Optimization and study of an autonomous solar desiccant cooling system

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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 an auxiliary heat regeneration source. Several parameters are studied in order to decrease primary energy consumption and increase system performance. The models are implemented in an object-oriented simulation environment called SPARK which allows system coupling with building models. Simulations show that airflow rate during inoccupation period is a key component in the operation of the system for it can help decreasing required regeneration hours. Our results suggest that an airflow rate of 2.8 ACH during inoccupation decreases the required regeneration hours of about 40% with an increasing of equipment electrical consumption of 8%.

1. 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 the tertiary 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 are used in Western Europe and especially in Germany where there are about 7 solar desiccant installations. In the commerce room of Fribourg in Germany, an autonomous solar desiccant cooling system allows cooling two meeting rooms of 65 and 148 m^2 containing 120 persons (Climasol, 2005). 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. Actually in France, a solar desiccant system is installed in Chambery (in Eastern France) to meet the cooling demand for a training room of 70 m^2. This system will be studied in this paper.

First, system operation and its control strategy are described and then component models are presented briefly. These models are implemented in an equation-based simulation environment called SPARK (Sowell and Haves, 2001). The main advantage of SPARK is its modularity which allows the creation of very flexible tools and allows user to build complex simulations. Using simulations, several parameters are investigated such as the effect of storage volume, collector slope and air flow rate during inoccupation period on system performance and equipment electrical consumption.

2. SYSTEM OPERATION

Figure 1 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 air supply and return air streams before the thermal wheel. This system allows cooling and dehumidifying air without using conventional refrigerants. The desiccant wheel contains

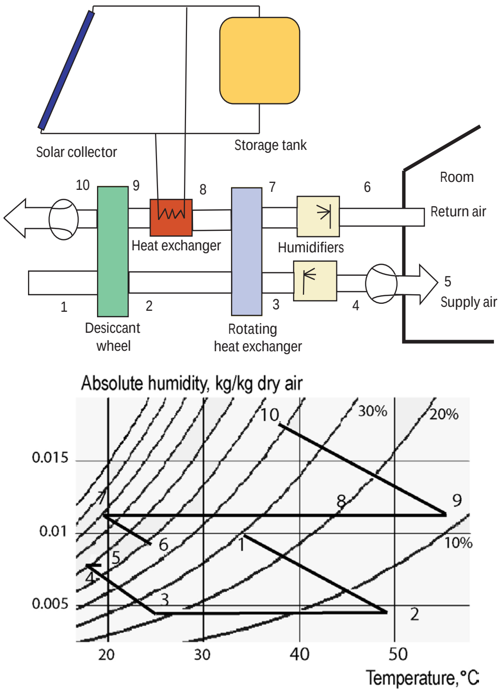

Figure 1: Schematic representation of the desiccant cooling system coupled with the solar installation and the corresponding air evolution in the psychometric chart.

a desiccant material (Lithium Chloride) which needs to be regenerated with an external heat source (Klingenburg, 2005). 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, the 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 psychometric chart.

3. CONTROL STRATEGY

The control strategy depends on room occupation (Stabat, 2003). During occupation period, the system can work with desiccant mode, ventilation mode or indirect mode as shown in Figure 2. Air flow rate is supposed constant equal to 0.6 kg/s. When the room is occupied at 9 AM the system can work either in ventilation mode or in indirect mode. This depends upon the temperature difference ΔT1 between the outdoor air and the 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 the ventilation mode. If room temperature exceeds 26°C and storage temperature is higher than 50°C (to provide the minimum required regeneration temperature of 40°C), the system switches to the desiccant mode till the temperature reaches 23°C. In case

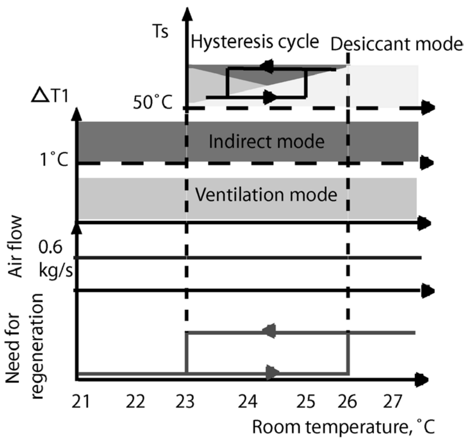

Figure 2: Schematic representation of the system control strategy during occupation period, ΔT_1 is the temperature difference between outdoor air and the outlet of the return humidifier, T_s is the storage tank temperature.

storage temperature is lower than 50°C, the installation runs on either indirect humidification or ventilation mode depending on ΔT1 though they are not efficient. System optimisation will be based on minimizing the occurrence of this case.

In the inoccupation period the system can run under the ventilation or the 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 3. All the 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).

4. COMPONENT MODELLING

To evaluate system performance and its cooling potential, it should be coupled with a building for different climatic. This evaluation requires an object oriented program. That is why the models are developed with SPARK, an equation based modelling environment. The main advantage of this environment is that the models of individual components are input/output free.

Room model is based on a heat balance model. It is modelled as a well-mixed zone with uniform temperature (Mora et al., 2004).

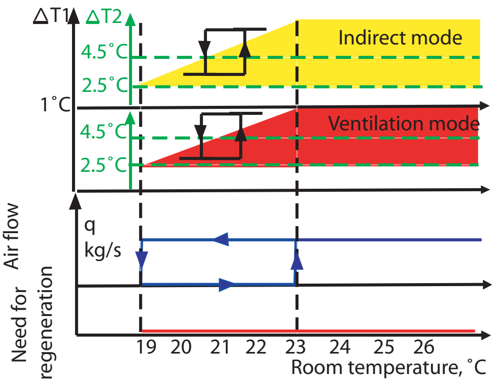

Figure 3: Schematic representation of the system control strategy during inoccupation period, ΔT_2 is the temperature difference between room temperature and supply air temperature, q is the air flow rate in kg/s.

The components of the desiccant cooling system have been implemented in SPARK with the aim to be accurate and easy to use. The modelling of the desiccant wheel is based on the analogy theory (Banks, 1972). The equations for coupled heat and mass transfer are reduced to two uncoupled differential equations of two independent variables called characteristic potentials which replace humidity ratio and enthalpy. In the psychometric 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 and London, 1984, Incropera and Dewitt, 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 and Marchio, 2004). 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 the effect of incidence angle for direct and diffuse radiation (Perers and Bales, 2002).

5. SIMULATION RESULTS

The room has a volume of 210 m^3. It has a double glazing southern facade and a vegetalized roof. Having a collector surface of 14.8 m^2, the simulations were run in order to optimize the storage volume. Besides they were used to study the effect of solar collector slope and air flow rate during inoccupation period on system performance and on system electrical consumption. The simulations were run for a period of three months from the first of June till 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 on system performance are studied by calculating several parameters such as the cooling requirement factor defined as (Belarbi and Allard, 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).

Figure 4 shows the evolution of the *IB* 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 almost the same). In fact, for smaller volumes the stored energy is limited because the storage temperature approaches 95°C and the IB curve is higher. 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 and the IB increases.

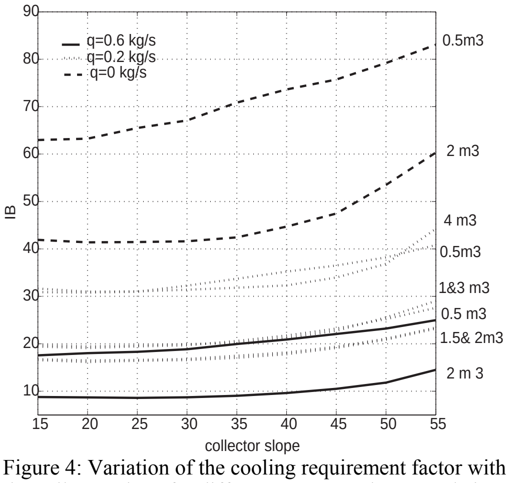

Figure 4: Variation of the cooling requirement factor with the collector slope for different storage volumes and air flow rates during inoccupation time.

Another key value was also investigated. It is the amount of hours where the storage temperature is less than 50°C while there is a need for regeneration. Its evolution is also similar to the curves of Figure 4, which is why it was not shown. From the Figure 4, 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.

Concerning equipment 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 on several parameters. It can be seen that as the air flow rate increases the installation needs to run less in desiccant mode and thus more solar energy is stored and the amount of hours where storage tank temperature is less than 50°C decreases but 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)) with the less electrical consumption growth (8%). The highest room temperature reached for more than 15 minutes during occupation period is also shown in table 1. More details are given in Figure 5 where room temperature and system running mode are plotted for q=0 and q=0.2 kg/s.

It can be seen that control strategy during inoccupation period has effect on room temperature especially before noon. As air flow rate increases, room temperature decreases and the installation switches to the desiccant mode more later (first day). This leads to storing more solar energy which allows the installation to run in case there is need for regeneration (second day before noon, for q=0 there is a need for regeneration but there is not enough stored energy to run the desiccant mode). In the end of the second day, it can be seen that for both flow rates, storage energy is consumed and that storage temperature is unable to provide regeneration temperature higher than 40°C (with slight room temperature decrease for q=0.2 kg/s). The same

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

Air flow rate (kg/s)	IB (°C hr)	Required desiccant time where Tstorage <50°C (hr)	Desiccant mode running time (hr)	Total electrical consumption (kWh)	Tmax (°C)
Q=0	43.25	69.33	127.8	833	28
Q=0.2	16.7	22.5	98.6	895	27.7
Q=0.4	11.01	12.25	82.41	1300	27.6
Q=0.6	8.8	9.75	80.91	1694	27.5

remark can be noticed for other days. So the effect of control strategy during inoccupation time is to decrease the installation desiccant mode running time, which allows storing more solar energy in the tank and decreases the amount of auxiliary hours where there is a need for regeneration

6. CONCLUSION

In this paper, an autonomous solar desiccant cooling system was investigated. The effect of storage volume, solar collector slope and control strategy during inoccupation period was studied. 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 showed the impact of air flow rate during inoccupation period. It was shown that, during inoccupation period, with even 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 the effect of building inertia and studying room comfort conditions).

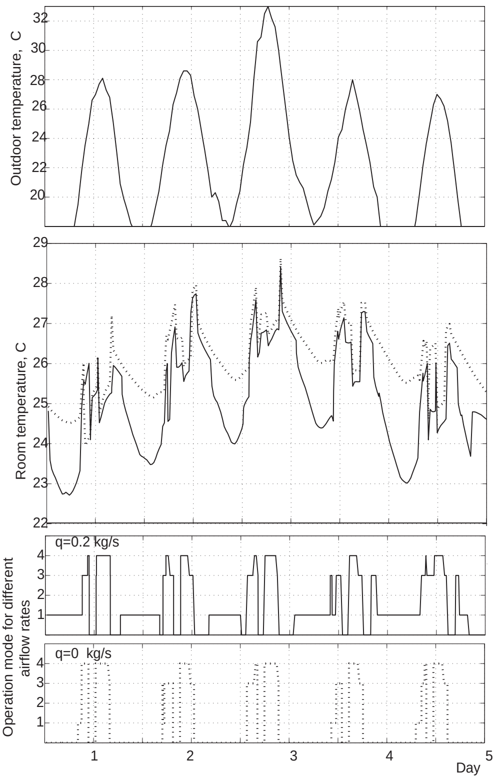

Figure 5: Evolution of outdoor and inside room temperature for different air flow rates during inoccupation time for the hottest week, with system running mode for each case: 1 ventilation mode, 3 indirect humidification mode and 4 desiccant mode.

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). The model will be validated experimentally. Then the simulations will be used to evaluate the cooling potential of the system for different climatic conditions and building configurations.

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