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## 3 **A dynamic multi-level model for adsorptive solar cooling**

4 Giulio Santori, Alessio Sapienza, Angelo Freni

### 6 **Abstract**

7 This paper focuses on the development of a dynamic multi-level model for simulating of a solar cooling system  
8 adopting an adsorption chiller. The model integrates detailed simulation of the adsorption cycle (component level) into  
9 the transient simulation of the solar cooling system (system level). The chiller investigated was a standard two-bed  
10 silica gel/water unit. The model was used to ascertain the feasibility of solar-driven adsorption cooling and for  
11 optimization purposes. In the base case simulated, the adsorption chiller cooled down the outdoor air to 16.1°C. The  
12 daily average COP of the chiller was 0.18. Consequently, a spectral analysis was performed on these data for  
13 identification of the correlation among the variables involved in the solar cooling system in order to study the effects of  
14 the input parameters on the outputs. The outcomes were that the  $COP$  depends mainly on  $T_{ev} > Q_{cond} > Q_{cool} > Q_{heat}$  and the  
15 thermal power delivered to the hot storage unit correlates  $T_{z1} > T_c$ .

17 **Keywords:** *solar cooling, adsorption, dynamic model, silica gel, refrigeration, spectral analysis.*

### 19 **1.Introduction**

20 Vapor compression refrigeration systems demand large quantities of electrical energy and involve mechanical parts in  
21 motion. The sustainability of using synthetic refrigerants is also an issue. On the other hand, adsorption refrigeration  
22 systems need low-temperature thermal energy to perform a thermodynamic cycle and they do not need a compressor,  
23 reducing electrical energy consumption. Adsorption refrigeration systems also have the advantages of being  
24 environment-friendly, as water can be used as working fluid having no ozone depletion or global warming potential.  
25 Adsorption refrigeration technologies driven by waste heat or renewable energy have thus attracted interest in recent  
26 years [1, 2]. The technologies are still under development but few commercial products are ready in the market [3-5].  
27 Studies have been conducted on numerous adsorption systems driven by solar energy, focusing mainly on machines  
28 using silica gel and water because they have a relatively low regeneration temperature (<100°C) [6-8]. Field tests have  
29 shown that silica gel adsorption chillers can be efficiently operated using a solar energy source [9].

30 Different theoretical models have been proposed to describe the chiller's operation [5, 10-13]. The earlier models  
31 presented the response of the machine operating in steady-state conditions for few successive cycles [13]. More  
32 recently, a model was proposed for predicting the behavior of a silica gel/water chiller by studying the functioning of  
33 numerous successive cycles in terms of dynamic heating conditions, condensation and evaporation, which revealed a  
34 good agreement with experimental data [5]. To characterize the machine in transient conditions, theoretical studies were  
35 performed [10, 11] in which various time-dependent functions (linear, sinusoidal, step, pulse) were applied to the  
36 temperature of the heat carrier fluid for desorption. The development of these models has shown that:

- 37 1) it is difficult to generalize on the machine's behavior for a given plant setting;
- 38 2) The start-up transient, up until the machine reaches a steady state, must be thoroughly investigated;
- 39 3) the management of the machine's internal logic (end-phase conditions) needs to be accurately studied in order to  
40 optimize the COP in variable weather conditions.
- 41 4) a slow response to changes in the input data may occur, which makes it difficult to quantify the effect of the  
42 parameters on the chiller's operation [12];
- 43 5) the machine must be inserted in a realistic hydraulic context, with the consequent problem of controlling the  
44 evolution of a large number of parameters;
- 45 6) the machine's management, during the start-up transient and in the passage between the chiller's operating stages,  
46 must be optimized when the conditions of the external hydraulic loops are not stationary and predictable.

47 To address these issues is necessary improving previous models and a deeper analysis of derived data. This means  
48 developing mathematical models with two main features:

- 49 1) dynamic: capable of processing weather data input that varies in time;
- 50 2) multi-level [14]: able to follow the evolution of the main quantities in the single subsystems (adsorption machine,  
51 storages, air treatment unit, etc..) and contemporary the solar cooling system, to perform black box type or detailed  
52 considerations;

53 The present work describes a dynamic multi-level model for assessing the performance of an adsorption chiller and its  
54 interactions with a realistic solar-powered setting. The two main parts into which the model can be divided are  
55 described separately below. Finally a spectral analysis of the data is proposed in order to estimate the correlations  
56 among the variables and the climatic data.

### 58 **2.Plant description (base case)**

59 Fig. 1 shows the layout of the solar cooling system for air conditioning. It consists of an adsorption chiller (rated 3kW)

1 operating with four heat exchangers for the condensation, evaporation and ad/desorption of two beds. The considered  
 2 adsorption chiller was similar to the chiller reported in [15]. The heat of condensation and adsorption are dissipated by a  
 3 remote dry cooler (rated 8kW). The evaporator is used to produce the cooling effect by providing refrigerated water.  
 4 The chiller is connected to two thermal storage units for managing the peak cooling demand and thermal availability.  
 5 Both the storage units are stratified in three overlapping layers. The hot storage unit (2400 liters) is powered by a flat  
 6 field of solar thermal collectors (40m<sup>2</sup>) and a backup boiler (rated 30kW). The cold storage unit (900 liters) serves an air  
 7 treatment unit (ATU). The ATU takes outdoor air (0.2m<sup>3</sup>/s) and treats it in cooling and dehumidification, adiabatic  
 8 mixing and post-heating sections. For the solar collector field, the feed pump is operated just when the temperature of  
 9 the lower layer in the hot storage unit ( $T_{h,lay3}$ ) is lower than the mean temperature of the solar collector field plates ( $T_{fp}$ ).  
 10 All the components are operated from 9.00 to 19.00 hours, except for the solar collector field which is activated  
 11 independently.

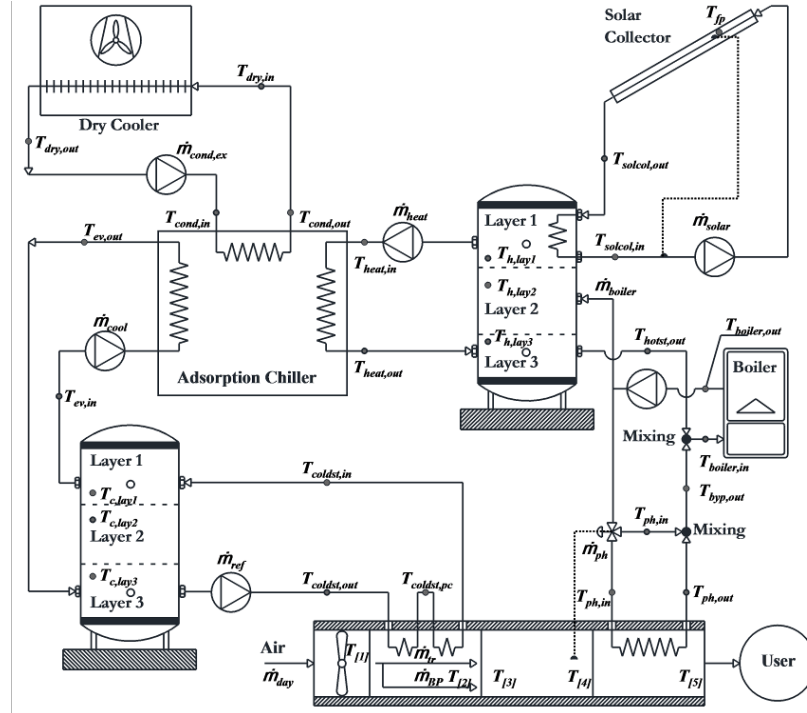


Figure 1. Diagram of the process and instrumentation at the plant

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### 15 3.The input data

16 The climatic data for the town of Ancona, Italy were acquired hourly. The data available were the average values of  
 17 global radiation (horizontal plane), ambient temperature and relative humidity. Data were acquired from 1 June 2007 to  
 18 30 August 2007. The radiation data were adjusted to give the global radiation on a plane tilted at 30° [16]. Then the data  
 19 were statistically processed to obtain a single day representative of the summer period. Each hourly data outside the  
 20 99.5% confidence interval were excluded. The hourly means were calculated on the remaining values, to obtain one  
 21 representative day for each month considered so obtaining three days. The three days was averaged obtaining a single  
 22 day which was repeated three times, determining the set of climatic data input. Figure 2 shows the trends of the solar  
 23 radiation, temperature and relative humidity as input for the model.

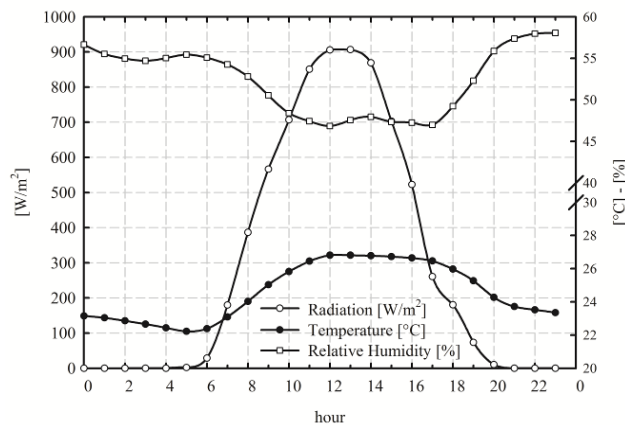


Figure 2. Trend of climatic data for the fictitious day

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#### 4. The adsorption chiller model

The unit involved in this study is a two-bed, silica gel-water adsorption chiller with no regeneration. Even though the previously proposed multi-bed regenerative and re-heated chillers show higher performance [17, 18] this basic configuration was adopted to avoid introducing effects on the results relating to the type of machine, and thereby obtain more general information. The following assumptions were stated:

- 1) the adsorbent was always in equilibrium with the adsorbate
- 2) the gaseous phase behaves as an ideal gas
- 3) the properties of the solid and the gaseous phase are assumed constant
- 4) all the thermal losses are negligible

The mathematical system describing the operation of the chiller is summarized in by eqs. (1) to (5). The indices for the eqs. from (1) to (5) are given in Table 1. Since the equations recurring at each stage are very similar and the model of the plant as a whole contains a very large number of equations, it seemed preferable to present them in compact form. Thus, by following the indexing in Table 1, we can track the energy and mass balances for each phase. Energy and mass balance in the adsorption machine (considering one bed) is:

$$(M_z c_{z,eq} + M_{z,ex} c_{z,ex}) \frac{dT_z}{dt} - M_z \Delta H_{ads} \frac{dw}{dt} \delta = U_{z,ex} A_{z,ex} (T_{\alpha,in} - T_{\alpha,out}) \quad (1)$$

The energy balances of the packed bed heat exchanger (eq. 16) and for the water inside the machine (eq. 17) are:

$$T_{\alpha,out} = T_z + e \frac{A_{z,ex} U_{z,ex}}{\dot{m}_\mu c_{pw}} (T_{\alpha,in} - T_z) \quad (2)$$

$$M_z \frac{dw}{dt} L_w = -U_{\beta,ex} A_{\beta,ex} (T_\gamma - T_{\beta,ex}) \delta - c_{\beta,ex} m_{\beta,ex} \frac{dT_{\beta,ex}}{dt} \delta \quad (3)$$

The energy balances for the water in the evaporator and condenser (machine side eq. 18, plant side eq. 19):

$$M_{\beta,ex} c_{\beta,ex} \frac{dT_{\beta,ex}}{dt} = U_{\beta,ex} A_{\beta,ex} (T_\gamma - T_{\beta,ex}) \delta + \dot{m}_{\beta,ex} c_{pw} (T_{\beta,in} - T_{\beta,out}) \delta \quad (4)$$

$$T_{\beta,out} = T_{\beta,ex} + e \frac{A_{\beta,ex} U_{\beta,ex}}{\dot{m}_\sigma c_{pw}} (T_{\beta,in} - T_{\beta,ex}) \quad (5)$$

Auxiliary equations needed to solve the system are given in the Appendix. Figure 3 shows the chiller circuit in the various phases of the process and how the fluid flows are diverted during the different phases.

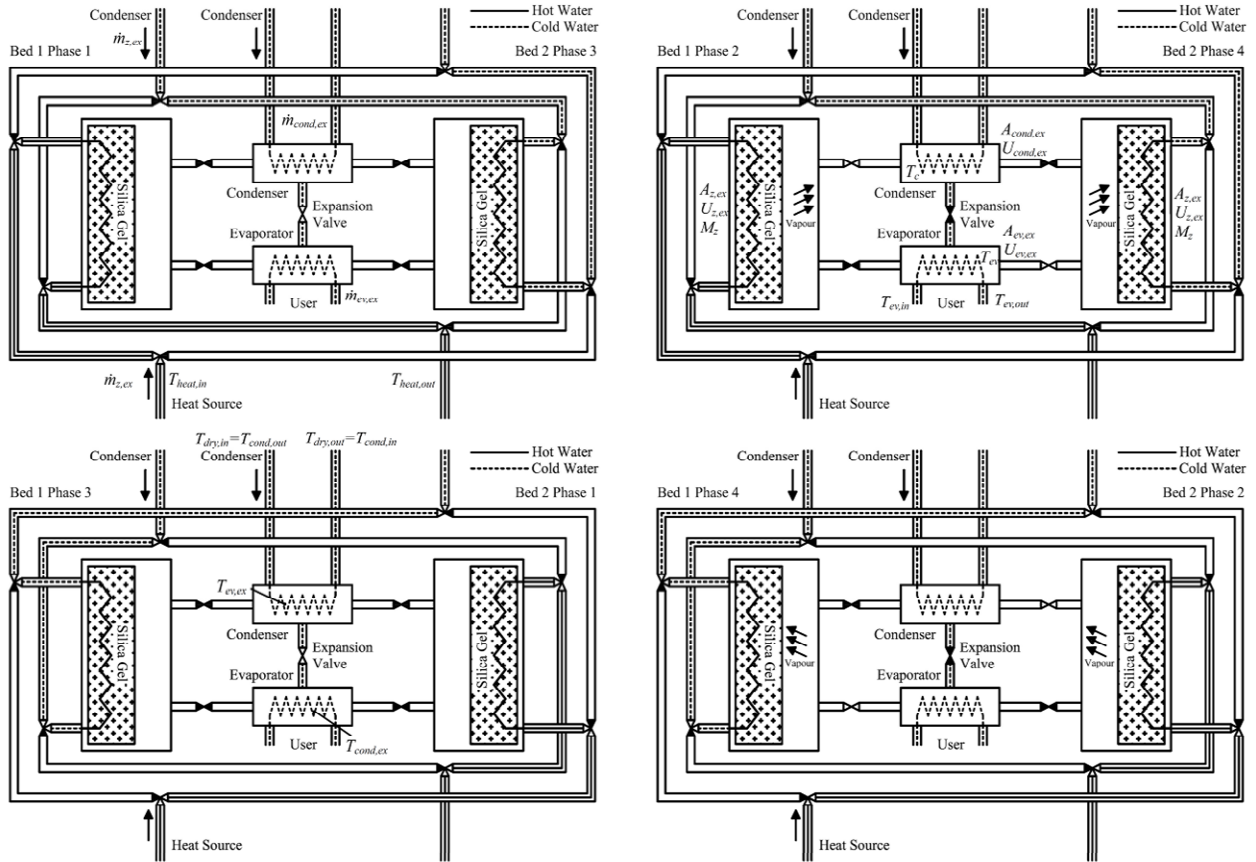


Figure 3. Chiller phases

The equations to solve are enabled or disabled according to the succession of phases in Figure 3. It was consequently implemented a control system that not only determines the right sequence of the phases for each bed, but also guarantees the proper offsetting of the thermodynamic cycles of the two beds. The two conditions for enabling the passage from one phase to the other are:

- 1) thermodynamic condition: when certain temperature and pressure conditions are reached (Table 1).
- 2) temporal condition: the maximum time allowable to complete a phase; if the thermodynamic condition is not met, the phase changes anyway after a certain preset time interval (Table 1).

Table 1 details the thermodynamic and temporal conditions for each phase in the thermodynamic cycle.

Table 1: End of phase conditions and indexing for equations (1)-(5)

	Ph.I	Ph.II	Ph.III	Ph.IV
$\alpha$	heat	heat	dry	Dry
$\beta$	cond	cond	ev	Ev
$\gamma$	c	c	ev	Ev
$\sigma$	cond	cond	ev,ex	ev,ex
$\mu$	heat	heat	ev,ex	ev,ex
$\delta$	0	1	0	1
End of phase conditions				
Thermodynamic <sup>a</sup>	$p_z > p_c$	$T_z > T_{sup}$	$p_z > p_{ev}$	$T_z > T_{inf}$
Temporal <sup>b</sup>	$t_{end,1}$	$t_{end,2}$	$t_{end,3}$	$t_{end,4}$

<sup>a</sup> the temperature  $T_{sup}$  is calculated as the sum of the temperature at the start of the phase plus  $\Delta T_{sup}$  (Table 3); the temperature  $T_{inf}$  is calculated as the difference, i.e. the temperature at the start of the phase less  $\Delta T_{inf}$  (Table 3); the pressures at the end of phases 1 and 3 are the saturation pressures at the water condensation and evaporation temperatures, respectively.

<sup>b</sup>  $t_{end,1,2,3,4}$  are calculated from the sum of the phase start time plus a fixed  $\Delta t$  (Table 3).

If only one bed satisfies the thermodynamic condition, such bed waits until the thermodynamic condition or the temporal condition of the other bed is met. On the other hand, if one of the two beds satisfies the temporal condition, the phase transition is allowed even if the other bed has not satisfied the thermodynamic or temporal condition. The thermodynamic condition thus functions as an AND type of logical gate, the temporal as an OR type of logical gate.

The logic for differentiated variables initialization was a fundamental aspect to run the model. About such logic the values used for initializing the variables on the first and on subsequent days during the start-up transient were different. For a limited number of variables, for the days after the first, the ambient temperature was taken as initializing temperature, in other cases the last temperatures of the previous simulated day was taken.

Finally the values of  $T_{ev}$  and  $T_c$  were obtained from the two beds following the logic of the phases. The value of  $T_{ev}$  or  $T_c$  was thus read from one bed or the other, depending on which bed is active.

## 5. The plant model

The study of the solar cooling system demands the mathematical description of the loops involved and the storage units, therefore the use of several variables in addition to those concerning the chiller's operation. Apart from the adsorption chiller, the other systems involved in the plant are shown in Figure 1.

The mathematical equations of the plant were implemented based on the following assumptions:

- 1) negligible internal temperature gradients in every component (except the storage units);
  - 2) air was considered as an ideal gas;
  - 3) the specific heat and density of the air and water were assumed to be constant within the operating temperature range;
- The governing equations are listed in Table 2. The model is useful both for drawing general conclusions and for more detailed considerations on each subsystem of the plant.

The storage units were assumed to consist of three overlapping layers and the ATU of sections for cooling, dehumidification, adiabatic mixing and post-heating. In the ATU the outside air is taken at ambient conditions and subsequently treated according to the straight line of the ambient thermal factor until the dew point where dehumidification begins. Successively the air is sent for adiabatic mixing and post-heating. Then it is released into a user-room. This section operates between 9.00 and 19.00 hours.

However this multi-level approach demands the introduction of control systems to harmonize the operation of the subsystems and to make the start-ups and shut-downs consistent. This means implementing a logic to reinitialize the differentiated variables according to their value at a previous point in time.

The operating parameters are shown in Table 3. The initial conditions for the hot and cold storage units and the metal plate of the solar collector, respectively, are:

$$T_{hlay,1} = T_{hlay,2} = T_{hlay,3} = 60^{\circ}\text{C}; T_{clay,1} = T_{clay,2} = T_{clay,3} = 10^{\circ}\text{C};$$

$$T_{fp} \Big|_{t=start} = T_{Amb} \Big|_{t=start}$$

These are the initial conditions at time t=0, when the whole simulation begins. The values for the initial conditions at the moment when the components are enabled on subsequent days differ from the previous values for the first day because during the night a subset of equations continues to be solved to take the heat losses into account. These losses are calculated for all the components except the chiller, which is assumed to be perfectly insulated. The post-heating valve control relies on a trend proportional to the adiabatic mixing temperature. The flow sent to the post-heating section and the boiler operating curve comply with:

$$\left[ \begin{array}{l} \dot{m}_{ph,in} = \dot{m}_{ph} \quad T_{[4]} < T_{inf} \\ \dot{m}_{ph,in} = \dot{m}_{ph} \left( \frac{T_{sup} - T_{[4]}}{T_{sup} - T_{inf}} \right) \quad T_{inf} < T_{[4]} < T_{sup} \\ \dot{m}_{ph,in} = 0 \quad T_{[4]} > T_{sup} \end{array} \right]$$

$$\left[ \begin{array}{l} Q_{boiler} = P_{boiler} \dots \dots \dots \text{if } T_{boiler,in} < T_{setboiler,min} \\ Q_{boiler} = \frac{P_{boiler} (T_{setboiler,max} - T_{boiler,in})}{T_{setboiler,max} - T_{setboiler,min}} \dots \dots \dots \\ \dots \dots \dots \text{if } T_{setboiler,min} < T_{boiler,in} < T_{setboiler,max} \\ Q_{boiler} = 0 \dots \dots \dots \text{if } T_{boiler,in} > T_{setboiler,max} \end{array} \right]$$

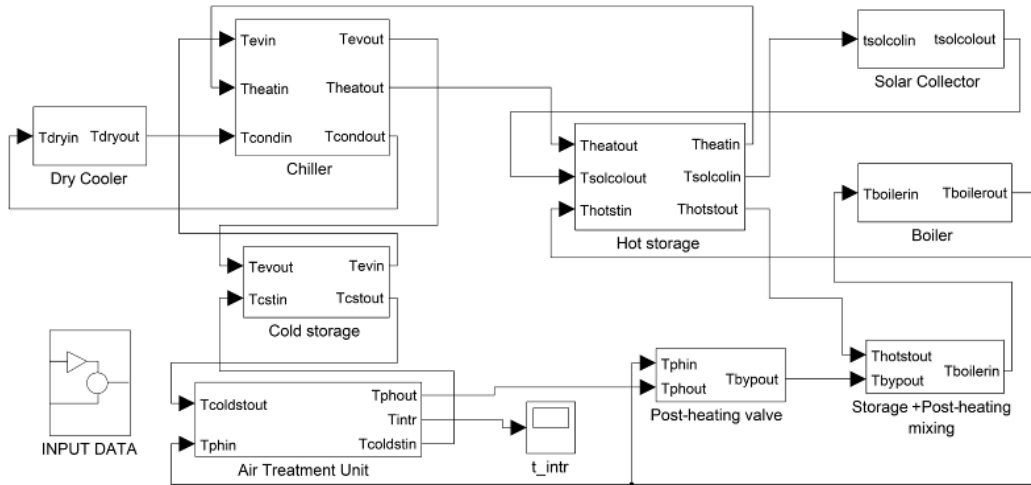


Figure 4. Matlab/Simulink® model of the system

The mathematical model was developed entirely with Matlab/Simulink® 7.6.0 using a block programming approach. The model's graphical interface is shown in Figure 4. The differential-algebraic equations were solved numerically with a second-order modified Rosenbrock solution method named ode23s.

Table 2: Equations governing the solar cooling system.

Energy and mass balances in the ATU. Cooling (eqs. 9-12), dehumidification (eqs. 13-14), adiabatic mixing (eqs. 15-17), post-heating (eqs. 18-20):

$$\varepsilon \dot{m}_{tr} c_{pa} (T_{[1]} - T_{coldst,out}) = \dot{m}_{tr} \left[ c_{pa} (T_{[1]} - T_{[2]}) + r (x_{[1]} - x_{[2]}) \right] \quad (9)$$

$$RF \varepsilon \dot{m}_{tr} c_{pa} (T_{[1]} - T_{coldst,out}) = \dot{m}_{tr} c_{pa} (T_{[1]} - T_{[2]}) \quad (10)$$

$$\dot{m}_{tr} \left[ c_{pa} (T_{[1]} - T_{[2]}) + r (x_{[1]} - x_{[2]}) \right] = \dot{m}_{ref} c_{pw} (T_{coldst,pc} - T_{coldst,out}) \quad (11)$$

$$x_{[R]} = x_{[1]} - \frac{(x_{[1]} - x_{[2]}) (T_{[1]} - T_{[R]})}{T_{[1]} - T_{[2]}} \quad (12)$$

$$\varepsilon \dot{m}_{tr} c_{pa} (T_{[R]} - T_{coldst,out}) = \dot{m}_{tr} \left[ c_{pa} (T_{[R]} - T_{[3]}) + r (x_{[R]} - x_{[3]}) \right] \quad (13)$$

$$\dot{m}_{tr} \left[ c_{pw} (T_{[R]} - T_{[3]}) + r (x_{[R]} - x_{[3]}) \right] = \dot{m}_{ref} c_{pw} (T_{coldst,in} - T_{coldst,pc}) \quad (14)$$

$$\dot{m}_{air,day} = \dot{m}_{tr} + \dot{m}_{BP} \text{ with } \dot{m}_{BP} = BP \dot{m}_{air,day} \quad (15)$$

$$\dot{m}_{air,day} x_{[4]} = \dot{m}_{tr} x_{[3]} + \dot{m}_{BP} x_{[1]} \quad (16)$$

$$\dot{m}_{air,day} H_{[4]} = \dot{m}_{tr} H_{[3]} + \dot{m}_{BP} H_{[1]} \quad (17)$$

$$\dot{m}_{ph,in} c_{pw} (T_{ph,in} - T_{ph,out}) = \dot{m}_{day} c_{pa} (T_{[5]} - T_{[4]}) \quad (18)$$

$$x_{[5]} = x_{[4]} \quad (19)$$

$$\dot{m}_{ph,in} c_{pw} (T_{ph,out} - T_{ph,in}) = U_{ph,ex} A_{ph,ex} \Delta T_{lm,ph} \quad (20)$$

Energy balances in the mixing points downstream from the post-heating exchanger (eq 21) and before the boiler inlet (eq 22):

$$\dot{m}_{ph} c_{pw} T_{byp,out} = (\dot{m}_{ph} - \dot{m}_{ph,in}) c_{pw} T_{ph,in} + \dot{m}_{ph,in} c_{pw} T_{ph,out} \quad (21)$$

$$(\dot{m}_{boiler} + \dot{m}_{ph}) c_{pw} T_{boiler,in} = \dot{m}_{boiler} c_{pw} T_{hotst,out} + \dot{m}_{ph} c_{pw} T_{byp,out} \quad (22)$$

Solar collector (eqs. 23-24), boiler and dry cooler (eq. 25) and dry cooler (eq. 26) energy balances:

$$M_{fp} c_{fp} \frac{dT_{fp}}{dt} = (\tau \alpha)_{eff} I_{tilt} A_{coll} - \dot{m}_{solar} c_{pw} (T_{solcol,out} - T_{solcol,in}) - U_{fp/Amb} A_{fp/Amb} (T_{fp} - T_{Amb}) \quad (23)$$

$$U_{fp} A_{fp} \Delta T_{lm,fp} = \dot{m}_{solar} c_{pw} (T_{solcol,in} - T_{solcol,out}) \quad (24)$$

$$T_{boiler,out} = T_{boiler,in} + \frac{Q_{boiler}}{P_{boiler}} \quad (25)$$

$$T_{dry,in} = T_{Amb} + (T_{dry,out} - T_{Amb}) e^{-\frac{U_{dry} A_{dry}}{c_{pw} (\dot{m}_{cond,ex} + \dot{m}_{z,ex})}} \quad (26)$$

Energy balances in the cold (eqs. 27-30) and hot water (eqs. 31-35) storage units:

$$V_{c,lay1} c_{pw} \rho_w \frac{dT_{c,lay1}}{dt} = \dot{m}_{ref} c_{pw} (T_{coldst,in} - T_{c,lay1}) + \dot{m}_{ev,ex} c_{pw} (T_{c,lay2} - T_{c,lay1}) + \dots \quad (27)$$

$$\dots U_{coldst} A_{c,lay1} (T_{Amb} - T_{c,lay1}) + U_w A_{c,lay} (T_{c,lay2} - T_{c,lay1})$$

$$V_{c,lay2} c_{pw} \rho_w \frac{dT_{c,lay2}}{dt} = \dot{m}_{ref} c_{pw} (T_{c,lay1} - T_{c,lay2}) + \dot{m}_{ev,ex} c_{pw} (T_{c,lay3} - T_{c,lay2}) + \dots \quad (28)$$

$$\dots U_{coldst} A_{c,lay2} (T_{Amb} - T_{c,lay2}) + U_w A_{c,lay} [(T_{c,lay1} - T_{c,lay2}) + (T_{c,lay3} - T_{c,lay2})]$$

$$V_{c,lay3} c_{pw} \rho_w \frac{dT_{c,lay3}}{dt} = \dot{m}_{ref} c_{pw} (T_{c,lay2} - T_{c,lay3}) + \dot{m}_{ev,ex} c_{pw} (T_{ev,out} - T_{c,lay3}) + \dots \quad (29)$$

$$\dots U_{coldst} A_{c,lay3} (T_{Amb} - T_{c,lay3}) + U_w A_{c,lay} (T_{c,lay2} - T_{c,lay3})$$

$$T_{ev,in} = T_{c,lay1}; T_{coldst,out} = T_{c,lay3} \quad (30)$$

$$V_{h,lay1} c_{pw} \rho_w \frac{dT_{c,lay1}}{dt} = \dot{m}_{solar} c_{pw} \Delta T_{lm,h,lay1} + \dot{m}_{heat} c_{pw} (T_{c,lay2} - T_{c,lay1}) + \dots \quad (31)$$

$$\dots U_{hotst} A_{h,lay1} (T_{Amb} - T_{c,lay1}) + U_w A_{c,lay} (T_{c,lay2} - T_{c,lay1})$$

$$V_{h,lay2} c_{pw} \rho_w \frac{dT_{h,lay2}}{dt} = \dot{m}_{boiler} c_{pw} (T_{boiler,out} - T_{h,lay2}) + \dot{m}_{heat} c_{pw} (T_{c,lay3} - T_{c,lay2}) + \dots \quad (32)$$

$$\dots U_{hotst} A_{h,lay2} (T_{Amb} - T_{h,lay2}) + U_w A_{h,lay} [(T_{h,lay3} - T_{c,lay2}) + (T_{c,lay1} - T_{c,lay2})]$$

$$V_{h,lay3} c_{pw} \rho_w \frac{dT_{h,lay3}}{dt} = \dot{m}_{heat} c_{pw} (T_{heat,out} - T_{h,lay3}) + U_{hotst} A_{h,lay3} (T_{Amb} - T_{h,lay3}) \dots \quad (33)$$

$$\dots U_w A_{h,lay} (T_{h,lay2} - T_{h,lay3})$$

$$\dot{m}_{solar} c_{pw} (T_{solcol,out} - T_{solcol,in}) = U_{stsolcol,ex} A_{stsolcol,ex} \Delta T_{lm,h,lay1} \quad (34)$$

$$T_{hotst,out} = T_{h,lay3}; T_{heat,in} = T_{h,lay1} \quad (35)$$

Table 3: Parameters of the differential-algebraic system

$A_{coll}$	40	$m^2$	$U_{hotst}$	0.6	$W/(m^2 K)$	$A_{cond,ex}$	10	$m^2$ [15]
$A_{drycool}$	40	$m^2$	$U_{solcol,ex}$	200	$W/(m^2 K)$	$A_{ev,ex}$	10	$m^2$ [15]
$A_{fp}$	90	$m^2$	$U_w$	10	$W/(m^2 K)$	$A_{z,ex}$	40	$m^2$ [15]
$A_{fpamb}$	40	$m^2$	$V_{c,lay1/2/3}$	0.3	$m^3$	$c_z$	950	$J/(kg K)$
$A_{ph,ex}$	10	$m^2$	$V_{h,lay1/2/3}$	0.8	$m^3$	$M_{cond,ex}$	5	kg [15]
$A_{c,lay1/2/3}$	1	$m^2$	$\varepsilon$	0.9		$M_{ev,ex}$	5	kg [25]
$A_{h,lay1/2/3}$	1	$m^2$	$\dot{m}_{air,day}$	0.3	kg/s	$M_{z,ex}$	5	kg (assumed)
$A_{c,lay}$	0.1	$m^2$	$\dot{m}_{boiler}$	1	kg/s	$M_z$	30	kg [9]
$A_{h,lay}$	0.5	$m^2$	$\dot{m}_{heat}$	2	kg/s	$U_{cond,ex}$	600	$W/(m^2 K)$ [26]
$A_{solcol,ex}$	10	$m^2$	$\dot{m}_{ph}$	0.1	kg/s	$U_{ev,ex}$	600	$W/(m^2 K)$ (assumed)
$BP$	0.3		$\dot{m}_{ref}$	1.5	kg/s	$U_{z,ex}$	50	$W/(m^2 K)$ [15]
$M_{fp}$	900	kg	$\dot{m}_{solar}$	10	kg/s	$\dot{m}_{z,ex}$	0.43	kg/s
$P_{boiler}$	24	kW	$(\tau a)_{eff}$	0.8		$\dot{m}_{cond,ex}$	2	kg/s
$RF$	0.9					$\dot{m}_{ev,ex}$	1	kg/s
$U_{drycool}$	80	$W/(m^2 K)$						
$U_{fp}$	500	$W/(m^2 K)$	$T_{inf}$	16	$^{\circ}C$			
$U_{fp,amb}$	1.5	$W/(m^2 K)$	$T_{sup}$	19	$^{\circ}C$	$t_{end,1/3}$	150	sec
$U_{ph,ex}$	80	$W/(m^2 K)$	$T_{set boiler,MAX}$	80	$^{\circ}C$	$t_{end,2/4}$	600	sec
$U_{coldst}$	0.6	$W/(m^2 K)$	$T_{set boiler,min}$	50	$^{\circ}C$	$\Delta T_{inf/sup}$	20	$^{\circ}C$

2

3 **6.Evaluation of the performace**

4 The chiller's cooling power is calculated by:

5 
$$Q_{cooling} = \dot{m}_{cool} c_{pw} (T_{ev,in} - T_{ev,out}) \quad (36)$$

6 The heating power in the packed beds exchangers is:

7 
$$Q_{heat} = \dot{m}_{heat} c_{pw} (T_{heat,out} - T_{heat,in}) \quad (37)$$

8 The condensing power is expressed as:

9 
$$Q_{cond} = (\dot{m}_{cond,ex} + \dot{m}_{z,ex}) c_{pw} (T_{dry,out} - T_{dry,in}) \quad (38)$$

10 The COP and the SPC are:

11 
$$COP = \frac{Q_{cooling}}{Q_{heat}} \quad (39)$$

12 
$$SCP = \frac{Q_{cooling}}{2M_z} \quad (40)$$

13 For the boiler, the actual thermal power of the heat generator is calculated as:

14 
$$Q_{boiler} = (\dot{m}_{boiler} + \dot{m}_{ph}) c_{pw} (T_{boiler,out} - T_{boiler,in}) \quad (41)$$

15 The natural gas consumption in the boiler during the simulation time is:

16 
$$V_{NG} = \int_{0s}^{end\ time\ sim.} \frac{Q_{boiler}}{Q_{cal,NG}} dt \quad (42)$$

17 where  $V_{NG}$  are the cubic meters of natural gas consumption in standard conditions and  $Q_{cal,NG}$  is the natural gas calorific value, assumed to be 39835.4 kJ/m<sup>3</sup>.

19 The total incident radiation is:

20 
$$I_{inc} = I_{tilt} A_{coll} \quad (43)$$

21 The heating power absorbed by the solar field is:

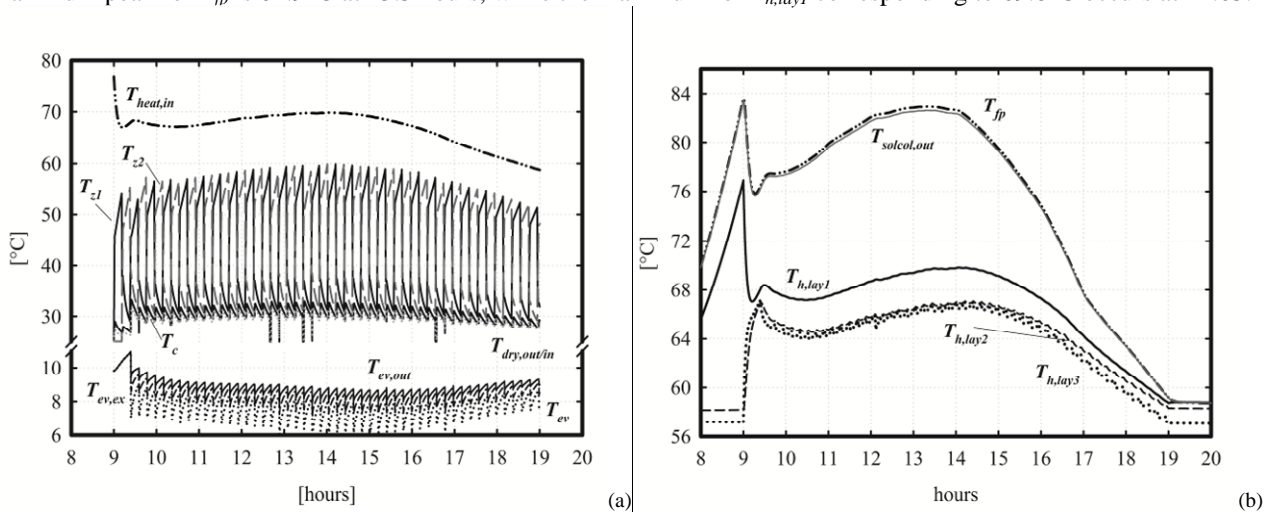
22 
$$I_{abs} = \dot{m}_{solar} c_{pw} (T_{solcol,out} - T_{solcol,in}) \quad (44)$$

23

24 **7.Results**25 **7.1.Time-dependent patterns**26 The main results are presented below studying the transients of several quantities on the second day simulated. Only the  
27 second day was considered because the results obtained on the first are usually influenced by the initial conditions  
28 adopted. The patterns of the main chiller variables are quite similar to those previously reported in the literature [19,  
29 20]. The novel aspect of present work consists in the ability to follow the adsorption machine end phases relating to the  
30 bed thermodynamics, which were not considered in previous transient models covering some consecutive days. Figure 5

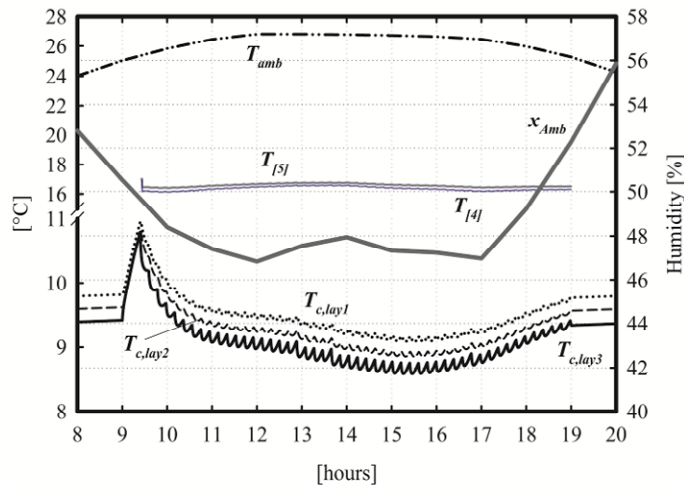


1 shows the effect of the variable radiation on the trend of the maximum bed temperatures in each cycle. The values of  $T_z$   
 2 for beds 1 and 2 ( $T_{z1}$ ,  $T_{z2}$ ) vary from 29.2°C to 59.8°C. The temperature  $T_{heat,in}$  at the inlet to the heat exchangers of the  
 3 adsorbent beds varies within a range of approximately 18°C throughout the day. The temperatures at the inlet and outlet  
 4 of the chiller's cooling circuit ( $T_{ev}$ ,  $T_{ev,ex}$ ,  $T_{ev,out}$ ) follow the same qualitative trends but with an average temperature  
 5 difference of approximately 1°C. For the condenser circuit, the temperatures  $T_{dry,in}$  and  $T_{dry,out}$  are very similar, differing  
 6 from  $T_c$  by about 0.5°C.  
 7 Looking at the maximum peaks of  $T_{heat,in}$ ,  $T_{z1}$ ,  $T_{z2}$  we see that the maximum value for  $T_{heat,in}$  of 69.8°C occurs at 14.0  
 8 hours, while the peaks for  $T_{z1}$  and  $T_{z2}$  are quite similar and amount to 59.7°C, obtained at 14.2 hours and 14.0, revealing  
 9 a delay of 12 min for bed 1. The minimum value of  $T_{ev,out}$  was 8.0°C. Considering a single cycle of one of the beds, the  
 10 change in uptake is of the order of 4%, due to the low regeneration temperatures. Similar result was found in [21, 22].  
 11 Figure 5 shows also the trends of the temperatures of the solar energy collection sub-system. As concerns the hot  
 12 storage unit, the temperatures of the three layers follow the same trends. The hot storage unit's average temperature is  
 13 64.4°C, with a maximum of 76.9°C in layer 1 and a minimum of 57.1°C in layer 3. The metal plate inside the solar  
 14 collector and the carrier fluid leaving the collector have very similar temperatures. Neglecting the first hours trend, the  
 15 maximum peak for  $T_{fp}$  is 82.9°C at 13.3 hours, while the maximum for  $T_{h,lay1}$  corresponding to 69.8°C occurs at 14.03.



16  
 17 Figure 5. (a) Temperatures relating to the chiller's operation and (b) temperatures relating to the solar energy collection  
 18 sub-system  
 19

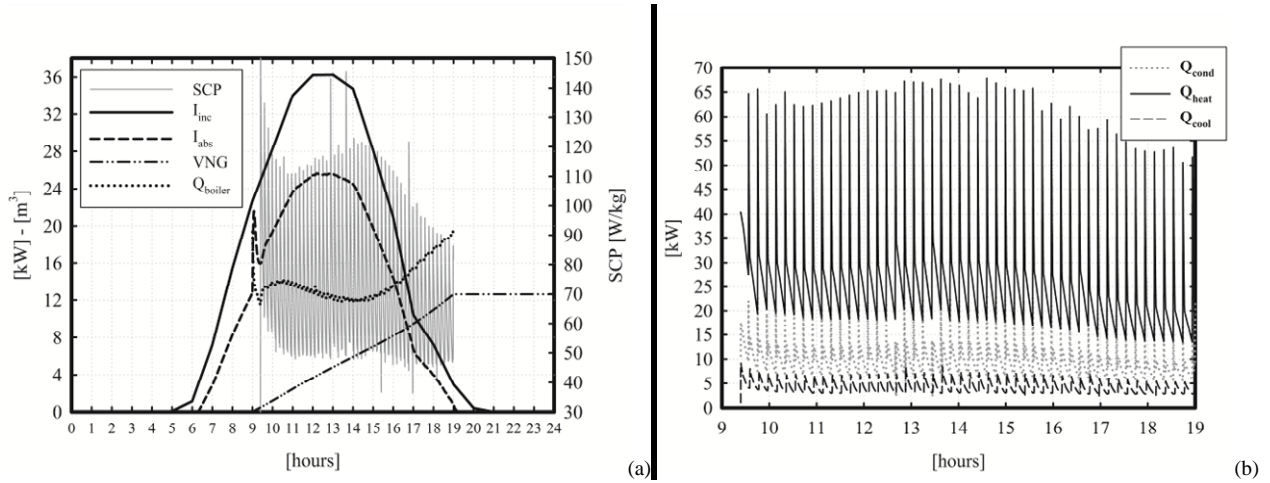
20 Figure 6 shows the trend of the ambient temperature (between 24°C and 27°C during the system's operation), of the air  
 21 temperatures after adiabatic mixing  $T_{[4]}$  and at the user  $T_{[5]}$  and of the temperatures of the three layers in the cold  
 22 storage unit. The trend of the ambient relative humidity is also shown. The temperatures of the three layers of the cold  
 23 storage unit vary between 8.6°C and 10.9°C, with negligible differences (below 0.5°C) among layers. The treated air  
 24 temperature  $T_{[5]}$  ranged between 16.1°C and 17.2°C.



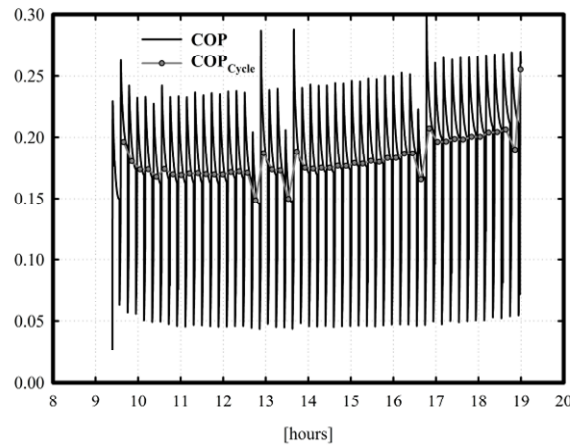
25  
 26 Figure 6. Temperatures and relative humidity relating to the air treatment unit  
 27

28 Figure 7 shows that the peak incident radiation is 36.2 kW, while the peak absorbed power is 25.7 kW. The boiler's  
 29 maximum capacity is 19.4 kW and its daily consumption of natural gas is 12.6 Nm<sup>3</sup>. The SCP, disregarding the start-up  
 30 transient, averages 69.3 W/kg. The cooling power  $Q_{cool}$  is stable and acquires a mean value of 4.1 kW. The heating

1 power of the two adsorbent beds  $Q_{heat}$  has a mean value of approximately 24.7 kW and the condensing power has a  
 2 mean value of around 9.2 kW.



3  
 4 Figure 7. (a) the power and performance of the main subsystems in the plant (b) powers relating to the adsorption  
 5 chiller's operation



6  
 7 Figure 8. instantaneous  $COP$  and  $COP$  of each cycle (calculated over the cycle time) of the adsorptive chiller  
 8

9 Figure 8 shows the instantaneous  $COP$  and the  $COP$  of every cycles. The daily average is 0.18, changing  
 10 instantaneously between 0.04 and 0.3. The  $COP$  of the single cycle at the time of starting up the plant is just below 0.3,  
 11 then drops and settles for most of the period of operation at around 0.2, ranging between 0.15 and 0.25. In the final  
 12 hours of operation of the chiller, there is a rising trend of the  $COP$  because the cooling power remains at the same levels  
 13 as during the previous hours, but the heat coming from the heating section of the system decreases as the radiation  
 14 decreases. The results from this base case are consistent with the literature in the  $COP$  [9, 22-24] but the plant  
 15 efficiency is not optimized. Therefore it is necessary a further analysis to identify a reduced number of operational  
 16 parameters which are most influencing the performance of the whole solar cooling system and each its subsystem.  
 17

## 18 7.2.Spectral analysis

19 The study of the variables transient can be simplified regressing them with Fourier series. Such regression demands a  
 20 preliminary understanding of the main frequencies characterizing the variable being investigated. This approach is  
 21 commonly applied to signal analysis, but can prove useful for analyzing the results of the present model too. Using the  
 22 Fast Fourier Transform of the variables enables us to know the most important frequencies comprising each signal.  
 23 Figure 9 shows the normalized periodograms of the most important temperatures, powers and climate data  
 24 characterizing the adsorption chiller's operation and the  $COP$ . It is evident that the transient of these quantities is  
 25 described by a limited number of fundamental frequencies. For some variables and for the input climate data the  
 26 important frequencies are lying below 1 cycle/hour, even though  $T_{c,lay3}$  has a negligible component around 5 cycles/hour  
 27 as visible also in figure 6.

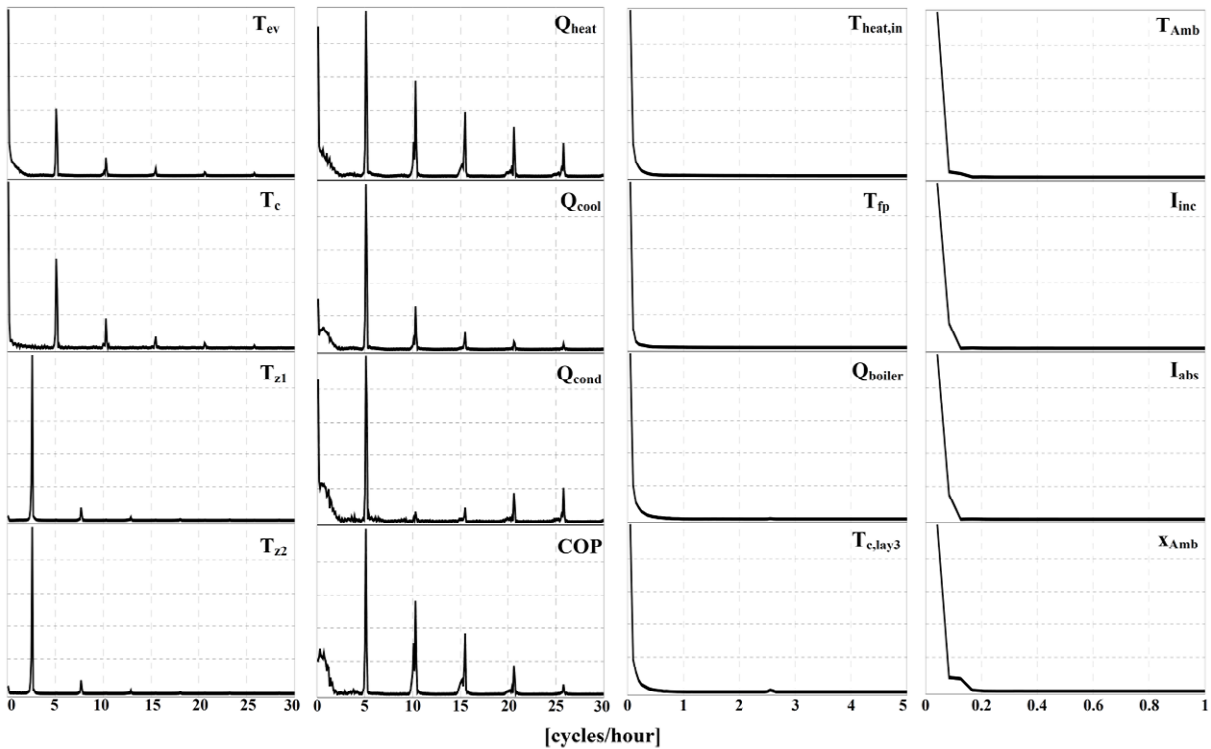


Figure 9. Periodograms of some variables involved in the the solar cooling system operation

The correlation between the signals is present when the peaks have the same or multiple frequencies. The frequency-based analysis also enables us to see which frequencies are most important for reconstructing the signal. In particular, the climate data and the variables relating to the subsystems other than the adsorption chiller are dominated by low frequencies, whereas the variables relating to the adsorption chiller's operation also involve high frequencies. So uncoupling the low-frequencies bands (typical of climate data) composing a signal and the high-frequencies bands composing the same signal (typical of the logic involved in controlling the solar cooling system and the adsorption chiller), it is possible to highlight the correlation between the signals.

### 7.3. Cross-Correlations

Performing the discrete Fourier cosine transforms of the chiller operating variables and considering only the first 20 modes enables the high-frequency trends to be omitted, so that we can concentrate on the lower frequencies alone. Figure 10 shows the regressions of several chiller operating variables, considering only the first 20 modes.

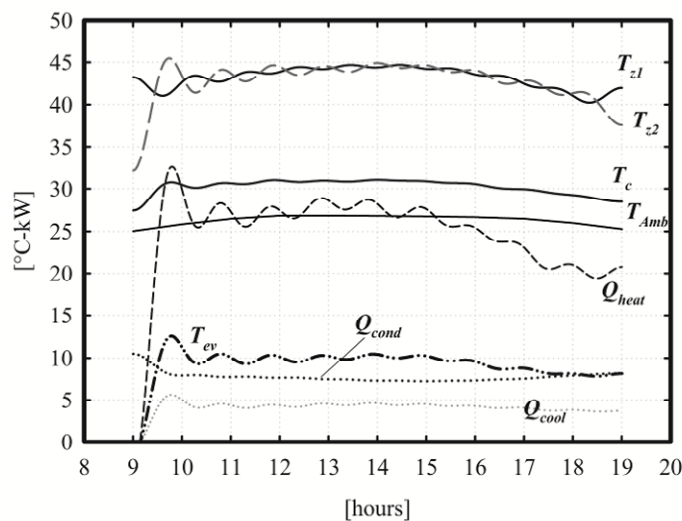


Figure 10. Discrete cosine transforms of the chiller's operating variables at low frequencies (first 20 modes)

1 Figure 11 shows the cross-correlation between the most important operating parameters considering only the first 20  
 2 modes for the chiller operating variables, and the original signal for  $T_{Amb}$  and the sum of  $Radiation+Q_{boiler}$  (the thermal  
 3 power released to the hot storage unit). A study of the cross-correlations between the signals clearly shows any  
 4 correlations, even if they have a phase shift. Cross-correlation also enables the correlations to be arranged from the most  
 5 to the least important and lets us establish whether the correlation is direct or inverse.  
 6 Cross-correlating the  $COP$  with the other chiller operating variables enables us to see which variables have the greatest  
 7 influence on the  $COP$ , and how much they influence the latter. Applying the cross-correlation to the climate data  
 8 enables us to see which variables are affected by variations in climate data, and to what degree they are affected. In the  
 9 following analysis, only the values outside the range of  $\pm 0.500$  were considered significant. The cross-correlations  
 10 identified the following influences (listed in order of importance and as absolute values).  
 11 1)  $COP=f(T_{ev}, Q_{cond}, Q_{cool}, Q_{heat})$  with  $T_{ev} > Q_{cond} > Q_{cool} > Q_{heat}$ . This means that when the  $COP$  increases, then  $T_{ev}$  decreases,  
 12 while  $Q_{cool}$ ,  $Q_{cond}$  and  $Q_{heat}$  increase. To maximize the  $COP$ , we therefore need to take the following steps, in order of  
 13 priority: i) to always ensure an accurate condensation; ii) to pay attention to the design of the evaporator and iii) to use  
 14 high flow rates and heat exchange surface areas for the adsorbent beds.  
 15 2) The thermal power delivered to the hot storage unit ( $Radiation+Q_{boiler}$ ) is correlated with  $(T_{z1}, T_c, T_{ev}, Q_{heat})$ , where  
 16  $T_{z1} > T_c > T_{ev} > Q_{heat}$ , but  $T_{ev}$  and  $Q_{heat}$  have cross-correlation coefficients below 0.6, so the correlation is less important.  $T_{z1}$   
 17 and  $T_{ev}$  have a phase shift of 53 minutes and 2.85 hours, respectively, due to inertias in the solar cooling system.  $T_c$  is  
 18 influenced by  $Radiation+Q_{boiler}$  because the radiation correlates strongly with the ambient temperature.  
 19

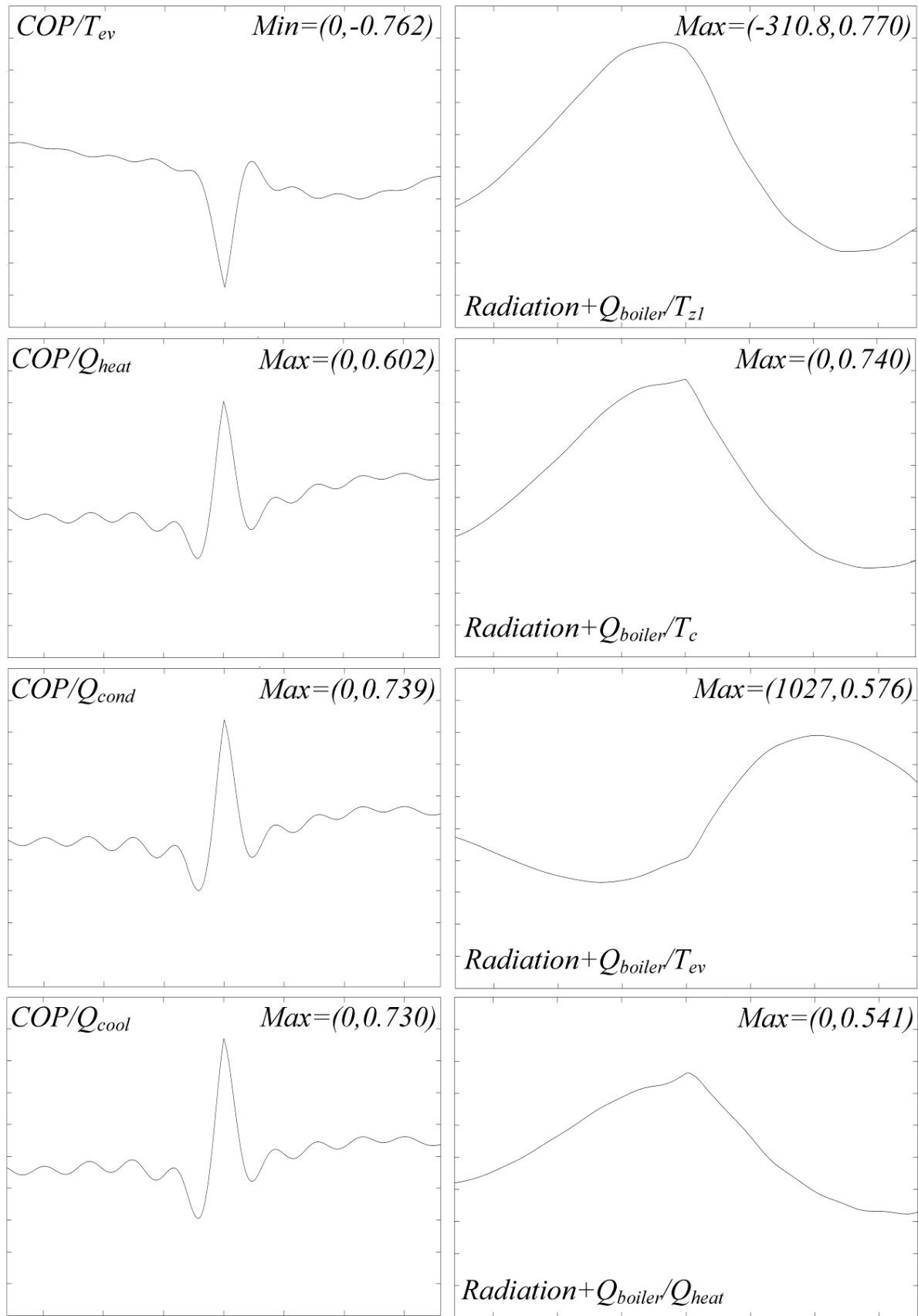


Figure 11. Selected cross-correlations of the operating variables for the solar cooling system

1  
2  
3  
4  
5

## 8. Conclusions

A dynamic multi-level mathematical model was developed to study the operation of a two-bed silica gel-water adsorption chiller for solar cooling applications. Variable temperature, humidity and solar radiation conditions were taken into account. The model provides accurate results for use in the further optimization and investigation of the system as a whole, the sub-systems and the chiller. The operating transients of the system can be tracked from start-up to shut-down on any given day of operation. The model was verified on a baseline case and the results were in the form of arrays of time-dependent values.

The obtained results are consistent with the previous models in literature. The results show that the solar cooling system can be further optimized, but a preliminary identification of the correlations among the variables influencing the performance of the solar cooling system and its subsystems have to be performed. The cross-correlations of the variables allow to select a reduced numbers of them for the optimization. The main result from the model is the obtained classification of importance among the variables in the transients. Hence the model enables a sensitivity analysis on the most influential parameters for some interesting outputs. Conclusions can be drawn on the basis of these results on the most appropriate control strategy and on the response of the adsorption chiller over a lengthy operational period. A spectral analysis was proposed on these data for identification of the correlation among the variables involved in the solar cooling system in order to study the effects of the input parameters on the outputs. Concentrating the study just a limited number of variables describing the chiller operation, the outcomes highlight that the *COP* depends mainly on  $T_{ev} > Q_{cond} > Q_{cool} > Q_{heat}$  and the thermal power  $Radiation + Q_{boiler}$  influences  $T_{z1} > T_c > T_{ev} > Q_{heat}$ .

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## Appendix

- The adsorbent/adsorbate equilibrium was calculated as:

$$\ln p_z = A_z(w) + \frac{B_z(w)}{T_z} \quad (\text{A.1})$$

where  $A_z(w)$  and  $B_z(w)$  are polynomials with coefficients determined experimentally at the CNR-ITAE laboratories ( $a_0 = 12.17$ ;  $a_1 = 1.495$ ;  $a_2 = -0.07295$ ;  $a_3 = 0.00107$ ;  $b_0 = -4178$ ;  $b_1 = -312.3$ ;  $b_2 = 16.78$ ;  $b_3 = -0.2501$ )

- The adsorption/desorption enthalpy  $\Delta H_{ads}(w)$  was calculated as:

$$\Delta H_{ads}(w) = -B_z(w) \frac{R}{M_g} \quad (\text{A.2})$$

where  $R$  is the universal gas constant and  $M_g$  is the molecular mass of the water.

## Nomenclature:

Symbols:

$A_{coll}$	active surface of the solar field; $m^2$
$BP$	by-pass factor (range 0-1)
$COP$	coefficient of performance
$c_{pa}$	specific heat of air; $J/(kg \text{ K})$
$c_{pw}$	specific heat of water; $J/(kg \text{ K})$
$H$	enthalpy of the moisture, $J/kg$
$I_{tilt}$	solar radiation on collector with $\beta$ tilt angle ( $30^\circ$ ); $W/m^2$
$L_w$	heat of water evaporation; $J/kg$
$\dot{m}_{air,day/night}$	air mass flow circulating in the ATU with the system enabled during the day/night; $kg/s$
$\dot{m}_{boiler}$	water mass flow rate in the boiler; $kg/s$
$\dot{m}_{BP}$	bypassing air mass flow rate ( $\dot{m}_{BP} = BP \dot{m}_{air,day}$ ); $kg/s$
$\dot{m}_{cond}$	mass flow rate of the water circulating in the condenser of the adsorption chiller; $kg/s$
$\dot{m}_{cond,ex}$	mass flow rate of the water circulating in the dry cooler; $kg/s$
$\dot{m}_{ev,ex}$	mass flow rate of the water circulating in the refrigeration loop; $kg/s$
$\dot{m}_{heat}$	mass flow rate of the hot water delivered to the adsorption chiller; $kg/s$
$\dot{m}_{ph}$	mass flow rate of the water for post-heating exchanger upstream from the reg. valve; $kg/s$
$\dot{m}_{ph,in}$	mass flow rate of the water in the post-heating exchanger; $kg/s$
$\dot{m}_{ref}$	mass flow rate of the water circulating between the cold storage unit and ATU; $kg/s$
$\dot{m}_{solar}$	mass flow rate of the water in the solar loop; $kg/s$
$\dot{m}_{tr}$	mass flow rate of the air in contact with heat exchanger ( $\dot{m}_{tr} = (1-BP) \dot{m}_{air,day}$ ); $kg/s$
$\dot{m}_{z,ex}$	mass flow rate of the water in the packed bed heat exchanger for the cooling phases; $kg/s$
$P_{boiler}$	rated thermal power of the boiler; $kW$
$Q_{boiler}$	actual heating power generated by the boiler; $W$
$Q_{cooling}$	cooling power generated by the chiller; $W$
$Q_{heat}$	heating power used by the chiller; $W$
$Q_{cond}$	condensation power of the chiller; $W$
$r$	heat of water condensation, $J/kg$
$RF$	ambient thermal factor
$RH$	relative humidity, %
$SCP$	specific cooling power; $W/(kg \text{ of adsorbent})$
$T_{inf/sup}$	min./max. control temperature of the reg. valve for the post-heating exchanger, $^\circ C$
$T_{setboiler,min/max}$	min./max. set point temperature of the boiler, $^\circ C$
$U$	overall heat transfer coefficient, $W/(m^2 \text{ K})$
$U_{coldst}$	overall heat transfer coefficient between cold storage unit and environment, $W/(m^2 \text{ K})$

1	$U_{hotst}$	overall heat transfer coefficient between hot storage unit and environment, $W/(m^2 K)$
2	$U_w$	overall heat transfer coefficient between two layers in the storage units, $W/(m^2 K)$
3	$x$	specific humidity; $g/kg$
4	$w$	uptake; %
5		
6	Greek letters:	
7	$\rho_w$	water density, $kg/m^3$
8	$\Delta H$	adsorption enthalpy; $J/kg$
9	$\Delta T_{lm}$	logarithmic mean temperature difference; $^{\circ}C$
10	$\varepsilon$	efficiency of cooling and dehumidifying exchanger
11	$(\tau\alpha)_{eff}$	effective absorption transmissibility coefficient
12		
13	Subscripts:	
14	$ads$	adsorption (in $\Delta H_{ads}$ )
15	$Amb$	ambient
16	$byp,out$	outlet of the tee between the post-heating outlet and the byp. valve (in $T_{byp,out}$ )
17	$boiler,in/out$	boiler inlet/outlet (in $T_{boiler,in}, T_{boiler,out}$ )
18	$BP$	by-passed (in $\dot{m}_{BP}$ )
19	$c$	condensation (in $T_c$ )
20	$cond,ex$	of the condensation heat exchanger (in $T_{cond,ex}, M_{cond,ex}, A_{cond,ex}, U_{cond,ex}$ )
21	$cond,in/out$	of the chiller's condensation loop inlet/outlet (in $T_{cond,in}, T_{cond,out}$ )
22	$c,lay$	between the layers in the cold storage unit (in $A_{c,lay}$ )
23	$c,lay1/2/3$	layer 1/2/3 of the cold storage unit (in $T_{c,lay1}, A_{c,lay1}, V_{c,lay1}, T_{c,lay2}, A_{c,lay2}, V_{c,lay2}, T_{c,lay3}, A_{c,lay3}, V_{c,lay3}$ )
24	$coldst,in/out$	cold storage unit inlet/outlet (in $T_{coldst,in}, T_{coldst,out}$ )
25	$coldst,pc$	in the post-cooling state (in $T_{coldst,pc}$ )
26	$dry$	dry cooler (in $U_{dry}, A_{dry}$ )
27	$dry,in/out$	condenser inlet/outlet (in $T_{dry,in}, T_{dry,out}$ )
28	$ev$	evaporation (in $T_{ev}$ )
29	$ev,ex$	of the evaporation heat exchanger (in $T_{ev,ex}, M_{ev,ex}, A_{ev,ex}, U_{ev,ex}$ )
30	$ev,in/out$	of the chiller's refrigeration loop inlet/outlet (in $T_{ev,in}, T_{ev,out}$ )
31	$fp$	metal plate of the solar collectors (in $T_{fp}, A_{fp}, U_{fp}$ )
32	$heat,in/out$	bed exchanger inlet/outlet (in $T_{heat,in}, T_{heat,out}$ )
33	$h,lay$	between the layers of the hot storage unit (in $A_{h,lay}$ )
34	$h,lay1/2/3$	layer 1/2/3 of the hot storage unit (in $T_{h,lay1}, A_{h,lay1}, V_{h,lay1}, T_{h,lay2}, A_{h,lay2}, V_{h,lay2}, T_{h,lay3}, A_{h,lay3}, V_{h,lay3}$ )
35	$hotst,in/out$	hot storage unit inlet/outlet (in $T_{hotst,in}, T_{hotst,out}$ )
36	$lm,ph$	logarithmic mean in the post-heating exchanger (in $\Delta T_{lm,ph}$ )
37	$ph,ex$	post-heating exchanger (in $U_{ph,ex}, A_{ph,ex}$ )
38	$ph,in/out$	post-heating exchanger inlet/outlet (in $T_{ph,in}, T_{ph,out}$ )
39	$stsolcol,ex$	heat exchanger in the hot storage unit of the solar field (in $U_{stsolcol,ex}, A_{stsolcol,ex}$ )
40	$solcol,in/out$	inlet/outlet of the solar collector (in $T_{solcol,in}, T_{solcol,out}$ )
41	$tr$	treated (in $\dot{m}_{tr}$ )
42	$w$	water (in $\rho_w, U_w$ )
43	$z$	of the adsorbent material (in $T_z, M_z, c_z$ )
44	$z,ex$	of the heat exchanger in the adsorbent bed (in $c_{z,ex}, U_{z,ex}, A_{z,ex}$ )
45	[1]	ambient air intake condition (in $T_{[1]}, x_{[1]}$ )
46	[2]	air in the cooling battery with no dehumidification (in $T_{[2]}, x_{[2]}$ )
47	[R]	dew point condition (in $T_{[R]}, x_{[R]}$ )
48	[3]	treated air downstream from the cooling and dehumidification (in $T_{[3]}, x_{[3]}$ )
49	[4]	condition of air after adiabatic mixing (in $T_{[4]}, x_{[4]}$ )
50	[5]	condition of air flowing to the user (in $T_{[5]}, x_{[5]}$ )