

Fig.6. Effect of collector area on system and collector efficiencies

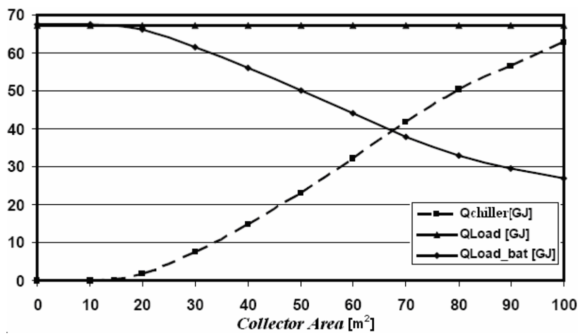

Fig.7. Effect of collector area on system energy flows

The collector and the system overall efficiencies are expressed by equations (9) and (10):

$$Eta\_coll = \frac{Q_{coll}}{I_{tot}} \quad (9); \qquad Eta\_syst = \frac{Q_{chiller}}{I_{tot} + Q_{Aux}} \quad (10)$$

Where $Q_{chiller}$ is the amount of energy produced by the absorption chiller, Q_{Load} is the load of the building (constant value) and Q_{Load_bui} is the load of the building when the absorption chiller is operated. Q_{Coll} is the amount of energy produced by the solar collector, I_{tot} is the total incident solar energy on collector surface and Q_{Aux} is the amount of heat produced by the auxiliary heater.

Fig.7 shows that as soon as the absorption chiller is operated, the building load decreases, this can be explained by energy storing: the cooling load on day j depends on the amount of thermal energy injected in the building during the previous day (j-1).

As shown in Fig.6 and 7, the collector and the overall system efficiencies decrease while collector area is increasing. In addition, Fig.7 shows that the absorption chiller covers the building load for collector areas higher than A_{coll}≈65m^2. The speculation of the different system parameters led us opt for a solar collector area A_{coll}=70m^2, the resulting absorption cooling system covers the building load with a solar fraction of 64% and the overall system efficiency is 19%. The solar fraction is the ratio of the amount of energy produced by the solar collector and the energy necessary to the absorption chiller to cover the building load.

6. Conclusion

We presented in this work a model we wrote using the Engineering Equation Solver to simulate the functioning of an ammonia-water absorption chiller, we analyzed the effect of the variation of the hot solution temperature and flow rate on chiller's performance, and we concluded that important temperatures are necessary for the optimal functioning of the absorption chiller. In addition, we noticed that even if it requires a higher temperature threshold, circulating the hot solution at a low flow rate is more beneficial since it ensures a continuous and uniform functioning of the machine in a wider range of temperatures. The model also calculates the thermodynamic properties at the different state points of the absorption system as well as the amounts of energy consumed and released by the different system components.

In the second part of this paper, we dealt with a series of simulations to determine the load of an office building during summer season. Then, we introduced the EES model in TRNSYS to simulate the functioning of an absorption chiller in a solar cooling system made up by the ammonia-water chiller, an evacuated tube collector and a storage tank. System optimisation simulations were then carried out and the selected system consists of a 70m^2 evacuated tube collector coupled to a 16m^3 storage tank and an ammonia-water absorption chiller producing 43,42 GJ. The overall system efficiency is 19%, the absorption cooling system covers the building load with a solar fraction of 64%. However, an additional economic analysis is necessary to decide of the feasibility of the selected system.

Nomenclature

COP	Coefficient of performance	T_h	Temperature of the hot solution feeding the desorber.
D	Expansion valve	*x[i]*	Ammonia mass fraction in liquid at state point i
Ech-1	Liquid-vapour heat exchanger	$\dot{W}_P$	Pump work
Ech-2	Solution heat exchanger		
$\dot{m}_i$	Solution flow rate at state point i		
$\dot{m}_h$	Flow rate of the hot solution feeding the desorber		
P_i	Pressure at state point i		
$\dot{Q}$	Amount of heat exchanged within the system component		
q[i]	Vapour quality		
T_i	Temperature at state point i		

Subscripts

Abs	Absorber
Cond	Condenser
Des	Desorber
Evap	Evaporator

References

[1]: F-Chart Software, EES 32: Engineering Equation Solver user's manual and program documentation, 1992-2001.

[2]: GA. Florides, Kalogirou SA, Tassou SA, Wrobel LC. "Modelling and simulation of an absorption solar cooling system for Cyprus". Solar Energy 2002;72(1):43–51.

[3]: H-M. Henning (Ed) (2004). "Solar assisted Air conditioning in Buildings-A handbook for planners". International Energy Agency (IEA), Springer Wien New York.

[4]: H.T. Chua, H.K. Toh, K.C. Ng (2002). "Thermodynamic modelling of an ammonia-water absorption chiller", International Journal of refrigeration, Vol.25, PP896-906.

[5]: K. Herold, Radermakher R., Klein S.A. (1995). "Absorption chillers and heat pumps", CRC Press.

[6]: K. Herold, Punwani D.V., Ryan W.A., Kalensky D.C. (2005). "Natural gas-fired cooling technologies and economics", Gas Technology Institute report (GTI-USA).

[7]: K. Sumathy Solar absorption cooling with low grade heat source- A strategy of development in South China, Solar Energy, Vol.72, No.2, PP155-165, 2002.

[8]: M. Medrano, M. Bourouis, A. Coronas (2001). "Double-lift absorption refrigeration cycles driven by low-temperature heat sources using organic fluid mixture as working pairs", Applied Energy, Vol . 68, PP173-185.

[9]: M. Soussi (2007) "Etude du potentiel d'une installation de climatisation par machine à absorption solaire" , Master, National School of Engineers of Monastir / National School of Engineers of Tunis, 90p.

[10]: N.Ben Ezzine, M. Barhoumi, Kh. Mejbri, S. Chemkhi, A. Bellagi (2004). "Solar cooling with the absorption principle: First and second law analysis of an ammonia-water double-generator absorption chiller", Desalination, Vol . 168, PP137-144.

[11]: S.A. Klein, (1982) "A model of the steady state performance of an absorption heat pump".

[12]: S.A. Klein and al (2000) a transient system simulation program TRNSYS Manual, University of Wisconsin, Madison, USA.

[13]: SPF (1999) LTS collector catalogue. Institut fur Solartechnik SPF, Rapperswil, Switzerland.

[14]: Z.F. Li, K. Sumathy, "Experimental studies on a solar powered air conditioning system with partitioned hot water storage tank, Solar Energy, Vol.71, No.5, PP285-297,2001.