CONTROL STRATEGIES OF OPEN CYCLE DESICCANT COOLING SYSTEMS MINIMISING ENERGY CONSUMPTION

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ABSTRACT

Open-cycle desiccant cooling systems generally operate either in indirect evaporative cooling or in desiccant cooling mode depending on outdoor air conditions. In addition to the switch point temperature between evaporative and desiccant cooling, the regeneration temperature and the supply airflow rate are key control parameters to achieve optimal cooling performance. The influence of these control strategy parameters on the efficiency of desiccant cooling has been investigated.

The study of the performance of the desiccant cooling system versus the regeneration temperature and the air flow rate underlines that specific control strategies must be defined depending on reactivation energy: free energy (waste heat, solar energy) or no free energy (natural gas, district heat, electricity).

The components of the desiccant cooling system have been modelled including desiccant wheel, rotary heat exchanger and humidifiers with the aim to account for variable regeneration temperature and supply air flow rate. Both the system model and a building simulation software have been implemented in Matlab-Simulink® environment. The control strategy has been integrated by using a finite state machine representation.

This paper gives optimised control strategies for both energy source types in office building applications for a reference hot day. The results show how these HVAC systems are able to meet the cooling demand for different cases of building loads and thermal inertia and in different climates.

INTRODUCTION

Desiccant evaporative cooling is an alternative technology to conventional air conditioning systems. This technology, which is driven by thermal energy and uses no refrigerant, can reduce energy consumption, peak electricity demand and improve indoor air quality.

The ventilation cycle desiccant cooling system selected for this study is the most commonly used, due to its simplicity and the fact that the supply air is 100% outside air.

Kang and MacLaine-Cross (1989) focused on the dehumidifier which is the key component of a desiccant cooling system. Kodoma et al. (2000) investigated the impact of desiccant wheel rotation speed, air velocity and regeneration temperature on the cooling COP. It is shown that it exists an optimal rotation speed. Moreover, the COP decreases and conversely the cooling capacity of the desiccant cooling system augments with the increase of the airflow rate and the regeneration temperature. Henning et al. (2001) studied the potential of solar assisted desiccant cooling pilot plant. The authors listed several operation modes: ventilation, indirect evaporative cooling, desiccant cooling.

In order to minimise the energy consumption and maintain the indoor set point temperature, a sub optimal control strategy should be defined. The control strategy key parameters are the regeneration temperature, the airflow rate and the choice of the operation mode according to indoor and outdoor conditions. Furthermore, the source of thermal energy can be diverse, i.e. solar, waste heat, natural gas, district heating. The desiccant cooling control strategy with a costly energy source can not be the same as with a free energy source, since in the former case, the main constraint is to limit the primary energy demand and in the latter case, it is the source temperature level.

This paper investigates how the control strategy parameters can influence the cooling capacity and the primary energy demand of the desiccant cooling system. A simulation tool has been developed in order to evaluate the potential of desiccant cooling system and to analyse the influence of different control strategies on system performance for two energy types (typically natural gas and waste heat) in two French climate conditions and for different internal building loads.

DESSICANT COOLING SYSTEM

Desiccant evaporative cooling (DEC) consists in a combination of a dehumidification and an "adiabatic" humidification process (Figure 1). The outside air is dehumidified through a desiccant wheel. During the adsorption of vapour water in the wheel, the sensible heat of air is increased. After that the airflow temperature is lowered in a heat recovery wheel, water is sprayed into the process air, cooling it down further.

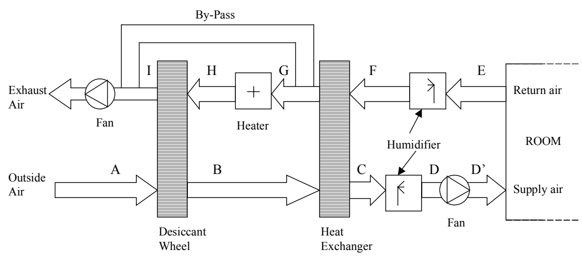


Figure 1: Schematic of a desiccant

The return air is cooled in an evaporative cooler and then is used to cool down the process air in the heat exchanger. Then, the return air is heated to regenerate the desiccant wheel. The states of the process and exhaust air are represented on a psychrometric chart (Figure 2).

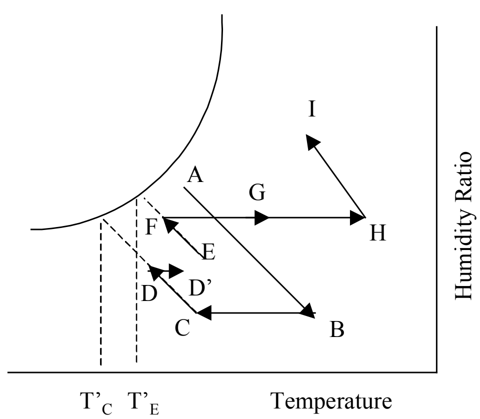


Figure 2: Psychrometric representation of desiccant cooling system

SIZING PARAMETERS

The desiccant evaporative cooling system is limited in cooling capacity and has too low performances in hot and humid climates. In order to augment the cooling capacity, the maximum regeneration temperature or the maximum air flow rate can be increased. However, the maximum airflow rate in desiccant wheels is limited by the maximum airflow velocity (about 3 m/s). Thus, high airflow rates can lead to very large sizes of air handling units. For that reason, simulations will be carried out with a maximum of 6 air changes per hour.

The characteristics of the desiccant cooling chosen for the simulations are described in Table 1. One part of the exhaust air is by-passed before the heater to limit the reactivation heat consumption. The pressure drop in the air-handling unit is more important than in usual units mainly due to desiccant wheel and rotary heat exchanger. The use of high efficiency fans limits electric consumption.

Table 1: Characteristics of the desiccant cooling components.

Component	At rating point
Desiccant wheel	Absolute humidity depression = 4.9g/kg at 34°C dry-bulb, 21°C wet-bulb and 95°C regeneration temperatures. Process area to regeneration area ratio: 50/50 Regeneration airflow rate to process air flow rate ratio: 0.8
Rotary heat exchanger	Effectiveness at equal hot and cold fluid heat capacity rates: $\frac{T_C - T_B}{T_F - T_B} = 0.8$
Direct evaporative humidifier	Saturation efficiency: $\frac{T_D - T_C}{T'_C - T_C} = 0.85$ With T, the dry bulb temperature and T', the wet bulb temperature.
Indirect evaporative humidifier	Saturation efficiency: $\frac{T_F - T_E}{T'_E - T_E} = 0.95$
Fans	Total efficiency: 0.8

SIMULATION TOOL

The simulations have been carried out in the Matlab Simulink® environment for two main raisons:

- Architecture of HVAC system and building simulation tool

The Matlab Simulink® tool is well adapted to link easily the HVAC components with each other and with the building model and what's more, simulation parameters such as time steps, error tolerances and solvers can be fitted to optimise both time and accuracy of the simulation. Output variables can be easily visualised with this modular tool. Moreover, a thermal building simulation program, SIMBAD Library (standing for SIMulator of Buildings And Devices) has been developed in this environment (CSTB, 2000).

The building model is based on a heat balance model (Riederer et al., 2000). Each thermal zone can be modelled as well–mixed with uniform temperature throughout. The description of the building envelope (dimensions, materials, windows, convection coefficients, boundary conditions…) is made easier by the user-friendly graphical interface.

A moisture balance model has been implemented in SIMBAD since the HVAC system performance is very dependent on humidity ratio. In the model (Riederer et al., 2000), moisture balance equations were not considered. The added mass balance module is a dynamic model considering air humidity as well-stirred.

The components of the desiccant cooling system have been specifically modelled in the Matlab Simulink® environment with the aim to be accurate and simple to parameterise. All the models are based on an analytical approach. The modelling of the desiccant wheel is based on the analogy theory (Banks, 1972, Banks, 1985). The equations for coupled heat and mass transfer are reduced to two uncoupled differential equations by introducing new dependent variables, called characteristic potentials, which replace enthalpy and humidity ratio. These new equations are analogous to heat transfer equations in a heat exchanger. Close and Banks (1972) noticed the characteristic potentials are closed to an enthalpy line and an adsorption isostere. Considering this assumption, the model has been developed.

The heat exchanger is modelled by a simple empirical formulation in ε-NTU (Kays and London, 1984). The variation of effectiveness with the airflow rate in the humidifiers is taken into account by using an ε-NTU model (Stabat et al., 2001).

- Control strategy representation

The simulation of different control strategies is made easier by using Stateflow®, a computer program linked to Matlab-Simulink®. This tool uses a finite state machine strategy (FSM) which is a representation of an event-driven (reactive) system.

In an event-driven system, the system transitions from one state to another occur by discrete events provided that the condition defining the change is true (Stateflow, 2001).

Usually, truth tables are used to represent relationships between inputs, outputs, and states of a FSM (Boole algebra). The resulting truth table describes the logic necessary to control the behaviour of the system (desiccant cooling components, e.g). The control strategy is described in this paper by a FSM made up of charts. A state-transition diagram (STD) gathers states linked by transitions and describes the logical operations of a subset of the FSM.

Figure 3 shows a Stateflow chart of the control of desiccant cooling mode. Two states are defined, one for desiccant wheel and direct humidifier active (state 2), and the other one for desiccant wheel and direct humidifier inactive (state 1). The transition represented by a change state arrow from one state to the other is caused if the condition on the indoor air temperature (T_{int}) is valid.

For instance, at initial conditions, if T_{int} is greater or equal to 25°C, the desiccant wheel and direct humidifier are on. If not, they are off. Assuming for example that state 2 is activated at start. While running, the state 2 is on as long as T_{int} is greater or equal to 25°C, and stops when T_{int} is lower than 24°C and state 1 becomes active. A hysteresis operation is thus computed.

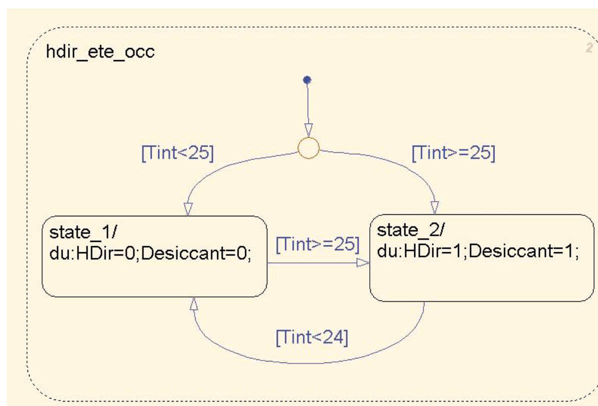


Figure 3: Graphical representation of the control of desiccant mode

CASE STUDIES

Inertia and loads of the building are key parameters to evaluate the DEC system potential in different climates. The study has been carried out on an office building zone with the following characteristics:

- Floor area of 3 m × 5 m = 15 m²
- Front wall of 3 m × 2.7 m including a double gazing window area of 3 m × 1 m
- Room height of 2.7 m

Two cases of inertia have been studied, medium and high following the definition of (Règles Th-I, 2000).

Two levels of internal gains (occupants and process such as computers, lighting…) are considered: 10 W/m² and 30 W/m² during occupancy. Only occupants are sources of moisture loads, the chosen value in all cases is of 7.3 g/(h.m²) during occupancy. The occupancy profile is taken from 8 a.m. to 6 p.m. during weekday. Two levels of solar gains are taken into account. A solar gain index (SGI) which represents the solar factor multiplied by window area and divided by floor area has been defined to aggregate solar gain parameters in one single variable (Table 2).

Table 2: Chosen values of Solar Gain Index

Inertia	medium		high	
SGI	0.05	0.13	0.05	0.15

Two French climatic zones are considered: Trappes (near Paris) and Nice. The design dry-bulb and wet bulb temperatures at 5% (ASHRAE, 1993) are 28°C and 19°C for Trappes respectively (28°C and 22°C for Nice). Weather data are those defined in French thermal regulation with a daily variation of humidity ratio (Bolher et al., 2001).

CONTROL STRATEGIES

Desiccant cooling control strategies depend on regeneration heat available. The open cycle desiccant cooling system can operate in ventilation, evaporative cooling or desiccant cooling mode according to the cooling demand.

- purchased reactivation energy

If the source of energy is not free of charge, the strategy should limit the use of the desiccant operation mode as much as possible. The principle consists in using indirect evaporative cooling (only the evaporative cooler in the return air and the energy recovery wheel) as long as it is sufficient to achieve the cooling demand (Figure 4).

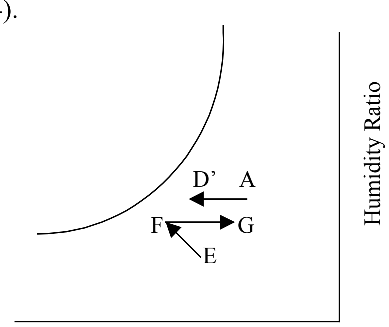


Figure 4: Psychrometric representation of indirect evaporative cooling system

As internal or external loads rise beyond the capabilities of indirect evaporative cooling, the desiccant wheel and the direct evaporative cooler are activated to operate in desiccant cooling mode. In this mode, the cooling power can be controlled by the reactivation temperature of the desiccant wheel and the supply airflow rate.

First of all, one can wonder which option between large air flow rates, $\dot{m}_{air}$, and high regeneration requires less energy (thermal plus electric). In first instance, Figure 5 shows that an increase in regeneration temperature, T_{reg}, from 55°C to 95°C does not improve so much the cooling power of the system, the supply air is decreased from 16.5 to 14.5°C only. The calculations have been made in steady state conditions for constant indoor and outdoor air conditions.

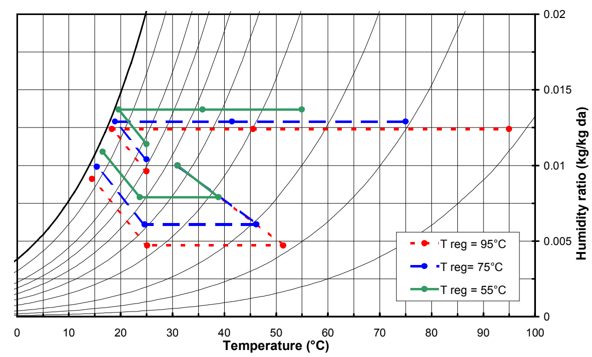


Figure 5: Psychrometric representation of an open desiccant cooling system for 3 regeneration temperatures

Figure 6 shows results for $40 \leq T_{reg} \leq 95$°C and $2 \leq \dot{m}_{air} \leq 6$ ac/h. It appears that high airflow rates is more interesting than high regeneration temperatures to achieve the cooling capacity.

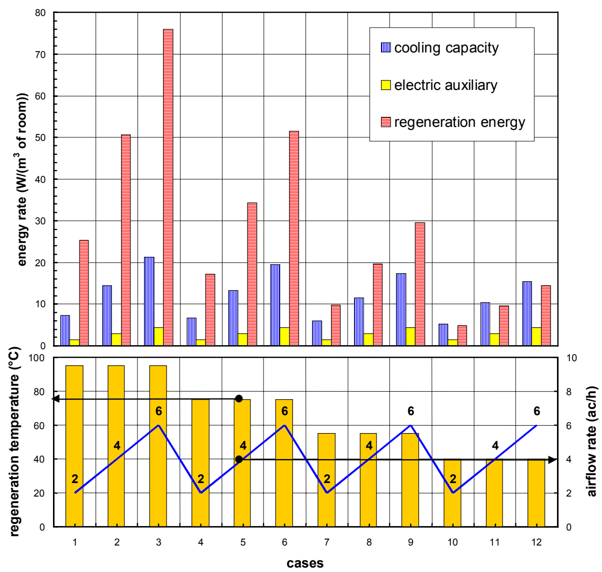
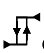
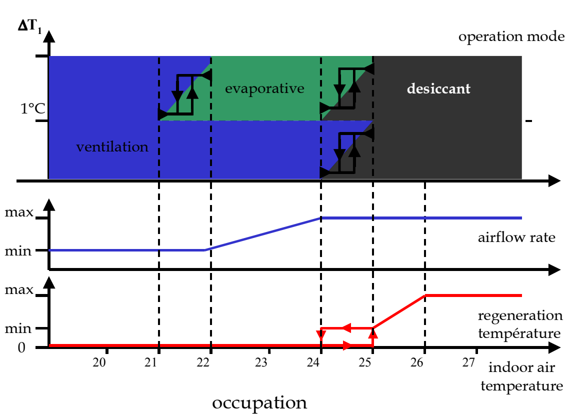


Figure 6: Impact of regeneration temperature and airflow rate on cooling capacity and energy rate for outdoor air conditions of 31°C and 35% and indoor air conditions of 25°C and 110g/h internal moisture gains

The minimum reactivation temperature has been taken to 40°C. Below this reactivation temperature, dehumidification of air is very low and the supply air temperature is closed to supply air temperature in evaporative mode.

As a result of this study, the control strategy defined for no free energy is described in Figure 7. Notice that:

 depicts hysteresis operation, $\Delta T_1 = T_A – T_F$ represents the difference between outdoor air temperature and air temperature at the outlet of the regeneration side humidifier and $\Delta T_2 = T_E - T_{D'}$ is the difference between indoor air temperature and supply air temperature.



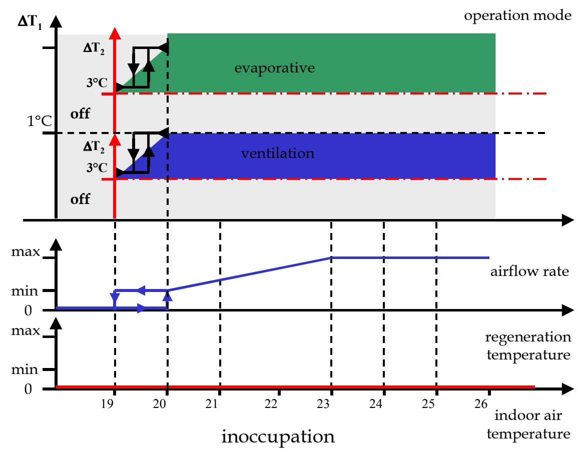


Figure 7: Control strategy diagram in case of purchased reactivation energy

A test on ΔT_1 compares if the evaporative cooling is more advantageous than ventilation. If evaporative and ventilation modes are authorised and if $\Delta T_1 > 1°C$, the DEC operates in indirect evaporative mode. A test on ΔT_2 checks if evaporative cooling or ventilation have a high potential of cooling. On this latter test, a hysteresis operation is computed in order to avoid short on-off cycles. If supply air ($T_{D'}$) is at least 4°C lower than room temperature (T_E), the HVAC system operates and if $T_E – T_{D'} < 3°C$, the HVAC system is stopped (this hysteresis operation is not represented on Figures 7 and 8).

At peak load days, the defined control strategy takes advantage of night cooling potential. Some tests have been carried out on a reference hot day to evaluate the impact of night cooling on the maximum air temperature during occupancy and on the global energy consumption of the plant (Tables 3 and 4).

Table 3: Performance of a night cooling strategy at Trappes

climate	TRAPPES			
inertia	medium		high	
inertal gains	30 W/m²		10 W/m²	
SGI	0.13		0.05	
orientation	WEST	EAST	WEST	EAST
Performance of DEC system **with night cooling strategy** (6 ac/h and 95°C)				
maximum indoor temperature (°C)	27.3	25.9	23.5	23.4
daily auxiliary consumption (Wh)	2765	2619	828	803
daily regeneration energy cons. (Wh)	11432	17693	0	0
Performance of DEC system **without night cooling strategy** (6 ac/h and 95°C)				
maximum indoor temperature	28.8	26.9	24.3	24.1
daily auxiliary consumption (Wh)	1739	1734	1067	1171
daily regeneration energy cons. (Wh)	20328	27715	0	0

Table 4: Performance of a night cooling strategy at Nice

climate	NICE			
inertia	medium		high	
inertal gains	30 W/m²		10 W/m²	
SGI	0.13		0.05	
orientation	WEST	EAST	WEST	EAST
Performance of DEC system **with night cooling strategy** (6 ac/h and 95°C)				
maximum indoor temperature (°C)	30.7	30.0	25.3	25.7
daily auxiliary consumption (Wh)	3311	2671	2125	1739
daily regeneration energy cons. (Wh)	23902	30801	6740	9542
Performance of DEC system **without night cooling strategy** (6 ac/h and 95°C)				
maximum indoor temperature	32.7	31.2	26	25.7
daily auxiliary consumption (Wh)	1739	1739	1739	1739
daily regeneration energy cons. (Wh)	32052	32087	9325	9565

From comfort point of view, a night cooling control strategy turns out to be more efficient to maintain set point temperatures. Moreover, it reduces deeply the thermal energy consumption for regeneration. Night cooling strategy does not lead to a sharp increase of electric consumption due to pumps, fans and motors in low load buildings but implies a much higher electric consumption in high load buildings. For a high load building in Trappes with an west orientation and in Nice, the control strategy appears to be insufficient to maintain set point conditions.

To pull down the switch point temperature between evaporative and desiccant cooling does not improve the room comfort but increases the energy consumption of the HVAC system as shown in Table 5.

Table 5: Influence of switch point temperature between evaporative and desiccant cooling on the DEC performance

Building characteristics				
medium inertia, SGI = 0.13, internal gains: 30W/m², East orientation				
DEC system				
6 ac/h maximum airflow rate and 95°C maximum regeneration temperature				
climate	TRAPPES		NICE	
switch point temperature	25°C-26°C	24°C-25°C	25°C-26°C	24°C-25°C
maximum indoor temperature (°C)	25.9	25.6	30.0	29.9
daily auxiliary consumption (Wh)	2619	2570	2671	2698
daily regeneration energy cons. (Wh)	17693	25837	30801	31677

- free reactivation energy

If the thermal energy is free such as waste heat or heat from a co-generation plant which is not fully used during summer, the cooling strategy should take advantage of this free energy and limit the electric consumption. High airflow rates to overcome the building loads should be used in the last resort. Some tests have been carried out to find a strategy minimising electric consumption (Figure 8). One considers that thermal energy is available only during occupancy. Night cooling is essential to maintain set point temperature in most cases. If thermal energy is available during inoccupancy, the night strategy would be different.

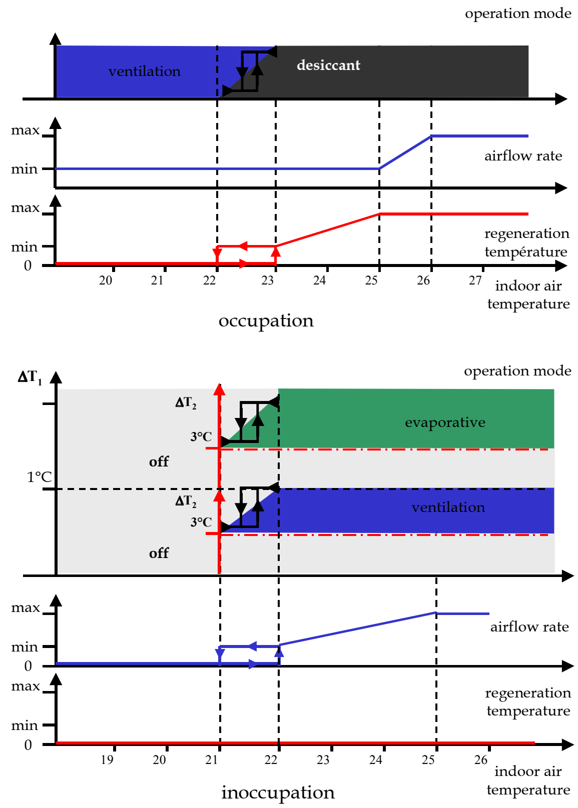


Figure 8: Control strategy diagram in case of free energy during day

A partial sensibility analysis has been carried out for this control strategy, only. An example of results is given hereafter in Table 9.

SIMULATION ANALYSIS

The potential of desiccant cooling system has been investigated (Tables 6 and 7) with control strategy on Figure 7 for the different case studies presented above. In low load buildings, DEC system can maintain set point conditions in Trappes and Nice. In Trappes, indirect evaporative cooling system turns out enough in low load buildings if used with high airflow rates. In high load buildings, DEC system is adapted in Trappes and can be applied in Nice only with high airflow rates and high inertia building. The thermal building inertia smoothes the daily thermal load curve.

Table 6: Performance of DEC system in Trappes

Maximum indoor air temperature in TRAPPES (purchased reactivation energy)					
inertia		medium		high	
SGI		0.05	0.13	0.05	0.15
internal gains		10 W/m²	30 W/m²	10 W/m²	30 W/m²
Max Airflow rate	Reac. Temp.				
2 ac/h	55°C	25.8		25.0	
	75°C	25.6		25.0	
	95°C	25.5		25.0	
4 ac/h	55°C	25.0		23.9	25.8
	75°C	25.0		23.9	25.6
	95°C	25.0		23.9	25.5
6 ac/h	55°C	24.5	26.8	23.4	25.0
	75°C	24.5	26.2	23.4	25.0
	95°C	24.5	25.9	23.4	25.0

with the following legend:

minimum regeneration temperature only (40°C)
evaporative cooling only
maximum indoor air temperature >27°C

Table 7: Performance of DEC system in Nice

Maximum indoor air temperature in NICE (purchased reactivation energy)					
inertia		medium		high	
SGI		0.05	0.13	0.05	0.15
internal gains		10 W/m²	30 W/m²	10 W/m²	30 W/m²
Max Airflow rate	Reac. Temp.				
2 ac/h	55°C				
	75°C			26.5	
	95°C			26.2	
4 ac/h	55°C			26	
	75°C	26.3		25.6	
	95°C	26.0		25.4	
6 ac/h	55°C	26.0		25.6	
	75°C	25.5		25.4	26.9
	95°C	25.4		25.3	26.5

In some cases, two DEC system sizing options can be considered, either 6 ac/h maximum airflow rate and 55°C maximum regeneration temperature or 4 ac/h maximum airflow rate and 95°C maximum regeneration temperature. Table 8 shows results on the daily consumption of the DEC system in two options for a reference hot day. The solution with high airflow rates and low reactivation temperature seems to be more advantageous even if more electricity is used, since the thermal energy consumption is decreased. The choice of one option depends on the relative price between thermal energy and electricity and initial costs since a 6 ac/h air-handling unit plant is more expensive.

The COP has been calculated as:

$$COP = \frac{\text{total cooling energy}}{\text{thermal energy}}$$

Fans are not taken into account in the calculations of the COP since they do not contribute to cooling but are an indispensable equipment in every central air-handling unit. Motor and pump consumption can be neglected compared to thermal energy consumption. It should be pointed out that DEC systems induce water consumption. The daily water consumption for the reference hot day and for this 15 m² room is about 13 litres for the 4 ac/h option (calculations have been made with a bleed off rate of 20% of evaporation rate). High airflow rate option implies a slight increase of water consumption.

Table 8: daily energy comparison between two DEC system sizing options

Case: Nice, medium inertia, SGI=0.05, 10W/m² internal gains		
maximum air flow rate	4 ac/h	6 ac/h
regeneration temperature	95 °C	55 °C
maximum indoor temperature (°C)	26.0	26.0
minimum supply air during occupancy (°C)	18.7	19.2
daily electricity consumption (Wh)	1319	1899
daily thermal energy consumption (Wh)	13321	10260
total cooling energy (Wh)	4298	4278
sensible cooling energy during occupancy (Wh)	3253	3235
sensible cooling energy during inoccupancy (Wh)	267	259
electricity consumption during inoccupancy (Wh)	160	160
mean COP during occupancy	0.30	0.39

In case of a free energy source, some results are presented in Table 9. The control strategy for a free energy source gives similar results in terms of comfort compared to the one for no free energy. In Trappes, even in the worst case studied here, the DEC system turns out to be well adapted to these climatic conditions. In Nice, the DEC system is insufficient to meet the cooling demand in high load buildings. During night, the indoor temperature does not drop below 25°C even with evaporative cooling. The use of desiccant operation mode during night turns out to be necessary to pre-cool the building and so anticipate day loads.

Table 9: Performance of DEC system in Nice and in Trappes with a control strategy adapted to a free thermal energy source

Building characteristics		
medium inertia, SGI = 0.13, internal gains: 30W/m², East orientation		
DEC system		
6 ac/h maximum airflow rate and 95°C maximum regeneration temperature		
Climate	Trappes	Nice
maximum indoor temperature (°C)	26.2	30.0
minimum supply air during occupancy (°C)	15.0	15.9
daily electricity consumption (Wh)	1860	2608
daily thermal energy consumption (Wh)	28476	31189
sensible cooling capacity during occupancy (Wh)	6970	7726
sensible cooling capacity during inoccupancy (Wh)	2025	1417
electricity consumption during inoccupancy (Wh)	685	912

CONCLUSIONS

The tests on control strategies show that the night cooling is essential to anticipate the day thermal loads since the cooling capacity of the DEC system is quite limited. The use of high regeneration temperature does not improve so much the cooling capacity of the system.

The use of the graphical computer tool is an interesting way to test several control strategies. The key control parameters such as switch points, regeneration temperatures, air flow rates can be easily and quickly modified. On the one hand, the simulation time is quite long, 6 day simulation takes about fifteen minutes with Pentium IV 1.9 GHz. On the other hand, short cycle problems can be pointed out thanks to short time steps.

Desiccant evaporative cooling system is suitable for climates like Trappes. In Nice, it could only be used in low thermal load buildings. Electric consumption of this air-conditioning system is reduced compared to conventional mechanical cooling however the thermal energy consumption is quite high due to a low cooling COP in desiccant operation. Cheap or free thermal energies can help to the development of DEC systems. Solar energy has the advantage to be the only free energy available everywhere. Future work should be led to determine an adapted control strategy for a DEC system coupled with a solar collector.

ACKNOWLEDGEMENTS

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REFERENCES

ASHRAE (1993), *ASHRAE Handbook, Fun*damentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, inc., Atlanta.

Banks P.J. (1972), 'Coupled equilibrium heat and single adsorbate transfer in fluid flow through a porous medium - I Characteristic potentials and specific capacity ratios', *Chemical Engineering Science*, Vol. 27.

Bolher A., Fleury E., Marchio D., Millet J.R., Stabat P. (2001), 'Indirect Evaporative Cooling System – Tools and Sizing Guidelines'*, Clima 2000*, Napoli, September 2001.

Close D.J., Banks P.J. (1972), 'Coupled equilibrium heat and single adsorbate transfer in fluid flow through a porous medium - II Predictions for a silica-gel air-drier using characteristic charts', *Chemical Engineering Science*, Vol. 27.

Banks P.J. (1985), 'Prediction of Heat and Mass Regenerator Performance Using Nonlinear Analogy Method: Part 1-Basis', *Transactions of ASME*, Vol. 107.

CSTB – SIMBAD Building and HVAC Toolbox (version 2.0.0, November 2001)

Riederer P., Marchio D., Gruber P., Visier J.C., Larech R., Husaundee A. (2000), 'Building zone modelling adapted to the study of temperature controlled systems', *ASHREA/CIBSE conference 2000*, *Dublin, Ireland.*

Kang T.S., McLaine-Cross I.L. (1989), 'High performance, solid desiccant, open cooling cycles', *ASME Journal of Solar Energy Engineering*, Vol. 111.

Kays W.M. and London A.L. (1984), *Compact heat exchangers*, 3rd edition, Mc Graw-Hill, New-York.

Kodoma A., Jin J., Goto M., Hirose T., Pons M. (2000), 'Entropic analysis of adsorption open cycles for air conditioning. Part 2: interpretation of experimental data', *International Journal of Energy Research*, Vol. 24.

Henning H-M., Erpenbeck T., Hindenburg C., Santamaria I.S. (2001), 'The potential of solar energy use in desiccant cooling cycles'*, International Journal of Refrigeration*, Vol. 24.

Règles Th-I 2000, *Guide Réglementation Thermique 2000,* Annexe 3, P. 13, CSTB Publications-Diffusions.

Stabat P., Marchio D., Orphelin M. (2001), 'Pre-design and design tools for evaporative cooling', *ASHRAE Transactions.*

Stateflow user's manual (2001), The Mathworks Company.