PARAMETRIC ANALYSIS OF A SOLAR DESSICANT COOLING SYSTEM USING THE SIMSPARK ENVIRONMENT

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ABSTRACT

A desiccant cooling unit powered by 14.8 m^2 of solar liquid collectors is implemented in a training room in Chambery in Eastern France. The system uses a Lithium Chloride sorption wheel and is optimized to work without auxiliary heat regeneration source. Several parameters are studied in order to decrease primary energy consumption and increase system performance. The models are implemented in SimSPARK a simulation environment linked with energy+ and able to solve complex problems. Simulations show that humidifiers efficiencies and rotating heat exchanger efficiency have major influences on the system performance. Besides it was shown that airflow rate during inoccupation period is a key component in the operation of the system. Our results suggest that an airflow rate of 2.8 AC/H during inoccupation decreases required regeneration hours by about 40%, decreases room required cooling factor about 62% while electricity consumption increases only about 8%.

INTRODUCTION

Desiccant evaporative cooling is an alternative technology to traditional air conditioning systems. Being heat driven, it can be coupled to solar collectors to produce a cooling system with low environmental impact. This technology has been widely used in USA and Northern Europe. In USA, the number of desiccant industrials has increased from 2 in 1980 to more than 10 in 2001 with more than 5700 systems installed in commercial buildings (Stabat, 2003). Several studies in Northern Europe (Dittmar, 1997, Lindholm, 2000), demonstrated that solar energy used with desiccant systems, can reduce annual gas consumption by as much as 70%. Recently, these systems were used in Western Europe and especially in Germany where there are about 7 solar desiccant installations. In the chamber of commerce from Fribourg in Germany, an autonomous solar desiccant cooling system allows cooling two meeting rooms of 65 and 148 m^2 containing 120 persons (Climasol). With 100m^2 of solar collectors and 60 kW of cooling capacity, reductions in primary energy consumption are about 30000 kWh and in CO_2 emissions about 8800 kg/year. Currently in France, a solar desiccant system is installed in Chambery (in Eastern France) to refresh a 70m^2-training-room (see figure 1). This paper presents results of the development of a dynamic simulation model for this system. We used a simulation platform called SimSPARK (Mora et al., 2003) developed at LEPTAB (La Rochelle, France). This tool generates building models automatically and uses SPARK (Sowell and Haves, 2001) as the solver. It is a suitable tool for adding new components such as our air conditioning system in this project, since it facilitates models coupling through a simple connection language. A general algebraic and differential equation solver handles the whole problem solution process.

Figure 1: A southern view of the room in which the desiccant system is installed.

First, system operation and its control strategy are described and component models are briefly presented. Then the simulation environment SimSPARK is described. Finally simulation results are shown. Two models were used for simulations. The first model couples the air conditioning installation to the building. It was used to study the effect of humidifiers and rotating heat exchanger efficiencies on the overall system performance. The second model couples the solar installation to the previous model. It was used to study several parameters such as the effect of storage volume, collector slope and air flow rate during inoccupation period on system performance and on equipment electrical consumption.

SYSTEM OPERATION

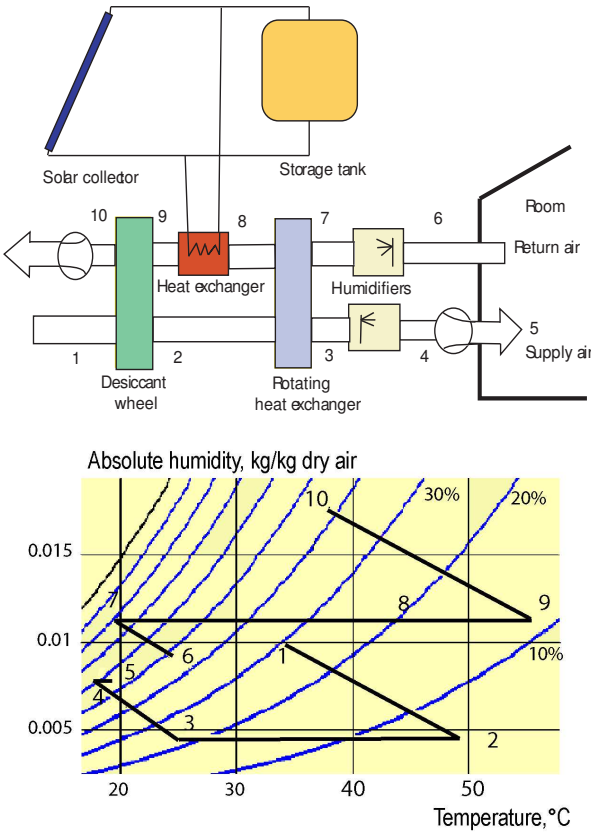

Figure 2: Schematic representation of the desiccant cooling system coupled with the solar installation and the corresponding air evolution in the psychrometric chart.

Figure 2 shows the desiccant cooling air-handling unit coupled with the solar installation. This unit comprises a desiccant wheel in tandem with a thermal wheel with evaporative coolers in both supply and return air streams before the thermal wheel. This system allows cooling and dehumidifying air without using conventional refrigerants. The desiccant wheel contains a desiccant material (Lithium Chloride) which needs to be regenerated with an external heat source (Klingenburg). This heat is taken from a solar installation consisting of a solar storage tank and solar collectors. Since the required regeneration temperatures are low (40°C to 70°C) liquid flat plate collectors are used. Depending on outside air conditions and on building loads, air installation has four operating modes (Hening et al., 2001):

- Ventilation mode in which only the supply fan (4-5) is running.

- Direct humidification mode in which supply air is directly humidified (3-4).

- Indirect humidification mode where supply air is sensibly cooled through a rotating heat exchanger (2-3). On the other side of the exchanger, return air is cooled by humidification (6-7).

- Desiccant mode in which outdoor air is dehumidified through the desiccant wheel. During the absorption of the vapour water in the wheel ,air is dehumidified almost adiabatically (1-2). After that the temperature is lowered in the rotating heat exchanger (2-3), and in the direct humidifier (3-4). The return air is cooled in an evaporative cooler (6-7) and is used to cool down the process air in the heat exchanger (7-8). Then it is heated from the solar installation (8-9) to regenerate the desiccant wheel (9-10). The states of the process and exhaust air are represented on the psychrometric chart.

CONTROL STRATEGY

The control strategy depends on room occupation (Stabat, 2003). During occupation period, the system works either in desiccant mode, ventilation mode or indirect mode as shown in the figure 3. Air flow rate is supposed constant and equal to 0.6 kg/s. When the room is occupied at 9 AM the system works either in ventilation mode or in indirect mode depending upon temperature difference ΔT1 between outdoor air and outlet of the return humidifier. If this temperature difference exceeds 1°C, the system runs in the indirect mode. In the opposite case, the indirect humidification is inefficient and the system works in ventilation mode. If room temperature exceeds 26°C and storage temperature is higher than 55°C, the system switches to the desiccant mode till the temperature reaches 23°C. When storage temperature becomes less than 50°C, the installation runs either in indirect humidification or ventilation mode depending on ΔT1 though they are not efficient (for lower temperatures, air regeneration temperature is less than 40°C, the minimum regeneration temperature required for Lithium Chloride). System optimisation will be based on minimizing the occurrence of this case.

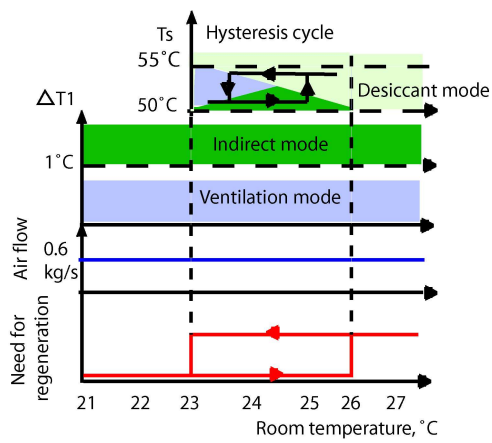

Figure 3: Schematic representation of the system control strategy during occupation period, ΔT_1 is the temperature difference between outdoor air and outlet of the return humidifier, T_s is the storage tank temperature.

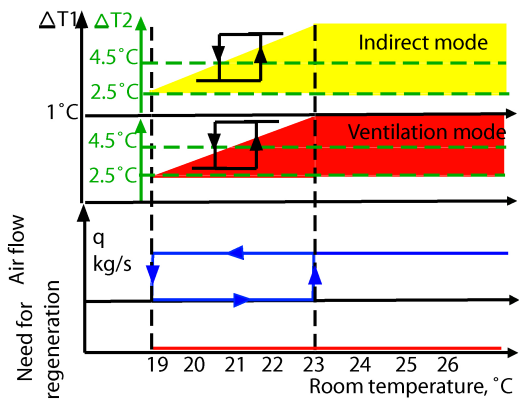

Figure 4 Schematic representation of the system control strategy during inoccupation period, ΔT_2 is the temperature difference between room temperature and supply air temperature, q is the air flow rate in kg/s.

In inoccupation period the system runs either in ventilation or in indirect humidification mode depending on the value of ΔT1 and only if room temperature exceeds 23°C and ΔT2 exceeds 4.5°C (ΔT2 is the temperature difference between supply air and room air temperatures to ensure that ventilation or indirect humidification are efficient). Details are given in figure 4. All hysteresis cycles are used to avoid continuous switching from one mode to another. The air flow rate during inoccupation period is taken to be q kg/s. Its effect is investigated using simulations (for q=0 to 0.6 kg/s).

SIMULATION ENVIRONMENT AND COMPONENT MODELLING

To evaluate system performance and its cooling potential, it should be coupled with a building for different climatic conditions. This evaluation requires a powerful modelling environment that allows user to test new models and run complex simulations with a short running time. That is why models are developed within the Simulation Problem Analysis and Research Kernel (SPARK), an equation based modelling environment developed by the Simulation Research Group at Lawrence Berkeley National Laboratory (Sowell and Haves, 2001). Description of a problem for SPARK solution begins by breaking it down in an object-oriented way. This means thinking about the problem in terms of its components, where each component is represented by a SPARK object. A model is then developed for each component. Since there may be several components of the same kind, SPARK object models, equations or group of equations, are defined in a generic manner called classes. Classes serve as templates to create any number of objects required to formulate the whole problem. The problem model is then completed by linking objects together. Using graph-theoretic techniques, SPARK reduces the size of the equation system and uses a Newton-Raphson iterative method to solve the reduced system and, after convergence, solves for the remaining unknowns.

For large and complex problems, building a SPARK simulation can be long and error prone. Therefore a tool called SimSPARK was developed at LEPTAB (La Rochelle, France), to automatically generate simulations for building applications (Mora et al., 2003) and visualize results.

In our simulation, the building model was generated by SimSPARK. It is based on a heat balance model and assumes a well-mixed zone with uniform temperature (Mora et al., 2003). Then components of the desiccant cooling system have been implemented in SPARK with the aim to be accurate and easy to use.

Modelling of desiccant wheel is based on analogy theory (Banks, 1972). Equations for coupled heat and mass transfer are reduced in two uncoupled differential equations of two independent variables called characteristic potentials which replace humidity ratio and enthalpy. In the psychrometric chart and for the lithium chloride, these potentials are represented by a constant relative humidity curve and an isenthalpic curve. This model was used and validated by Stabat (Stabat, 2003).

Concerning the rotary heat exchanger model, It is based on Ntu-effectiveness relations for heat exchangers (Kays et al., 1984, Incropera et al, 1996). (counter flow heat exchanger relations corrected for rotation). For the humidifiers, it was assumed that evaporative cooling was a constant wet bulb process. For the solar installation components, we assumed a uniform temperature for the storage tank with heat loss coefficient to the environment of 0.2% each hour (Filfli et al., 2004). Storage temperature was limited to 95°C. The model of solar collector is based on the "Hotte-Whillier-Bliss" equation for flat plate solar collectors in which additional correction terms where included to take into consideration effect of incidence angle for direct and diffuse radiation (Perers et al. 2002).

SIMULATION RESULTS

For the simulations, two models were developed:

- A model of air installation coupled to the building, in which regeneration temperature was assumed constant and equal to 50°C. This model was used to show the influence of various design conditions on the performance of the desiccant cooling system. Those conditions consist of direct humidifier efficiency, indirect humidifier efficiency and rotating heat exchanger efficiency. The simulations were run for a typical summer day and only for the air installation running in the desiccant mode.

- A model consisting of the previous model to which was also added the solar installation. This model was used to optimize solar installation components and to study the influence of operating conditions on the whole system performance. Control strategy shown in figures 3 and 4 was used. The simulations were run for summer period.

First model results

In this case, the performance of the desiccant cooling system is evaluated in terms of room temperature at 3 P.M. and system coefficient of performance (COP). The latter is defined as the heat removed from the process air stream divided by the energy input to the cycle (from humidifiers, pumps, wheels and fans). Regeneration energy is assumed free.

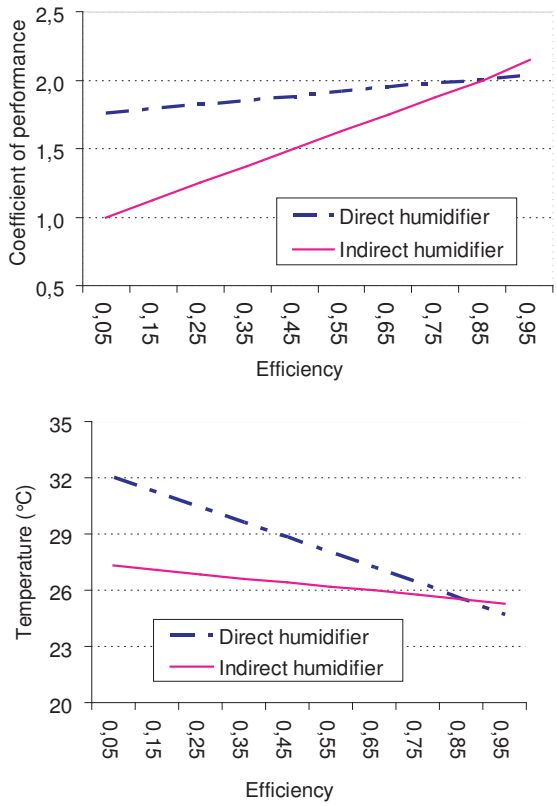

Figure 5: Evolution of system COP and room temperature at 3 PM as a function of direct and indirect humidifiers efficiency, heat exchanger's efficiency is 0.8 and regeneration temperature is 50°C. For each case the other humidifier efficiency is 0.85.

Figure 5 shows the COP and room temperature for various humidifier performances (for both the supply and return air humidifiers or respectively direct and indirect humidifiers).

It can be seen that direct humidifier efficiency has a little effect on the COP because the flow through the humidifier is a constant wet bulb process (almost isenthalpic). So air enthalpy changes a little, but its sensible heat is transformed to latent heat (air temperature decreases and its humidity increases). The role of this component is to cool supply air which is dried and heated in the desiccant wheel. It controls room comfort level.

Indirect humidification system components (indirect humidifier and rotating heat exchanger) have more influence on the COP. When return humidifier's efficiency increases from 0.05 to 0.95, the COP increases from 1 to 2.2 (figure 5). Besides, changing heat exchanger efficiency from 0.4 to 1 causes an increase in coefficient of performance from 0.45 to 3.1. In fact, as the efficiency of the indirect humidification system increases sensible heat transfer through the heat exchanger increases and supply air temperature at the heat exchanger outlet decreases.

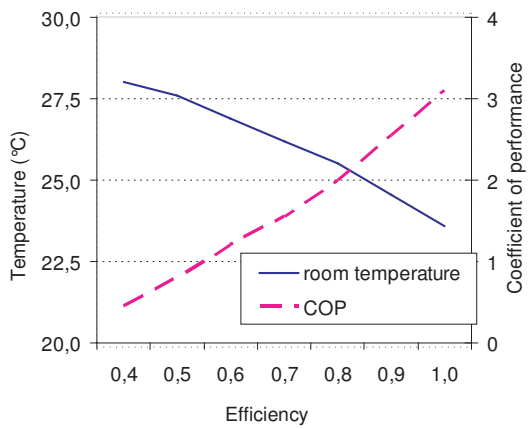

Figure 6: Evolution of system COP and room temperature at 3 PM as a function of heat exchanger's efficiency, for a regeneration temperature of 50°C and humidifiers efficiency of 0.85.

Figures 5 and 6 show that increasing the efficiency of a component of 10% leads to a decrease in room temperature of 0.8°C for the direct humidifier, 0.2°C for the indirect humidifier and 0.9°C for the rotating heat exchanger. These results confirm the results obtained by Lindholm (2000).

For the next simulations, we have considered an efficiency of 0.8 for the heat exchanger and 0.85 for both humidifiers.

Second model results

Having a collector surface of 14.8 m^2, the simulations were run in order to optimize storage volume, solar collector slope and air flow rate during

inoccupation period. They were run for a period of three months from the first of June to the end of August. The room is occupied from 9 AM till noon and from 2 PM to 6 PM. During this period internal heat loads consisted of latent and sensible heat gain from occupants (40 persons) and sensible gains from lighting (600 W).

The effects of storage volume, collector slope and inoccupation airflow rate are studied by calculating several parameters such as the cooling requirement factor defined as (Belarbi et al., 2001; Maalouf et al. 2005):

$$IB = \int_{occupation} \left(T_{room}(t) - T_{Set\ point}\right)\ \delta(t)\ dt$$

$$\delta(t) = 1 \text{ if } T_{room}(t) >= T_{Set\ point}$$

$$\delta(t) = 0 \text{ if } T_{room}(t) < T_{Set\ point}.$$

$T_{Set\ point}$ was chosen to be equal to 26°C (it was assumed that higher temperatures lead to uncomfortable situations without taking into consideration outdoor conditions and adaptative comfort).

Another key value was also investigated. It is the amount of hours where solar installation is out of work while there is a need of regeneration (Tstorage is less than 50°C). It is also equal to the amount of hours an auxiliary heat source would run to provide regeneration power (if there was an auxiliary heat source such as electrical or gas burner).

Figures 7 and 8 show the evolution of the *IB* and the auxiliary hours for inoccupation airflow rates values of 0, 0.2 and 0.6 kg/s, storage volume variation from 0.5 to 4 m^3 (details are shown for q=0.2 kg/s) and for collector slope variation from 15° to 55° with a step of 5°. Each curve corresponds to a given airflow rate and a given storage volume. For a given airflow rate, it can be seen that the optimal storage volume is about 1.5 or 2 m^3 (their curves are very close). In fact, for smaller volumes the stored energy is limited because the storage temperature approaches 95°C and for higher volumes, storage temperature needs more energy to reach the temperature of 50°C which allows regeneration.

Concerning collector slope, it can be seen that its effect is negligible for values less than 35°. For higher values the global radiation in the collector plane decreases and thus stored energy decreases which leads to an increase in the IB and in the amount of auxiliary hours. From figures 7 and 8, it can be seen that volumes of 1.5 or 2 m^3 and collector slopes less than 35°C give usually satisfying results. In the following sections we have considered values of 1.5 m^3 for the storage volume and 25° for the collector slope.

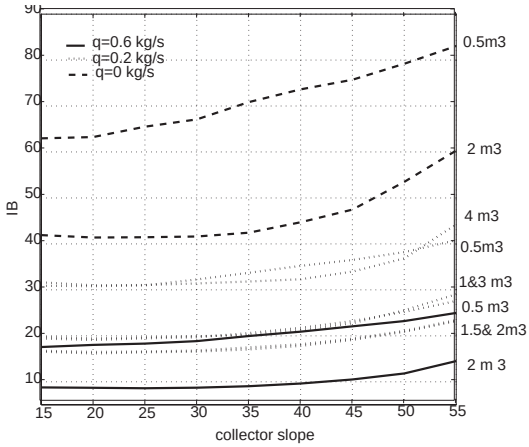

Figure 7: Variation of the cooling requirement factor with the collector slope for different storage volumes and air flow rates during inoccupation time.

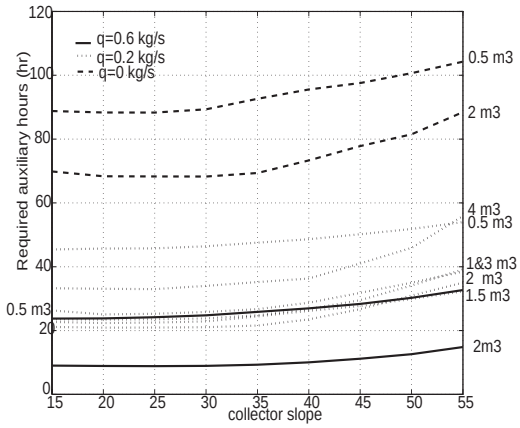

Figure 8: Variation of the required auxiliary hours (if there was an auxiliary heat source) with the collector slope for different storage volumes and air flow rates during inoccupation time.

Concerning electrical consumption, results show that it is mainly dependant on the ventilation airflow rate during inoccupation time. For a flow rate of 0.2 kg/s we have a consumption of 895 kWh and for a flow rate of 0.6 kg/s the consumption becomes 1694 kWh. Table 1 shows the effect of the ventilation air flow rate during inoccupation period on several parameters. It can be seen that as the air flow rate increases the installation runs less in desiccant mode, the amount of auxiliary hours decreases, the cooling requirement factor IB decreases and electrical energy consumption increases. For an airflow rate of 0.2 kg/s we have the higher benefits (decrease of 40% in total required desiccant hours (sum of the second and the third columns of table1) and decrease of 62% for the IB), with the less electrical consumption growth (8%).

Table 1: Effect of air flow rates during inoccupation period on several factors.

Air flow rate	IB	Required auxiliary hours	Desiccant mode running time	Total electrical consumption	Tmax	COP
(kg/s)	(°C hr)	(hr)	(hr)	(kWh)	(°C)	
Q=0	43.25	69.33	127.8	833	28	1.691
Q=0.2	16.7	22.5	98.6	895	27.7	1.788
Q=0.4	11.01	12.25	82.41	1300	27.6	1.827
Q=0.6	8.8	9.75	80.91	1694	27.5	1.862

The seasonal coefficient of performance for the desiccant mode is also shown in the table 1. This coefficient is defined as the heat removed from the process air stream divided by the energy input to the cycle (from humidifiers, pumps, wheels and fans) when the system runs in the desiccant mode and for the whole summer season. An increase in air flow rate tends to improve system performance by decreasing desiccant mode running time and increasing the amount of stored solar energy. This results in higher regeneration temperatures and higher desiccant mode cooling power. This is because an increase in regeneration temperature increases the desiccant wheel outlet process air temperature and decreases its moisture. Therefore, the process air enters the evaporative cooler at a lower wet bulb temperature, meaning that the final supply air temperature will be lower (Khalid et al., 2001, Maalouf et al., 2005).

Table 1 shows also the highest room temperature reached for more than 15 minutes during occupation period. More details are given in figure 10 where room temperature, its relative humidity and system running mode are shown for q=0, 0.2 and 0.6 kg/s during the hottest week in the year.

It can be seen that the control strategy during inoccupation period has effect on room temperature especially before noon. As air flow rate increases, room temperature decreases and installation switches to the desiccant mode later (first and fifth day). In the second day before noon, solar installation is out of work for q=0 kg/s while it keeps working for other flow rates (figure 9). End of the second day, it can be seen that for all flow rates, storage energy is consumed and storage temperature is lower than 50°C. For the third and fourth day storage temperature has the same evolution for all flow rates because desiccant mode cannot run until storage temperature reaches 55°C. It is noticed that solar energy is stored before noon and consumed after noon when outdoor conditions are hotter (figure 9).

Concerning room relative humidity, it fluctuates around 65% with a slight growth for higher air flow rates because air temperature is lower (figure 10). So effect of control strategy during inoccupation period consists to decrease installation desiccant mode running time allowing to store more solar energy in the tank and to decrease the amount of auxiliary hours where there is a need for regeneration.

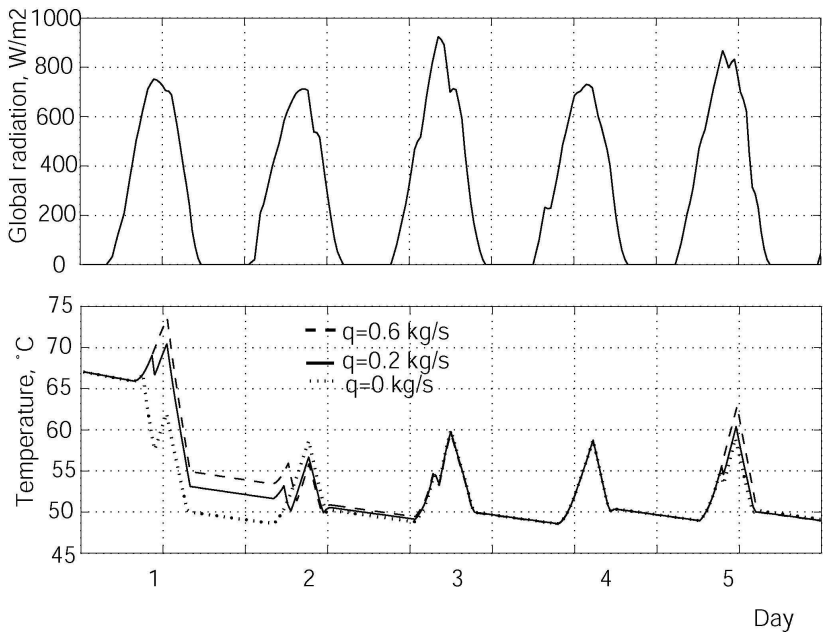

Figure 9: Evolution of storage tank temperature, for different air flow rates during inoccupation time for the hottest week with the global radiation falling on collector plane.

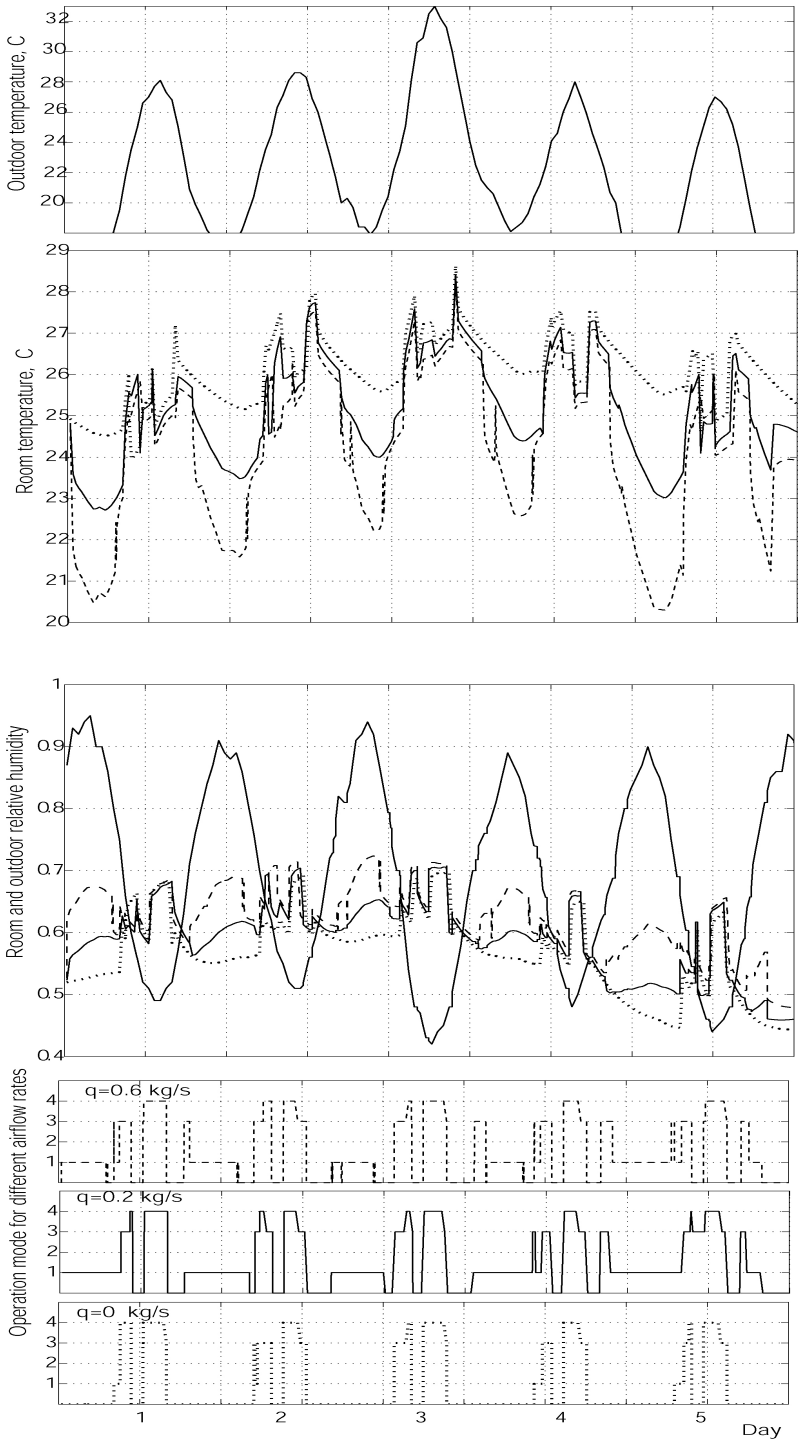

Figure 10: Evolution of outdoor and inside room temperature and relative humidity for different air flow rates during inoccupation time for the hottest week, with system running mode for each case: 1 ventilation mode, 3 indirect humidification mode and 4 desiccant mode.

CONCLUSION

In this paper, an autonomous solar desiccant cooling system was investigated. The performance of this system is greatly influenced by the efficiency of the supply air humidifier, the return air humidifier and the rotating heat exchanger. The effect of storage volume, solar collector slope and control strategy during inoccupation period was also investigated. Using a set point temperature of 26°C, storage tank volume and collector slope were chosen in order to minimize room cooling requirement factor. Then a detailed study shows the impact of air flow rate

during inoccupation period. It was shown that, during inoccupation period, even with low air changes per hour (about 2.8); there is an improvement in room indoor conditions with a slight increase of electrical energy consumption (the next step will be also investigating effect of building inertia and studying room comfort conditions).

The simulation environment used, SPARK, is a very powerful tool for such studies. The advantages of this tool are its modularity which allows building complex simulations and its strict syntax that permits having the simulations automatically generated (building model). Results will be compared with an experimental study which will take place this summer in Chambery. Then simulations will be used to evaluate the cooling potential of the system for different climatic conditions and building configurations (this project has been proposed by LEPTAB for annex 44 of IAE).

ACKNOWLEDGEMENTS

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