Modeling and Global Sensitivity Analysis of a Solar Powered Air Cooling System Using Liquid Desiccant

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Abstract: Solar energy technology is an option for energy saving in building air conditioning. A theoretical investigation of an open cycle solar air cooling system using aqueous Lithium Chloride solution as liquid desiccant is presented in this paper. The purpose of this work is to analyze the influences of both internal loads and external forcing, on the studied system by developing a computational code related to its mathematical model. The simulation results justify the choice of the system design. Indeed, it was highlighted that the higher is the outdoor temperature; the better is the coefficient of performance (COP) of the system. Furthermore, a global sensitivity analysis of the system model, achieved using Fourier Amplitude Sensitivity Test method, allowed us to identify the most influential factors that were ranked in a decreasing order of their influence degree on the system COP. Hence, key factors to be controlled for improving the system overall performance are specified.

Keywords: Solar air conditioner, open cycle, lithium chloride, simulation, global sensitivity analysis, FAST method, complex system.

1. INTRODUCTION

Air conditioning is necessary to achieve standards of comfort in buildings. As solar energy availability coincides with the period of cooling load in buildings; powering cooling systems with this type of renewable energy is an option to reduce consumption of electricity that drives most of the traditional vapor compression air conditioning systems.

Besides, among open cycle air cooling processes, the utilization of liquid desiccants is suited for solar heating as their regeneration occurs at relatively low temperature. Apart from their operational flexibility, liquid desiccants are also widely used because of their capability of absorbing pollutants and bacteria [1].

Several experimental as well as theoretical works have been conducted in the field of air conditioning using liquid desiccant since this process was first proposed by Löf [2].

On the experimental domain, we can particularly mention few works among others. For example, Kessling *et al*. [3] conducted experiments for absorbers using LiCl and were the first to point out the benefit of energy storage thanks to the use of liquid desiccants. Jaina *et al*. [4], Pietruschka *et al*. [5], and Alizadeh *et al*. [6] proposed experimentally validated models of liquid desiccant systems while utilizing different kinds of liquid desiccants (LiCl, LiBr, and $CaCl_2$).

As cooling process using liquid desiccant combines dehumidification-humidification processes of the airstream with absorption-desorption processes related to the desiccant solution, we can categorize some experimental works on the basis of the technique used for bringing the desiccant solution into contact with the process air. The contact surface can be a wetted wall/falling film absorber [4, 7, 8], a spray chamber [9, 10], or a packed tower [11-13]. In addition, Oliviera *et al*. [14] also proposed another contact technique that consists of fiber needle impeller fans functioning as both evaporator and absorber.

Besides, simulating the model of a given liquid desiccant system is important for assessing its potential quantitatively and analyzing the cycle configurations and performance under varying working conditions and parameters [12]. Indeed, modeling and simulation are valuable assets to investigate the performance of cooling system, if we mention the works by Yadav [15], Henning *et al*. [16], Jain *et al*. [4], Khan *et al*. [17, 18], Kinsara *et al.* [19], and Tu *et al*. [20].

The present work also aims at investigating the performance of a cooling system using simulation. While being patented and experimentally tested, equipments proposed by Oliviera *et al*. [14] were adopted in the configuration of our system.

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However, in our case, we have opted for solar energy to provide the required heat for regenerating the desiccant solution which is chosen to be H_2O-LiCl solution. Whereas this heat resource is not specified in previous study [14] and also an H_2O-LiBr solution was used as desiccant solution. Indeed, while LiCl being the most stable liquid desiccant, having large dehydration concentration 30-40% [21], not being a toxic material, and not posing any health hazards [22]; cooling system using H_2O-LiCl solution as sorption material is one of the new open cycle air conditioner developments close to market introduction [23]. In addition to that, the whole cooling system, including the solar heating subcomponent, is modeled and simulated but not only the key components, such as absorber and regenerator. And finally, the overall performance of the system is analyzed using global sensitivity analysis. So far, a global sensitivity analysis of models of solar driven cooling systems with liquid desiccant has not been reported in the literature. Indeed, one advantage of global sensitivity analysis is that the sensitivity estimates of individual parameters are evaluated while all the other parameters are varied. In this way, the relative variability of each parameter is taken into account, thus revealing any existing interactions [24]. Moreover, apart from all other interaction effects obtained with global approach, the main effect of every influential parameter model informs the effect that this parameter would present with local approach. That is, effect that is obtained as the sensitivity analysis is performed by varying the parameters of interest one at a time, while keeping all others fixed at their design values. Such local approach was adopted in most of all previous works such as [17-20].

The main purpose of this work is then to analyze the behavior of the cooling system in response to the internal loads and the external forcing, as well as to identify key factors that allow us to improve the overall performance of the system. To this end, we elaborate on a model of the studied air conditioning system and conduct a global sensitivity analysis of the proposed model.

2. SOLAR AIR COOLING SYSTEM DESCRIPTION

The configuration of the studied air conditioning system is depicted in Figure **1**. It is an air conditioning technique based upon an open cycle cooling known as a direct treatment of the supply air. This technique consists of dehumidifying air with a dehydrating material, decreasing its temperature through a heat exchanger (a heat pipe for the studied case) and finally cooling it according to the required condition, with an evaporative process.

The supply air enters the system through the absorber (A-2) out of which its humidity is lowered and its temperature increases as the absorption is an exothermic process [25]. Then, that air reaches the heat pipe (2-3) where its temperature decreases while

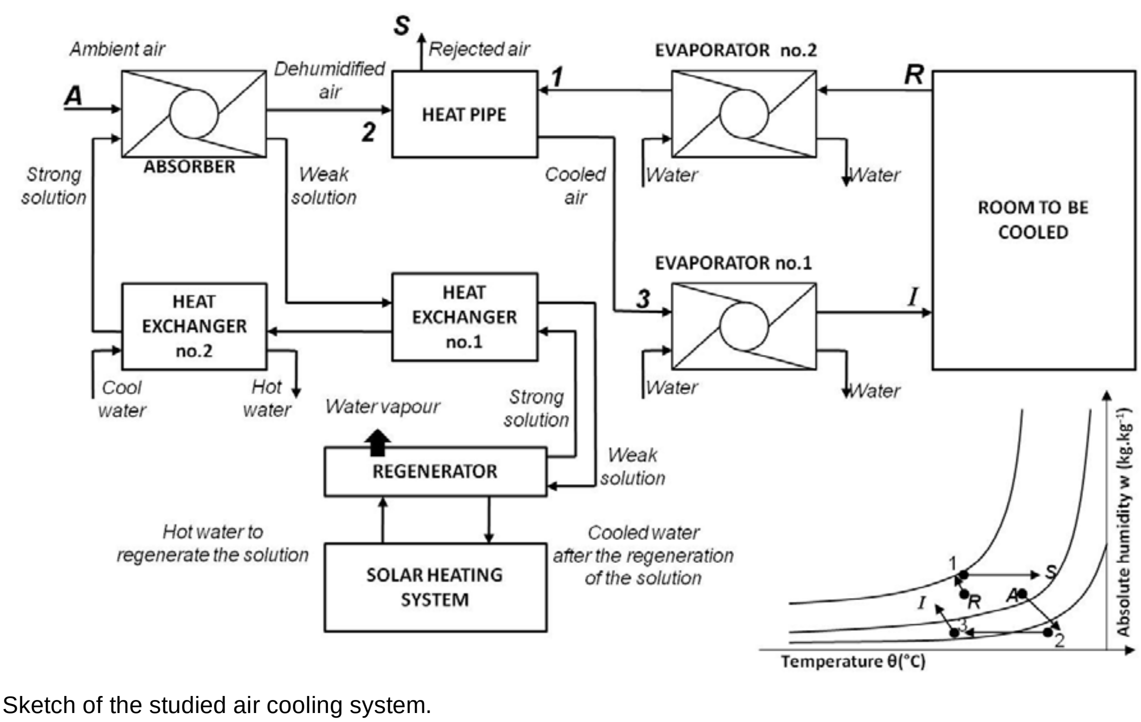

Figure 1: Sketch of the studied air cooling system.

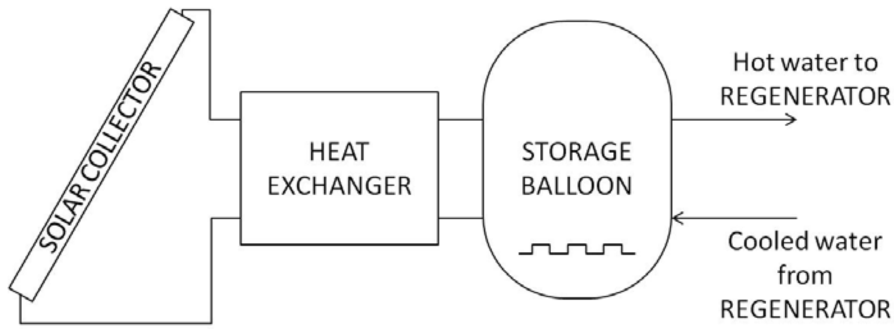

Figure 2: Sketch of the solar heating system.

transferring heat to the air extracted from the room. Afterward, the cooled air passes through the evaporator (no.1) and subsequently its temperature decreases again but its humidity increases.

Thus, the temperature and the humidity required for the comfort being obtained, the treated air is conducted into the room to be cooled. Air drained out of the room (R) is cooled through the evaporator (no.2) (R-1), gains sensible heat (heat transferred by air coming out of the absorber) inside the heat pipe, and is finally released in the atmosphere (1-S). The solar heating system is schematized in Figure **2**.

3. METHOD

3.1. Simplifying Hypotheses for the Air Conditioning System Modeling

The following assumptions were adopted to model the studied air conditioning system.

- The functioning temperature at the level of the regenerator is fixed constant in a range between 60 and 80°C as it is the available hot source temperature. The desorption ceases when there is balance between the solution and the vapor, that is, when the saturating vapor pressure on the solution free surface is equal to the pressure inside the regenerator. The temperature of the vapor in the interface is equal to that of the solution at the regenerator outlet.
- There is a perfect admixture of the fluids entering and outgoing the storage balloon. Thus, the natural stratification phenomenon that occurs in storage is not taken into account.
- At the level of the absorber, the absorption stops when there is balance between the air vapor partial pressure and the saturated vapor in the interface with the desiccant solution.
- The processes occurring in all the evaporators are adiabatic. Water evaporated from the air is at the wet bulb temperature of the air at each evaporator.

3.2. Mathematical Model

At the regenerator, mass and enthalpy balance equations of the dehydrating solution can respectively be written as:

$$\dot{m}_{\sin}\xi_{\sin} = \dot{m}_{ss}\xi_{ss} \tag{1}$$

$$Q_{gen} + \dot{m}_{\sin}h_{\sin} = \dot{m}_{ss}h_{ss} + \dot{m}_v h_v \tag{2}$$

Mass enthalpy of desorbed vapor at the regenerator is calculated by:

$$h_v = L + \Delta h_{dil} \tag{3}$$

Relative pressure in the regenerator, denoted as P_{rel} ( ), is introduced to characterize mass transfer inside it and is defined as follows [26]:

$$P_{rel} = \frac{P_{vsat}}{P_{vwat}} \tag{4}$$

Besides, Conde [26] formulated the relative pressure as:

$$P_{rel} = \pi_{25} f(\zeta, T_{red}) \tag{5}$$

$f(\zeta, T_{red})$ and π_{25} are calculated by:

$$f(\zeta, T_{red}) = \left\{2 - \left[1 + \left(\frac{\zeta}{\pi_0}\right)^{\pi_1}\right]^{\pi_2}\right\} + \left\{\left[1 + \left(\frac{\zeta}{\pi_3}\right)^{\pi_4}\right]^{\pi_5} - 1\right\} T_{red} \tag{6}$$

$$\pi_{25} = 1 - \left[1 + \left(\frac{\zeta}{\pi_6}\right)^{\pi_7}\right]^{\pi_8} - \pi_9 \exp\left(-\frac{(\zeta - 0.1)^2}{0.005}\right) \tag{7}$$

where, for H_2O-LiCl solution, $\pi_0 = 0.28$, $\pi_1 = 4.30$, $\pi_2 = 0.60$, $\pi_3 = 0.21$, $\pi_4 = 5.10$, $\pi_5 = 0.49$, $\pi_6 = 0.362$, $\pi_7 = -4.75$, $\pi_8 = -0.40$, and $\pi_9 = 0.03$.

At the absorber, the mass balance equation of the air and that of the solution are respectively given by:

$$\dot{m}_{as} w_{ain} = \dot{m}_{as} w_{aout} + \dot{m}_v \tag{8}$$

$$\dot{m}_{soin} \xi_{soin} = \dot{m}_{sout} \xi_{sout} \tag{9}$$

And enthalpy balance equation at the absorber can be written as follows:

$$Q_{abs} + \dot{m}_{as} h_{ain} + \dot{m}_{soin} h_{soin} = \dot{m}_{as} \xi_{aout} + \dot{m}_{sout} \xi_{sout} \tag{10}$$

The absorber efficiency $\varepsilon\,(\%)$ which is defined as the ratio between the real and the maximum humidity reduction is calculated by [14]:

$$\varepsilon = \frac{w_{ain} - w_{aout}}{w_{ain} - w_{aout,\min}} \tag{11}$$

The maximum humidity reduction $(w_{ain} - w_{aout,\min})$ is reached when the partial vapor pressure of the air at the absorber outlet is equal to the equilibrium pressure of saturated vapor above the dehydrating solution at the absorber inlet, as given by [14]:

$$w_{aout} = w_{aout,\min} \quad \Leftrightarrow \quad P_{vapout} = P_{solin} \tag{12}$$

Moreover, Oliviera *et al*. [14] have given an empirical equation which describes the absorber efficiency equation in terms of the ratio between the solution mass flow rate and that of the air:

$$\varepsilon = 0.1795\ \ln\left(\frac{\dot{m}_{soin}}{\dot{m}_{as}}\right) + 0.5326 \tag{13}$$

The evaporator efficiency $\eta\,(\%)$ can be computed by a relationship proposed by Stabat [27]:

$$\eta = 1 - \left(1 - \eta_{rat}\right)^r \tag{14}$$

with $r = \left(\dfrac{\dot{m}_{asrat}}{\dot{m}_{as}}\right)^{0.2}$

where $\eta_{rat}\,(\%)$ and $\dot{m}_{asrat}\ (kg.s^{-1})$ are respectively the evaporator efficiency and dry air flow rate at the nominal point of functioning.

Besides, η can also be calculated whether in terms of absolute humidity difference ratio by [27]:

$$\eta = \frac{w_{as} - w_{ae}}{w(\theta_h) - w_{ae}} \tag{15}$$

or in terms of temperature difference ratio by [28]:

$$\eta = \frac{\theta_{as} - \theta_{ae}}{\theta_h - \theta_{ae}} \tag{16}$$

Hence, once the wet bulb temperature θ_h is known, the temperature θ_{as} and the absolute humidity w_{as} of air at the evaporator outlet can be obtained. For this purpose, an iterative procedure was used in this investigation [27]:

set an initial guess for θ_h corresponding to which the saturated vapor pressure $P_{sat}\,(Pa)$ can be computed with the following relationship:

$$\log(P_{sat}) = \frac{7.625\ \theta_h}{241 + \theta_h} + 2.7877 \tag{17}$$

calculate the absolute humidity at saturation w_{sat} $(kg.kg^{-1})$ with:

$$w_{sat} = 0.622 + \frac{P_{sat}}{101325 - P_{sat}} \tag{18}$$

after computing the enthalpy at saturation $h_{sat}\ \ (J.kg^{-1})$, a new value of θ_h can be obtained by using the following equation:

$$Cp_e \theta_h = \frac{h_{sat} - h_{ae}}{w_{sat} - w_{ae}} \tag{19}$$

where Cp_e is the heat capacity of water vapour $(J.kg^{-1}.K^{-1})$.

To model the solar heating system, it was subdivided into two parts: the solar collector and the storage balloon.

The solar collector yield coefficient $\eta_{col}\ (\,)$ is defined as the ratio between the delivered heat power at the collector outlet $P_{del}\,(W)$ and the collected solar heat radiation $P_{inc}\,(W)$:

$$\eta_{col} = \frac{P_{del}}{P_{inc}} \tag{20}$$

with $$P_{del} = \dot{m}_{wat} Cp_{wat} (\theta_{down} - \theta_{up}) \tag{21}$$

where $(\theta_{down} - \theta_{up})$ is temperature difference between downstream and upstream of the collector (K).

Besides, P_{del} can be written in terms of the collector parameters, as follows [29]:

$$P_{del} = \dot{m}_{wat} Cp_{wat} \left[\tau \left(\frac{P_{inc}}{U} \right) - \theta_{up} + \theta_{amb} \right] . \left[1 - \exp\left(- \frac{UA}{\dot{m}_{wat} Cp_{wat}} \right) \right] \quad (22)$$

Online heat loss in ducts q_t $(W.m^{-2})$ was computed with [30]:

$$q_t = k_t \Delta\theta \quad (23)$$

where $\Delta\theta$ denotes the temperature difference between work fluid and outdoor ambient air (K) while the work fluid temperature being taken as arithmetic mean value of temperatures of water entering and outgoing the regenerator.

The useful solar heat Q_{sun} (J) is given by [30]:

$$Q_{sun} = \dot{m}_{wat} Cp_{wat} (\theta_{stin} - \theta_{st}) \quad (24)$$

At the storage balloon, thermal balance equation can be written as:

$$Storage\ gain = solar\ heat - consumed\ heat - heat\ loss \quad (25)$$

The consumed heat (or required heat for desorption) is calculated by [27]:

$$Q_{gen} = \dot{m}_{wat} Cp_{wat} (\theta_{stout} - \theta_{ret}) \quad (26)$$

Water enthalpy h_{wat} $(J.kg^{-1})$ inside the balloon is given by:

$$h_{wat} = \rho_{wat} V_{st} Cp_{wat} \theta_{st} \quad (27)$$

Heat loss Q_{Lb} (W) at the storage balloon is determined with the following relationship:

$$Q_{Lb} = k_b (\theta_{st} - \theta_{amb}) \quad (28)$$

Heat power related to electric supplement for eventually adjusting water temperature a_{pp} (W) is computed by:

$$a_{pp} = \dot{m}_{wat} Cp_{wat} (\theta_{req} - \theta_{stout}) \quad (29)$$

3.3. Indoor Air Enthalpy and Mass Balance Equations

3.3.1. During Temperature Decrease

When the indoor air temperature reaches a defined limit temperature value, the air conditioning system works so as to reduce the indoor air temperature; that is, to lower the indoor air temperature value to the required one at the room inlet.

Hence, enthalpy and mass balance equations of the indoor air can respectively be written as follows:

$$m_{ablow} h_{ablow} + m_{arem} h_{arem} + Q_{load} = (m_{ablow} + m_{arem}) h_{tot} \quad (30)$$

$$m_{ablow} w_{ablow} + m_{arem} w_{arem} + M_{vis} = (m_{ablow} + m_{arem}) w_{tot} \quad (31)$$

3.3.2. During Temperature Rise

When the room inlet air temperature value required for the comfort is reached, the air conditioning system stops. Due to the contribution of heat stemming from indoor sources, the indoor air temperature increases little by little to attain again the limit temperature.

Thus, indoor air enthalpy and mass balance equations can respectively be written as:

$$m_{a1} h_{a1} + Q_{load} = m_{a2} h_{a2} \quad (32)$$

$$m_{a1} w_{a1} + M_{vis} = m_{a2} w_{a2} \quad (33)$$

3.4. Cooling System Efficiency Assessment

Cooling potential of the surveyed system can be assessed by comparing air conditions at the absorber outlet (R) and at the room inlet (I). Hence, the system efficiency is estimated according to the value of its coefficient of performance:

$$COP = \frac{\dot{m}_{as} (h_A - h_I)}{Q_{gen}} \quad (34)$$

where, $(h_A - h_I)$ is mass enthalpy difference between ambient outdoor air (A) and air at the inlet of the room to be cooled (I).

3.5. Computational Procedure

Our model equations were coded into Matlab [31]. For a given set of inputs, the following algorithm can be used to get the outputs:

a) With a fixed temperature 70°C, calculate the concentration of the solution at the regenerator outlet.

b) Set an initial guess for the temperature of the concentrated solution at the absorber inlet.

c) Compute the parameters of the dehumidified air and the diluted solution at the absorber outlet using mass and enthalpy balance equations ((8) to (10)) together with humid air physical properties equations. Note that both the temperature of concentrated solution and the ambient air parameters are given in the inputs. One also assumes that the absorption process ceases when the partial pressure of the vapor is equal to the pressure of the saturated vapor at the interface.

d) Determine both the temperature of the concentrated solution and that of the diluted one at the outlet of the heat exchanger (no.1) while the temperature of concentrated solution at the regenerator outlet being fixed at 70°C.

e) Compute the temperature of the solution streaming out of the heat exchanger (no.2) and entering the absorber.

f) Go back to step c) until convergence of the temperature of the concentrated solution entering the absorber.

g) Calculate the energy required to regenerate the solution by combining equations (1) and (2) with the characteristic equations of the aqueous solution of lithium chloride (Equations (5) to (7)). It is noted that the relative pressure of the saturated vapor at equilibrium on the salt solution surface in the regenerator is set to 0.2. The desorption process (regeneration) is also assumed to cease when the solution and the vapor released are in equilibrium (i.e both pressures are equal).

h) Determine the values of the parameters related to the solar heating system used as a back-up supply and the temperature of the water in the storage balloon.

i) Calculate both the temperature and the humidity of the airstream reaching its saturation point at the evaporator (no.2) outlet.

j) Compute the temperature of the exhaust airstream through the end of the heat pipe, and that of the airstream leaving the heat pipe and flowing towards the evaporator (no.1).

k) Calculate the temperature and the humidity of the airstream which is both humidified and cooled through the evaporator (no.1) and finally brought into the room.

l) The coefficient of performance (COP) of the air cooling system is evaluated by assuming that the temperature of air extracted from the room is known.

3.6. Sensitivity Analysis Method

In order to identify key factors to be controlled for improving the system overall performance, a global sensitivity analysis was carried out using FAST method [32]. The proposed solar air cooling system model has 29 inputs as a whole. A set of well selected distinct prime numbers were chosen as frequencies of all these factors of the model in order to avoid frequency interferences during the analysis. According to Shannon criteria, the number of simulations to run should be superior to two times the maximum of frequencies (including the induced ones).Therefore, while 179 being the highest value of the aforementioned frequencies, 1432 simulations were run for this global sensitivity analysis and the system coefficient of performance (COP) was chosen as the observed output of the model.

4. RESULTS AND DISCUSSION

In order to cool the room, both the temperature and humidity of the airstream are adjusted *via* the evaporator (no.1).

Figure **3** shows the airstream temperature at the evaporator (no.1) outlet as a function of mass flow rate of the dry airstream at its inlet, at fixed temperature 25°C and for three values of the absolute humidity (0.0073, 0.0080, and 0.0087 kg.kg^{-1}) of the airstream reaching the evaporator. As can be seen from the figure, while the slopes of the three curves being almost similar, the airstream temperature at the evaporator (no.1) outlet increases along with either the dry air mass flow rate or the air absolute humidity at the inlet of this system subcomponent. This can be explained as follow: first, when the dry air mass flow rate increases, the process air has a shorter period of contact with the water wetted fiber needle surface of the impeller fan of the evaporator (no.1), and hence the water evaporation rate decreases, and this degrades the performance of the humidification process; second, when the inlet air humidity is higher, the process air can only absorb less amount of water vapor, and this also implies a decrease of water evaporation rate. Therefore, in order to make the evaporative process more efficient, the inlet airstream has to be dehumidified [33], and mounting the absorber at the evaporator (no.1) upstream is then required.

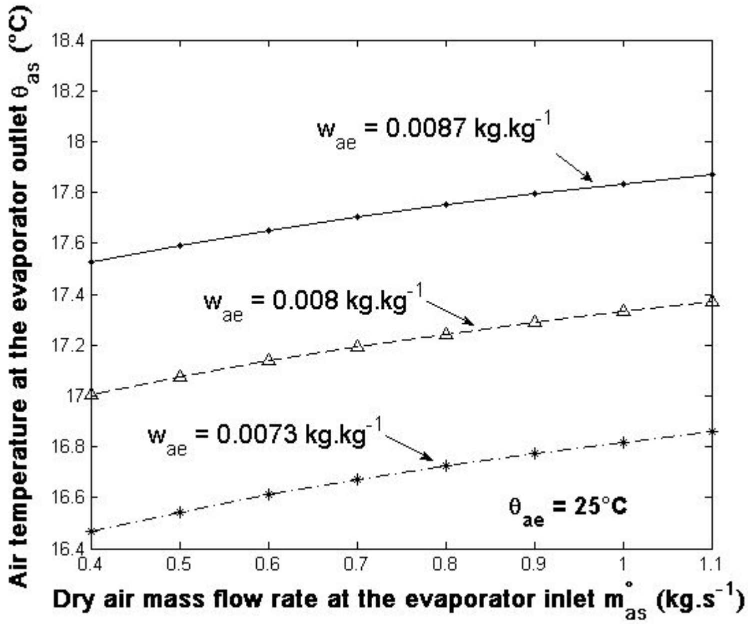

Figure 3: Influences of the mass flow rate and the absolute humidity of the air at the evaporator (no.1) inlet on the temperature of the air at its outlet.

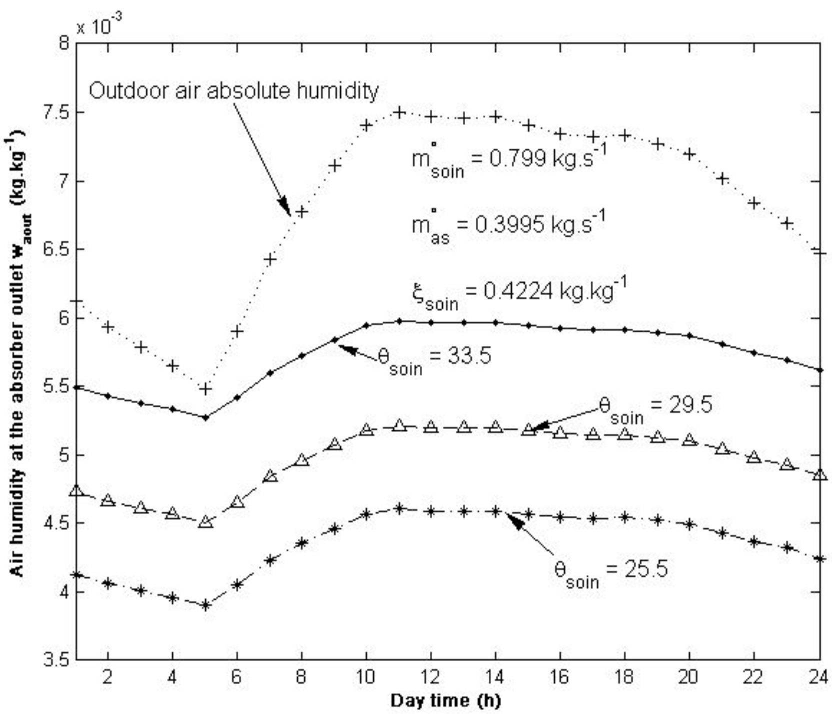

Figure 4: Influence of the temperature of the concentrated solution at the absorber inlet on the air absolute humidity at its outlet.

Figure **4** describes the variation of absolute humidity of the airstream at the absorber outlet as a function of time during a typical day, for three different values of the temperature of the concentrated solution at the inlet (25.5, 29.5, and 33.5°C). As expected, the trends of the three absolute humidity curves of the exit air are similar to that of the ambient air but with difference in magnitudes. Further, as can be observed from Figure **4** too, the air humidity at the absorber outlet increases with the desiccant solution temperature at its inlet. Thus, a high temperature of the solution at the absorber inlet induces a less efficient dehumidification process, and hence placing a heat exchanger (no.2) upstream of the absorber to reduce the concentrated solution temperature is justified.

Figure **5** illustrates, for three different concentration values of desiccant solution streaming towards the

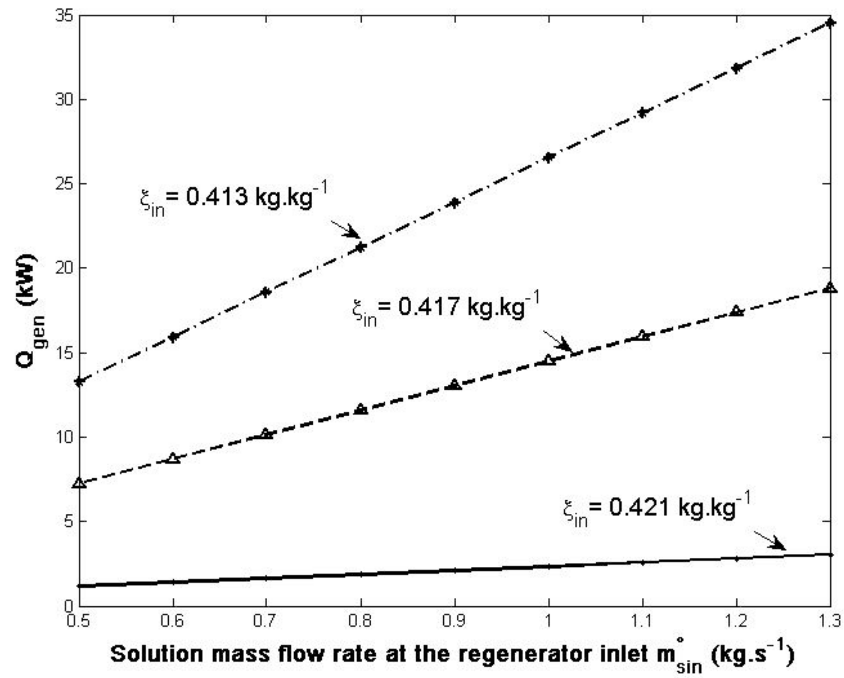

Figure 5: Influences of the mass flow rate and the concentration of the diluted solution at the regenerator inlet on the required heat power for regeneration of the desiccant solution.

regenerator inlet (0.413, 0.417, and 0.421 kg.kg^{-1}), the variation of the heat required for the regeneration of the solution as a function of the solution mass flow rate at the regenerator inlet. Clearly, the regeneration heat duty increases linearly with the solution mass flow rate. However, the slope of heat duty straight curves tends to become flat at the higher desiccant solution concentration values. This is due to the fact that the higher the concentration of the solution streaming at the regenerator inlet, the lower the required regeneration heat. Hence, pre-heating the diluted solution *via* the heat exchanger (no.1) before it reaches the regenerator is required. Besides, apart from having a strong concentrated solution at the absorber inlet, it is also preferable to reduce its mass flow rate to save energy.

Results obtained from performing a global sensitivity analysis of the studied model are shown in Figures **6**, **7** and **8**. As can be seen from Figure **6**, only 7 model factors out of 29 ones have main effects on the system COP, the dominating factors of the proposed model are ranked in a decreasing order of influence as follows: first is the relative pressure in the regenerator P_{rel}, second is the outdoor air temperature θ_{amb}, third is the outdoor air relative humidity hr_{amb}, fourth is the mass flow rate of the desiccant solution $\dot{m}_{soin}$, fifth is the efficiency of the heat exchanger (no.1) α_{avabs}, sixth is the mass flow rate of the inlet air $\dot{m}_{as}$, and seventh is the indoor air relative humidity hr_{ind}.

Furthermore, as a description of the behavior of factors having negative main effects on the model output, an increase of the relative pressure in the regenerator, the mass flow rate of the desiccant solution, the mass flow rate of the supply air, and the indoor air relative humidity, causes a decrease in the system COP and hence degrades the cooling system performance. On the contrary, for factors having positive main effects, the system COP is improved with an increase of the ambient air temperature and relative humidity as well as the efficiency of the heat exchanger (no.1). It is noted, as it was mentioned previously, that we could also arrive at the same results described above by performing the sensitivity analysis using one at a time parameter variation approach.

It follows that the less is the relative pressure in the regenerator; the better is the COP of the system. In other words, the cooling performance of the system highly depends on the quality of the desiccant solution. Besides, the temperature and relative humidity of outdoor air at the system inlet also play significant roles in the cooling performance. Taking into account the positive effects of both of those factors on the system performance, the studied cooling system is particularly adapted to tropical areas where the weather is hot and humid.

Moreover, the cooling performance of the system is improved as long as the values of both the airstream

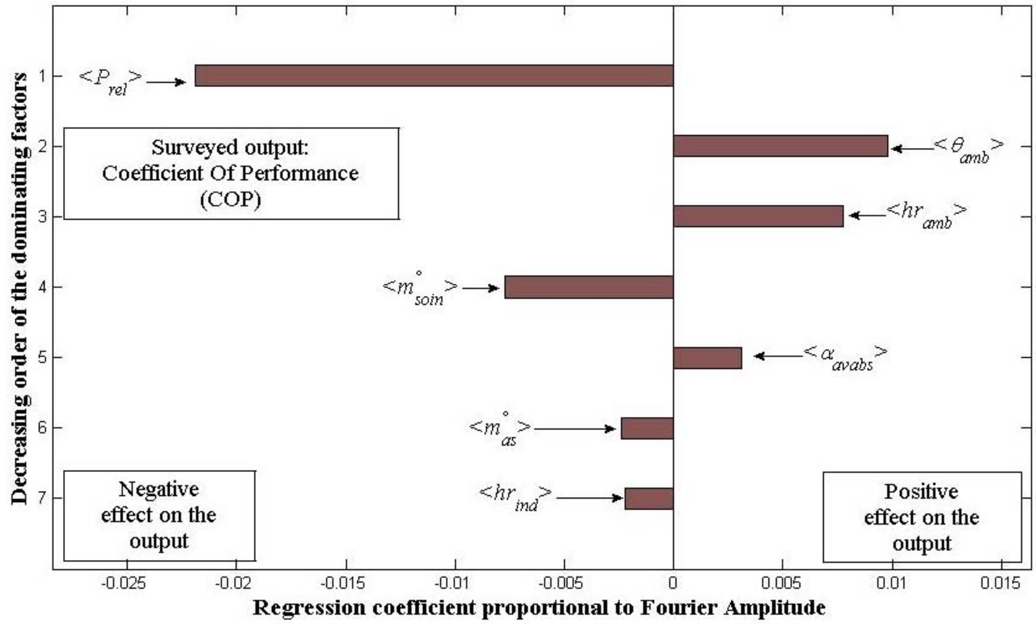

Figure 6: Ranking of the dominating factors of the proposed model according to their main effects on the system COP.

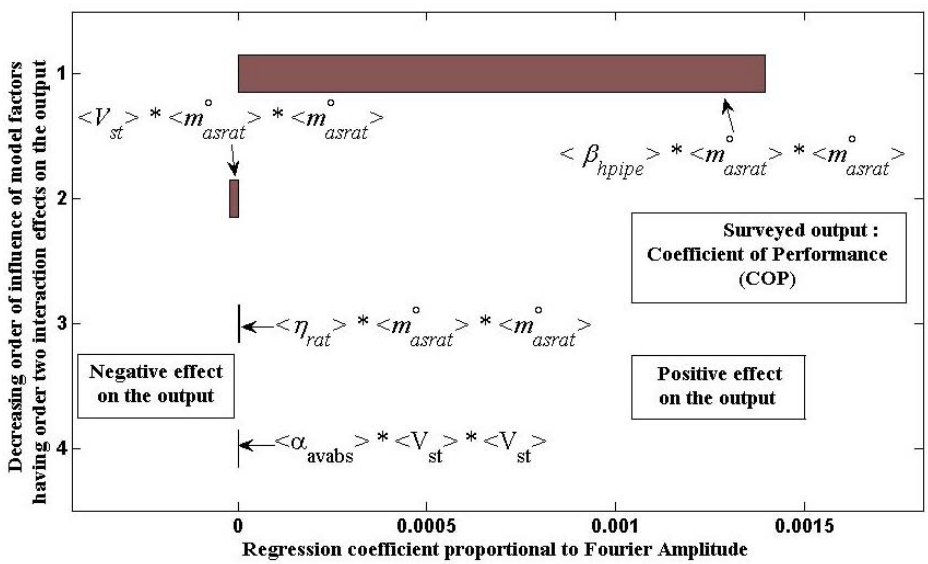

Figure 7: Ranking of factors having order two interaction effects on the system COP.

and the desiccant solution mass flow rates are kept low. Furthermore, the heat exchanger (no.1) should be chosen to have a very high efficiency in order to improve the system performance. Additionally, the room to be cooled should be designed according to the average number of occupants and all other eventual indoor vapor sources that may increase the indoor air relative humidity which has a negative effect on the system performance.

Figure **7** shows the advantage of performing a global sensitivity analysis of any chosen model. It can be noticed by comparing Figure **6** with Figure **7** that there are some factors of the model that do not have main effect on the system COP (Figure **6**) but nonetheless have order two interaction effects on that observed model output (Figure **7**), namely: the heat pipe efficiency β_{hpipe}, the dry air flow rate at the nominal point of functioning $\dot{m}_{asrat}$, the storage balloon volume V_{st}, and the evaporator efficiency at the nominal point of functioning η_{rat}. Hence, it would be beneficial to opt for a heat pipe with high efficiency. Optimization of the size of the storage balloon is required as its volume combined with the dry air flow rate at the nominal point of functioning has negative interaction effect on the system performance. However,

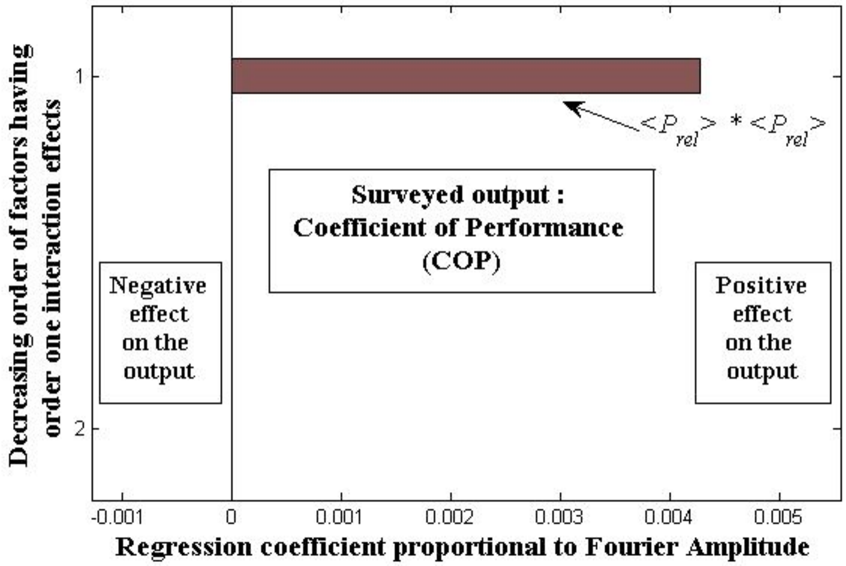

Figure 8: The only model factor having order one interaction effect on the system COP.

interaction effects of the factors listed above are insignificant compared to the main effects of the 7 factors presented in Figure **6**.

As shown in Figure **8**, the relative pressure in the regenerator is the only factor that has order one interaction effect on the observed model output. More precisely, that order one interaction effect is quadratic, that is, the effect is due to the factor interaction with itself. While the relative pressure in the regenerator has a negative main effect on the system COP, its quadratic effect on the same model output is a positive one. The fact that the main and quadratic effects have opposite signs naturally occurs in global sensitivity analysis of models but the main effect sign always prevails over the quadratic one.

Indeed, let λ and μ be respectively the absolute values of the regression coefficients of $\langle P_{rel} \rangle$ and $\langle P_{rel} \rangle \langle P_{rel} \rangle$ in the metamodel that computes the system COP. As the ratio (λ / μ) is greater than 5 (Figures **6** and **8**) and the standardized value $\langle P_{rel} \rangle$ of P_{rel} ranges between 0 and 1 when a small increase is allowed for the nominal value of P_{rel}, the sign of $\mu \langle P_{rel} \rangle \langle P_{rel} \rangle - \lambda \langle P_{rel} \rangle$ is negative. Hence, the overall effect of the most dominating factor P_{rel} of the proposed model on the system performance remains negative like its main effect.

5. CONCLUSION

A solar assisted air cooling system using aqueous Lithium Chloride solution as liquid desiccant was modeled in this work. The results of the simulations come to validate the proposed configuration of the cooling system. Indeed, a hotter concentrated solution at the absorber inlet induces a little reduction of the air humidity. Consequently, a heat exchanger should be mounted upstream of the absorber to adjust the temperature of the concentrated solution. Besides, the airstream is dehumidified, and hence cooled through the evaporator. Prior to that process of humidification through the evaporator are the dehydration *via* the absorber and cooling *via* the heat pipe.

Before it reaches the regenerator, the cold diluted solution is pre-heated *via* an exchanger so that less heat power is required for its regeneration. The results suggest that the cooling system efficiency (COP) depends on the outdoor air temperature, a better efficiency results from a higher value of that latter.

Finally, a global sensitivity analysis of the model which was achieved using a derived FAST method, allowed us to both identify and rank, in a decreasing order of their degrees of influence on the COP of the system, the most influential factors of the model. Our findings show that apart from choosing heat pipes and exchangers with high efficiency, the relative pressure in the regenerator is also among the key factors that should be controlled in order to improve the overall performance of the system. Indeed, it should be kept as low as possible in order to obtain a better performance of the system. In the same way, a low flow rate of both airstream and desiccant solution corresponds to a better efficiency of the cooling system.

APPENDIX: NOMENCLATURE

A = Collector surface area (m^2)

$CaCl_2$ = Calcium chloride

Cp = Heat capacity $(J.kg^{-1}.K^{-1})$

COP = Coefficient of performance

FAST = Fourier Amplitude Sensitivity Test

h = Mass enthalpy $(J.kg^{-1})$

H_2O = Water

hr_{ind} = Indoor air relative humidity $(\ )$

LiBr = Lithium Bromide

LiCl = Lithium Chloride

k_t = Linear transmission coefficient of ducts $(W.m^{-2}.K^{-1})$

k_b = Heat transmission coefficient of the balloon $(W.K^{-1})$

L = Vaporization enthalpy $(J.kg^{-1})$

M_{vis} = Mass of vapor stemming from indoor sources (kg)

m = Mass (kg)

$\dot{m}$ = Mass flow rate $(kg.s^{-1})$

P_{vapout} = Partial vapor pressure at the absorber outlet (Pa)

P_{vwat} = Saturated vapor pressure at the free surface of pure water (Pa)

P_{vsat} = Saturated vapor pressure at the free surface of the liquid (Pa)

P_{solin} = Equilibrium pressure of saturated vapor over the solution at the absorber inlet (Pa)

Q_{gen} = Required heat for desorption (W)

Q_{abs} = Rejected heat during the absorption process (W)

Q_{load} = Internal loads (J)

T_{red} = Reduced temperature (with water critical temperature) of the solution ( )

U = Heat loss flux $(W.m^{-2}.K^{-1})$

V_{st} = Storage balloon volume (m^3)

w = Air absolute humidity $(kg.kg^{-1})$

$w_{aout,\min}$ = Minimum absolute air humidity at the absorber outlet $(kg.kg^{-1})$

Greek Symbols

α_{avabs} = Efficiency of the heat exchanger (no.1) $(\ )$

β_{hpipe} = Heat pipe efficiency $(\ )$

Δh_{dil} = Differential enthalpy of dilution of the solution $(J.kg^{-1})$.

ε = Absorber efficiency $(\%)$

ζ = Mass fraction of solute in the desiccant solution ( )

η_{rat} = Evaporator efficiency $(\%)$

θ = Temperature (K)

θ_h = Wet bulb temperature (K)

τ = Shortwave radiation transmittance of the collector glass ( )

ξ = Concentration $(kg.kg^{-1})$

ρ_{wat} = Water density $(kg.m^{-3})$

Subscript

$a1$ = Air before contribution of sensible heat

$a2$ = Air after contribution of latent heat

$ablow$ = Air blown in the room

ae = Humid air at evaporator (no.1) inlet

amb = Outdoor air

ain = Air at the absorber inlet

$aout$ = Air at the absorber outlet

$arem$ = Remaining indoor air mass after air renewal

as = Dry air

ret = Cooled water streaming back to the storage balloon from the regenerator

req = Required for the desiccant solution regeneration

sin = Diluted solution at the regenerator inlet

soin = Concentrated solution at the absorber inlet

sout = Diluted solution at the absorber outlet

ss = Concentrated solution at the regenerator outlet

st = Water inside the storage balloon

stin = Water streaming at the storage balloon inlet

stout = Water streaming at the storage balloon outlet

v = Desorbed vapor

wat = Water

tot = Total indoor air

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