Modelling and dynamic simulation of a hybrid liquid desiccant system regenerated with solar energy.

Adriana Coca-Ortegón, Juan Prieto and Alberto Coronas[1]

[1] *Univertisat Rovira i Virgili, CREVER, Department of Mechanical Engineering, Av. Països Catalans 26, CP 43007, Tarragona, Spain.*

Abstract

The combination of liquid desiccant systems with conventional vapour compression chillers, usually known as hybrid liquid desiccant systems (HLDS), is a promising alternative when temperature and humidity need to be controlled in air conditioning applications. One of the advantages of this technology is that different kinds of energy can be integrated, particularly low temperature solar thermal energy, which can reduce the electrical consumption of the system.

These kind of systems are typically analysed by discrete steady-state simulations, which show how the system behaves in design conditions. However, dynamic simulations allow to know the seasonal performance and can help to set an appropriate control strategy.

This paper presents a description of the modelling and the dynamic simulation of a HLDS by using TRNSYS -. Because of there are not standard components for the main components of a LDS, absorber and regenerador, an alternative method based on performance tables for modelling them has been developed. The simulation is carried out for Kuala Lumpur, a city with high humidity and ambient temperatures, where the air conditioning is required during all year. In addition, the control strategy is also defined for the simulated system. Finally a sensitivity analysis is performed for the analysed case.

Keywords: Dehumidification, air conditioning, hybrid liquid desiccant system, solar energy.

1 Introduction

The combination of liquid desiccant systems with conventional vapour compression chillers, usually known as hybrid liquid desiccant systems (HLDS), is a promising alternative when temperature and humidity need to be controlled in air conditioning applications [1]. In contrast to conventional vapour compression systems (VCS), the dehumidification process using desiccant materials does not need to cool the air under the dew point and subsequently reheat it. Desiccants, then, make the air conditioning process more efficient, especially when latent loads are high [2–4].

Liquid desiccants are generally more able than solid desiccants to attract moisture and more flexible. For example, the absorber and regenerator can be physically separated, it is easier to adapt to specific pumping conditions, and a liquid-liquid heat exchanger can be used to improve the efficiency of the system [4,5]. In addition, this technology can integrate different kinds of energy

[1] *Corresponding author. adriana.coca@estudiants.urv.cat, Tel.: +34 977 257887*

sources, particularly low temperature solar thermal energy [5–7], which may decrease the electrical consumption of the system.

Typically these systems are analysed by discrete steady-state simulations, which show how the system behaves in design conditions. One example of this kind of simulation is the study by Ahmed et al [3], who simulate the performance of a HLDS made up of a vapour absorption chiller (VAC) with a liquid desiccant subsystem (LDS), and in which H_2O/LiBr is used as a refrigerant/absorbent mixture . In this case a COP sensitivity analysis was carried out, which showed that in nominal conditions the COP of HLDS is about 50% higher than that of a conventional vapour compression chiller.

Dai et al [8] did a similar simulation and validated the results with experiments. The HLDS analysed in this case was made up of an LDS, an indirect evaporative cooler and a vapour compression chiller. The sensitivity was analysed in terms of the electric coefficient of performance (ECOP), the thermal coefficient of performance (TCOP) and the general coefficient of performance of the system (COP) as efficiency indicators. The results were also compared with the performance of a conventional vapour compression system.

Other researchers such as Tu et al [9] also made a similar sensitivity analysis but using a more complex finite difference-based mathematical model for the packed columns of the LDS. Moreover, the system was optimized by using the exhaust air for heat recovery. The results show the influence of some key variables such as the solution temperature, ambient temperature, ambient humidity and mass flow on the some performance variables (COP, cooling capacity and exergy capacity).

Yamaguchi et al. [10] carried out a simulation and experimental analysis of an HLDS, using the simulation tool Simulink. The system is a conventional heat pump whose evaporator and condenser are working at the same time as the absorber and the regenerator of the LDS respectively. They evaluated the sensitivity of the COP in different experimental conditions and the influence of key variables such as the humidity ratio, the isentropic efficiency of the compressor, and the efficiency of heat exchangers.

Despite the usefulness of these discrete steady-state studies, some research has concluded that the behaviour of liquid desiccant systems needs to be analysed in transient conditions, so more accurate transient simulation validated with experimental works are still expected [11,12].

Other studies have pointed out the importance of knowing the seasonal performance of the HLDS so that energy savings can be quantified more accurately and the feasibility of these systems determined, particularly for HLDS which uses solar energy in the regeneration process [13]. Dynamic simulations, which take into account the changing weather and load conditions, may help to understand better the HLDS performance in long term and also to set an appropriate control strategy.

As far as this kind of dynamic simulation is concerned, Liu et al. [14] analyse the annual performance of an HLDS with a spray dehumidifier using the Equation Engineers Solver (EES) and the Fchart method. The study compares the system's behaviour in summer and winter seasons, and the results show that it performs better in summer. In comparison with a conventional vapour compression system, the energy consumption of the HLDS was about 78% during the summer and 62% during the winter for the case analysed in Beijing.

Zhang et al [15] analysed the summer and winter performance of an HLDS consisting of a vapour compression heat pump to treat the sensible load and an LDS to treat the latent load. The system also uses the heat pump to pre-cool the liquid desiccant, and the exhaust air of the regeneration process to prevent the heat pump from frosting during the winter.

Nomenclature

A_{COL}	Total solar collector area [m²]
G	Solar irradiance [kJ/m²]
$Q_{Airtotal}$	Energy in the air conditioning process (dehumidification and treatment temperature) [kJ]
Q_{Boiler}	Energy required by the auxiliary boiler [kJ]
Q_{Deh}	Energy in the air dehumidification process [kJ]
Q_{Reg}	Regeneration energy LDS [kJ]
Q_{Solar}	Solar tank energy output [kJ]
Q_{Total}	Total energy used in the system [kJ]
$SCOP_{LDS}$	Seasonal coefficient of performance in the air dehumidification process.
$SCOP_{GLOBAL}$	Seasonal coefficient of performance in the air conditioning process (dehumidification and treatment temperature).
$SCOP_{GLOBALNR}$	Seasonal coefficient of performance in the air conditioning process without renewable energy.
$SEFI_{COL}$	Seasonal efficiency of the solar collectors
SSF	Seasonal solar contribution factor
$W_{Chiller}$	Electrical energy required by the chiller [kJ]
PECF	Primary energy conversion factor

Lee and Lam made a more complex analysis [16]: a dynamic simulation of an HLDS made up of a ground heat pump subsystem and the LDS. All the components were modelled and programmed to solve the system by iterations using the Newton Raphson method for the refrigerant system, and the Gauss Seidel method for the LDS. The advantage of this methodology is that the results are very accurate, but the disadvantage is that the simulation takes a long time (in this case it was completed in 9 days).

More recently special tools have been used to develop dynamic simulations of HLDS, but even so very little work has been carried out in this line. The work done by Crofoot [17] is one example of this kind of study: an existing HLDS regenerated using evacuated tube solar collectors, was simulated in TRNSYS software, and specific components for the LDS were programmed in Fortran. The results show the annual performance of the system and compare them with the measurements of an experimental facility but do not analyse the control strategy using the seasonal results.

This paper presents a dynamic simulation of an HLDS using TRNSYS . An alternative method based on performance tables is applied to model the LDS components (absorber and regenerator). In addition, the control strategy has been defined and the seasonal performance evaluated for a specific case with high internal latent loads and environment humidity.

2 Case study

2.1 System description

Öberg and Goswani [5] published an extensive review of the HLDS that use solar energy in the regeneration process and more recently it was complemented by Mei and Dai [1]. According to these studies, HLDS usually consists of a liquid desiccant subsystem (LDS) that treats the latent load and a cooling subsystem (CS) that handles the sensible load.

In the LDS, the regeneration process with solar energy is carried out directly by solar regenerators (open or closed) or indirectly by conventional solar collectors; the absorbers and the regenerator use technologies such as packed beds, internally cooled packed beds, falling film plates, falling film extruded plates, and falling film tubes; and the most commonly used liquid desiccant material recently has been LiCl.

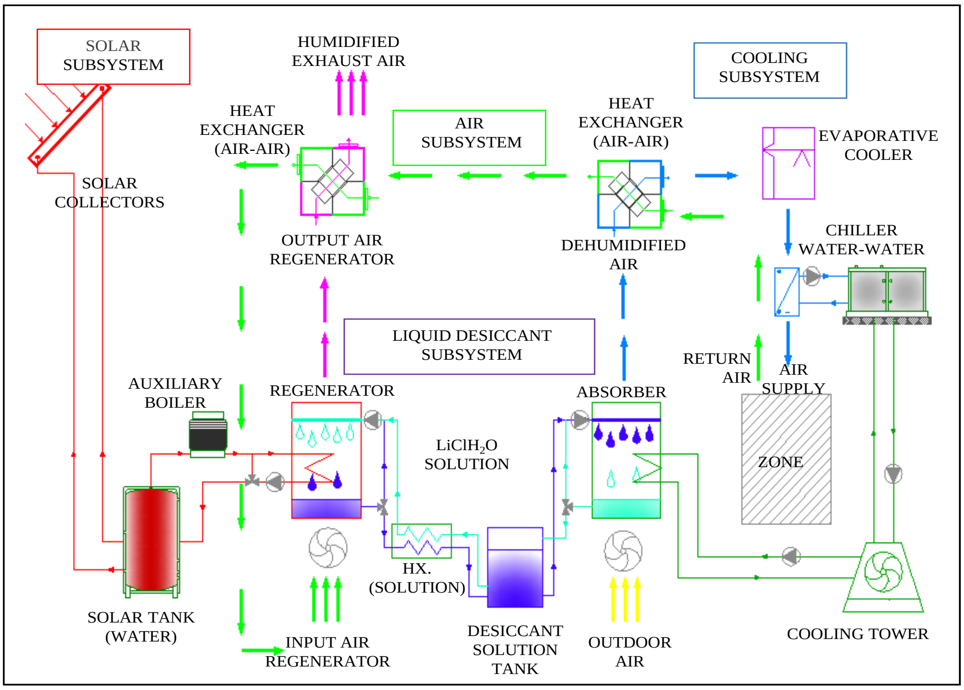

Fig.1. Hybrid liquid desiccant system (HLDS) for the case study

The CS can also incorporate such technologies as conventional vapour compression chillers, vapour absorption chillers, and evaporative coolers. The conditioning process can also be optimized using different configurations for heat recovery in the air subsystem (AS), as Das and Jain point out [18].

Taking these options into account, the proposed HLDS consists of four main subsystems: the liquid desiccant subsystem (LDS) with LiCl-H_2O solution, in which the supply air is dehumidified; the solar subsystem (SS) with conventional flat plate collectors, which supplies heat to the regeneration process; the cooling water subsystem, which handles the sensible load, with a vapour compression chiller; and the air subsystem with two air-to-air heat exchangers, an evaporative cooler and a cooling coil. The LDS consists of an absorber, a regenerator, a solution exchanger and such other elements as valves and pumps. Its operation is described in several previous publications [5,6,19].

The LDS is dimensioned at the design conditions (Table 1), this means that at different ambient and/or zone conditions, humidity after the absorber may be below the required supply humidity conditions. In order to reach the supply air humidity and decrease the air temperature an evaporative cooler is added after the absorber, as Öberg & Goswami explained [5], reviewing the works of Griffiths [20], Chebbah [21] and Gadhindasan [22]. After the evaporative cooler, a cooling coil, whose cold water is provided by a vapour compression chiller, makes the final adjustment of the air temperature. Finally the treated air is driven to the conditioning zone.

The return air of the conditioning zone is used as regeneration air, because it contains a lower humidity than the environment air. The temperature of the regeneration air is previously increased in the Air Subsystem (AS) through two air heat exchangers: one located after the absorber, and the other located after the regenerator.

The complete system is shown in Figure 1. The aim of this system is to use solar energy as the main resource, so the configuration incorporates the mentioned evaporative cooling to minimize the use of conventional energies and a cooling tower to provide cooling water to the chiller condenser and the LDS absorber. The system also has a storage water tank in the solar circuit and a desiccant tank in the LDS to optimize the seasonal solar factor. In addition, falling film tube has been used in the absorber and the regenerator because, according to previous studies, they are more efficient than other technologies such as packed bed contactors.

2.2 Load profile

The proposed HLDS was used in a case study, which consists of an air-conditioned zone in a typical fast food restaurant, characterized by a high latent load, in a tropical climate (Kuala Lumpur). As it can be noticed from Figure 2a, ambient temperature and relative humidity are high during all year, with average values above of 26.4ºC and 79.3% respectively. In addition this location has a good availability of solar resources, with average monthly global horizontal radiations between 3.91 and 4.63 kWh/m2-day (Figure 2b).

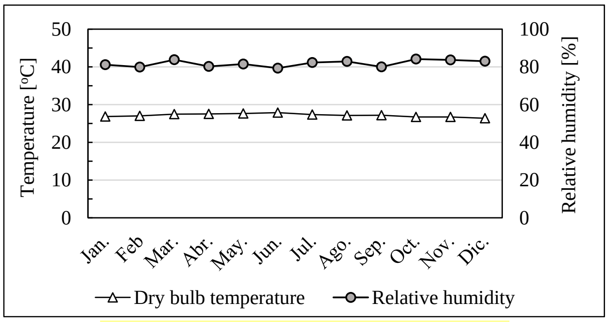

Fig. 2a. Monthly average dry bulb temperature and relative humidity in Kuala Lumpur

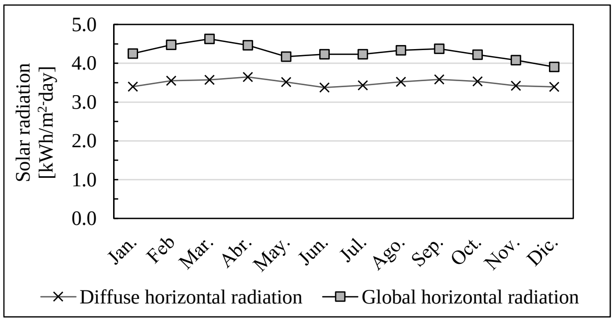

Fig. 2b. Monthly average global and diffuse horizontal solar radiation in Kuala Lumpur

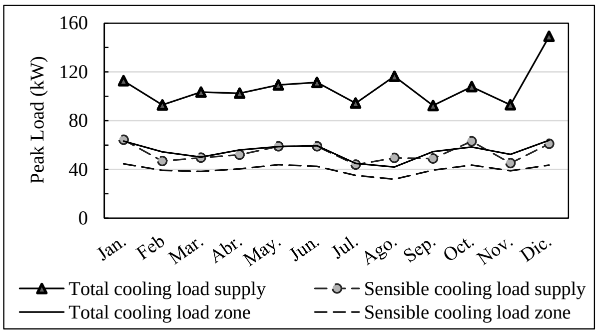

Fig. 2c. Monthly profile of peak cooling loads for the case study

At the design conditions (Table 1), the local internal loads are 20.29kW for the latent cooling load and 43.57 kW for the sensible cooling load. However, the high outdoor humidity increases the total latent and sensible load to 88.10kW and 60.97kW, respectively with a very low sensible heat ratio (SHR) of only 0.41. Moreover, as the profile in Figure 2c shows the cooling load is high throughout year, so the conditions are suitable for applying this kind of HLDS.

Table 1
Summary of design conditions

Description	Ambient conditions	Zone conditions	Supply conditions
Pressure [Pa]	99.77	99.77	99.77
Dry bulb temperature [ºC]	33.9	25.0	17.2
Relative humidity [%]	62.4	60.0	85.0
Humidity ratio [kg_w/kg_{da}]	0.02130	0.01208	0.01058

3 Simulation

3.1 Simulation methodology

The hybrid air conditioning system proposed was modelled on the dynamic simulation program TRNSYS v16. Most of the elements required in the HLDS are available in TRNSYS, except the absorber, the regenerator and the solution tank. Therefore, these elements have been implemented with alternative procedures, while the other elements of the solar, cooling and air subsystems, as well as the complementary elements of the LDS, have been modelled using standard components available in TRNSYS libraries. Table 2 summarizes the main types of component used in these subsystems.

Table 2
Summary of TRNSYS components applied in the model

Subsystem	Element	TRNSYS Component
LDS Subsystem	Absorber	Non-standard component
	Regenerator	Non-standard component
	Solution LiCl tank	Non-standard component
	Heat exchanger solution	Type 91
Solar Subsystem	Solar collectors	Type1b
	Auxiliary boiler	Type6
	Heat water storage	Type4a
Air Subsystem	Heat exchanger air to air	Type91
	Evaporative cooler	Type506c
	Cool battery	Equa
Cooling Subsystem	Chiller water-water	Type666
	Refrigeration tower	Type510
General elements for several subsystems	Pipes	Type31
	Pumps	Type114
	Valves (diverter and tee piece)	Type11f, Type 11h
	Differential control	Type2d
	Plotter online and printer	Type65d and Type25
Weather and loads	Weather, outdoor conditions.	Type15-3 (Energyplus)
	Load profile	Type 9c
	Supply conditions	Equa

The full model implemented in TRNSYS is illustrated in Figure 3, which shows all the subsystems (solar, cooling, air and LDS).

To model *non-standard elements* in TRNSYS, detailed component programming can be performed in Fortran, but this method is very time consuming because LiCl-H_2O properties must also be implemented. Another alternative is to use macros to connect with subroutines created in external applications such as Matlab and EES. However, in this case, numerous convergence problems often arise, especially for seasonal periods of simulation.

As an alternative to the above options, the absorber and regenerator have been implemented in TRNSYS by using the interpolation table model Type 581a from TESS libraries. That is, the *performance tables* for the absorber and regenerator have been developed and have been associated with the Type 581a component, which calculates the response of the element by interpolation and makes it easier to simulate these non-standard elements.

A key point in this alternative is to obtain the performance tables of the absorber and regenerator. The performance tables used in TRNSYS components are usually provided by manufacturers, although the liquid desiccant dehumidification systems (LDS) are still at an early stage of commercialization, and research centres are still developing this technology.

For this reason, these tables were calculated using the model presented by Hellman and Grossman [19] for the heat and mass exchanges with falling-film tube technology, which is based on the ε-NTU method and assumes that the water content and the enthalpy in the solution-air interface change linearly along the exchanger, so the energy and mass balance are calculated only at the input and the output to the absorber and regenerator, taking the heat and mass transfer coefficients from the experimental results obtained by Gommed et al [23].

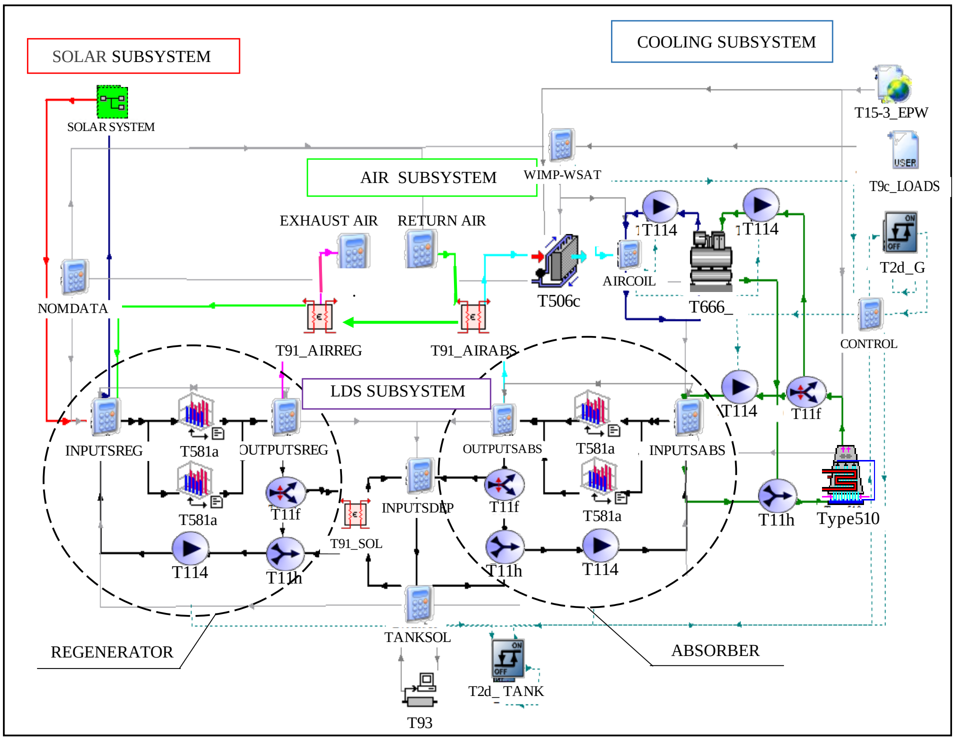

Fig. 3. Model of hybrid liquid desiccant system for air conditioning developed in TRNSYS

The procedure for developing these performance tables requires four main steps. The first step consists of identifying the inputs and outputs of elements to be modelled, but the Type 581a component used in the TRNSYS model, supports up to four inputs. Since the performance of the air-solution contactors depend on many variables, it is necessary to analyse which of them have a bigger effect at operation conditions. In addition the parameters whose values remain constant have also to be defined. Table 3 shows the parameters, inputs and outputs variables finally considered for the case study.

Table 3
List of inputs, outputs and parameters for the absorber and regenerator for the case study

Inputs		Outputs		Parameters	
Absorber					
$I1_A$	Inlet air temperature	$O1_A$	Outlet air temperature	$P1_A$	Inlet air volume flow
$I2_A$	Inlet air humidity ratio	$O2_A$	Outlet air humidity ratio	$P2_A$	Inlet solution mass flow
$I3_A$	Inlet solution concentration	$O3_A$	Outlet solution temperature	$P3_A$	Inlet water mass flow
$I4_A$	Inlet water temperature	$O4_A$	Outlet solution concentration		
		$O5_A$	Outlet solution mass flow		
		$O6_A$	Outlet water temperature		
Regenerator					
$I1_R$	Inlet air temperature	$O1_R$	Outlet air temperature	$P1_R$	Inlet air humidity ratio
$I2_R$	Inlet solution temperature	$O2_R$	Outlet air humidity ratio	$P2_R$	Inlet air volume flow
$I3_R$	Inlet solution concentration	$O3_R$	Outlet solution temperature	$P3_R$	Inlet solution mass flow
$I4_R$	Inlet water temperature	$O4_R$	Outlet solution concentration	$P4_R$	Inlet water mass flow
		$O5_R$	Outlet solution mass flow		
		$O6_R$	Outlet water temperature		

The inlet solution temperature at the absorber is calculated as a function of the inlet water temperature.

The second step is to define the operating range of the input variables of element to be modelled (absorber or regenerator). Some of these ranges depend on outdoor conditions, whose values of temperature, humidity and other variables change hourly during the annual period of simulation. To analyse them, it is useful to use the psychrometric chart of the location, which allows the maximum and minimum values for the outdoor variables to be defined.

Thus, in the case of the absorber, operating ranges of the inlet air temperature ($I1_A$) and inlet air humidity ($I2_A$) are already known because they correspond to the outdoor conditions, while the operating range of other variables such as inlet water temperature ($I4_A$) and inlet solution concentration ($I3_A$) can be set depending on the operational management of the installation.

For the regenerator, the procedure applied is the same as the one for the absorber. In this case particularly, the inlet air humidity is assumed constant, because the air return of the conditioning zone is used as regeneration air. The humidity ratio considered inside the zone is 0.01208 kg_w/kg_{da}. The inlet water temperature to regenerator varies between 60 and 70ºC because the solar regeneration subsystem with the auxiliary boiler is regulated to operate in this range of temperatures.

In the configuration analysed it is also important to see that the solution input conditions for the absorber and the regenerator are different because part of the solution output mass flow in these components is diverted to its input.

According to these criteria, the nominal values, limits, and ranges of operation adopted for the input variables and the parameters of the absorber and regenerator are presented in Table 4.

Table 4.
Nominal values and operating ranges of the parameters and input variables of the absorber and regenerator for the case study

	Description	Nominal value	Lower limit	Upper limit	Range	Number of intervals	Amplitude of intervals
Absorber							
-	*Inputs*						
$I1_A$	Inlet air temperature [ºC]	33.9	22.0	34.0	12.0	3	4.0
$I2_A$	Inlet air humidity [kg_w/kg_{da}]	0.0213	0.0140	0.0220	0.0080	2	0.0040
$I3_A$	Inlet solution concentration	0.3845	0.3500	0.4500	0.1000	4	0.0250
$I4_A$	Inlet water temperature [ºC]	27.7	25.0	35.0	10.0	2	5.0
-	*Parameters*						
$P1_A$	Inlet air volume flow [kg/s]	5.4	-	-	-	-	-
$P2_A$	Inlet solution mass flow[kg/s]	10.8	-	-	-	-	-
$P3_A$	Inlet water mass flow [kg/s]	13.5	-	-	-	-	-
Regenerator							
-	*Inputs*						
$I1_R$	Inlet air temperature [ºC]	45.4	42.0	50.0	8.0	2	4.0
$I2_R$	Inlet solution temperature [ºC]	57.8	53.3	63.8	10.5	4	2.6
$I3_R$	Inlet solution concentration	0.4006	0.3700	0.4570	0.0870	3	0.0290
$I4_R$	Inlet water temperature [ºC]	65.0	60.0	70.0	10.0	2	5.0
-	*Parameters*						
$P1_R$	Inlet air humidity ratio [kg_w/kg_{da}]	0.01208	-	-	-	-	-
$P2_R$	Inlet air volume flow [kg/s]	5.4	-	-	-	-	-
$P3_R$	Inlet solution mass flow [kg/s]	10.8	-	-	-	-	-
$P4_R$	Inlet water mass flow [kg/s]	13.5	-	-	-	-	-
Total of combination of the input variables at the absorber							180
Total of combination of the input variables at the regenerator							180

The third step is to develop the performance tables for the absorber and the regenerator, calculating the values of the output variables, for all possible combinations of the input variables. Table 4 also shows the number of intervals, the interval limits, and the total possible combinations of the input variables. Moreover, these tables have to consider the specific conditions of the case study and adapt them according to the requirements and formats of TRNSYS.

The fourth step to model the absorber and the regenerator is to associate the performance tables with one or more interpolation components 581a in TRNSYS, so these both air contactors can be dynamically simulated in this program.

Finally, the solution tank, which is also a non-standard component in TRNSYS, was modelled by using energy and mass balance equations and assuming a completed mixing solution inside. The total volume of the tank has been obtained by using the equation proposed by Kessling et al it [24]. The simulation time step has been reduced up to 0.01 hours in order to avoid as much as possible convergence problems produced by big changes in the solution properties inside the tank.

3.2 Control system

The control strategy is divided into the four main subsystems. For LDS operation, the absorber works only when there is a latent load, and the regenerator operates to maintain the LiCl mass concentration and the desiccant tank level between certain values.

When the conditioning zone is open, the absorber and the regenerator are activated through two different controls. The absorber starts to operate when there is latent load. LiCl mass fraction in the desiccant tank decreases then gradually. Once the lowest set value of LiCl mass fraction is reached the regenerator starts to operate increasing it progressively until reaching the highest limit. The highest limit has been one of the parameters modified in the sensitivity analysis. The obtained results are shown in section 4.

For implementing this control strategy, two operators were defined in the simulation, OPEABS for the absorber and OPEREG for the regenerator. OPEABS depends on the latent load and the OPEREG depends on the LiCl mass fraction in the desiccant tank, and desiccant tank level. To do this the TRNSYS model incorporated two differential controls (Type 2d) associated to these operators.

The final adjustment to the air humidity ratio is made at the air subsystem (AS) through the evaporative cooler, because its value after the absorber could be below the one required when latent loads or environment conditions are less demanding than the design conditions. Therefore the amount of air treated by the evaporative cooler must be regulated by a controller damper to ensure the desired output conditions.

In addition, the cooling subsystem (CS) adjusts the final air temperature to the required supply conditions, using a cooling coil whose chilled water is supplied by a vapour compression chiller, which operates at a constant water mass flow rate and a variable return temperature.

Finally, the control solar subsystem (SS) is also considered. In this case, the primary circuit is activated by a differential control, when the difference between the outlet temperature of the solar collectors and the return temperature of the water tank is at least 5 degrees. The secondary circuit is activated when regeneration is required, and a diverter valve is incorporated in order to keep the regeneration temperature below the maximum set value.

4 Results and discussion

In order to validate the model developed, two verification methods were performed: short term simulations to check that the control strategy is operating properly, and energy balances for each subsystem with long term simulations (yearly simulation) to confirm that the time step and tolerances are appropriate.

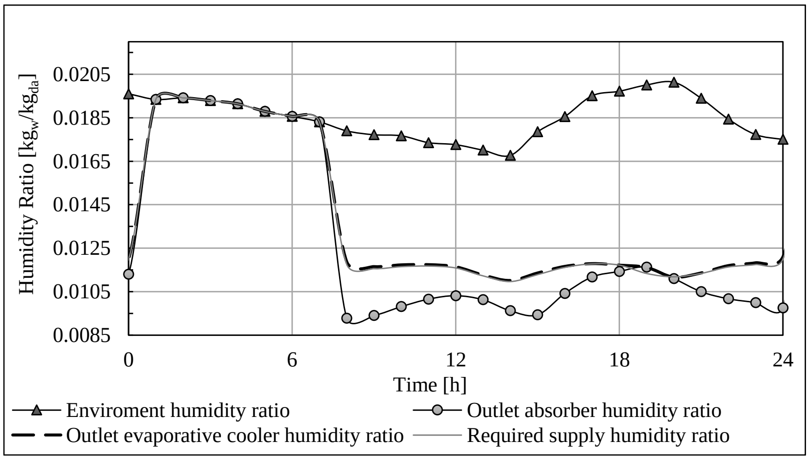

Fig. 4. Humidity control of the simulated HLDS for a typical day

The humidity control of the simulated system is shown for a typical day in Figure 4. It can be seen that the environment humidity ratio varies between 0.0163 and 0.0201 kg_w/kg_{da} during the day. When the conditioning zone is open and there is a latent load, the HLDS starts operating, so the air is dehumidified and the outlet absorber humidity ratio drops to values between 0.087 and 0.0116 kg_w/kg_{da}. These values are usually below those required for the supply humidity ratio so the evaporative cooler is regulated to reach the desired values. Only for the maximum outside humidity ratio there is a small difference of about 2.5% between the final supply humidity ratio and the required conditions.

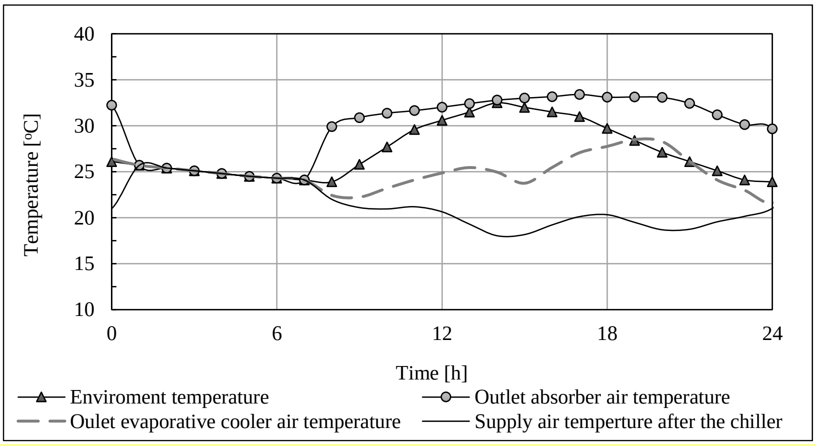

Fig. 5. Temperature control of the simulated HLDS for a typical day

The HLDS also has an adequate temperature control as Figure 5 shows for a typical day, in which the environment temperature varies between 23 and 32ºC. When the conditioning zone is open, the system starts to operate, and the air passes through the absorber, increasing the temperature up to values between 30 and 33ºC. Then there is heat transfer in the air to air heat exchanger and the temperature decreases; after this, the air passes through the evaporative cooler stage and the temperature decreases again up to values between 22 and 28ºC. Finally the cooling coil makes the final adjustment, to supply the required air temperatures, with values between 18 and 22ºC.

As it can be observed from Figures 4 and 5, the restaurant is closed from 00:00 hours to 7:00 hours, during this period the model assumes that the environment temperature and the environment humidity ratio are not modified by the system.

The regeneration process can be verified by the evolution of the main variables that control its operator (OPEREG): the LiCl mass fraction and the total mass inside the desiccant tank. Figure 6 shows the data for a typical day. According to these results, before the day starts, the LiCl mass fraction, H_2O mass and the LiCl solution mass are stable with values of 0.385, 536 kg and 834 kg respectively.

When the conditioning zone is open the regeneration process starts and the mass fraction gradually increases to the maximum value admitted (0.385) while the H_2O mass and the total LiCl solution mass decrease slowly to 477 and 776 kg. Because there is a latent load in the conditioning zone the absorber also operates, so after reaching the maximum mass fraction value, this variable decreases slowly to the minimum value admitted (0.35), while the H_2O mass and the total LiCl solution mass increase again to 553 and 852 kg. The cycle is repeated during the day until there is not dehumidification demand and throughout the operation the LiCl mass remains constant at a value of 298kg.

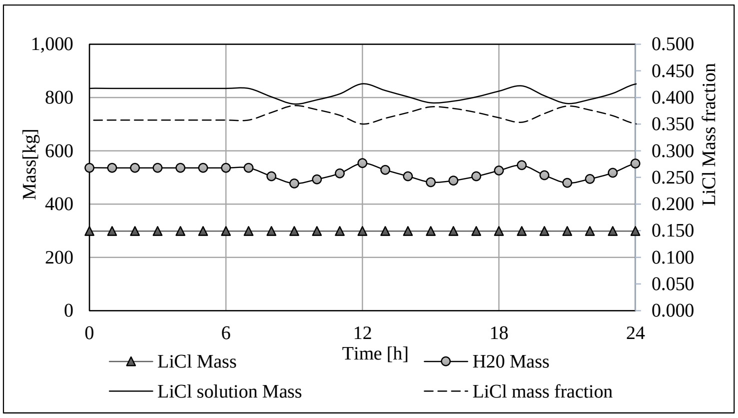

Fig. 6. Mass and LiCl mass fraction in the desiccant tank of the simulated HLDS for a typical day

Finally a seasonal sensitivity analysis of the size of the main components and the operational control variables was carried out. To do this, the efficiency of the system was evaluated on the basis of the seasonal coefficient of performance (SCOP). The calculation considered three coefficients: $SCOP_{LDS}$, $SCOP_{GLOBAL}$ and $SCOP_{GLOBALNR}$.

The $SCOP_{LDS}$ is evaluated by using Equation 1 to compare the energy in the dehumidification process between the entrance of the absorber and the output of the evaporative cooler (Q_{Deh}) with the energy used in the regeneration process (Q_{Reg}).

On the other hand, the $SCOP_{GLOBAL}$ is evaluated by comparing the total cooling energy provided to the supplied air. This includes both, the dehumidification and the temperature treatment of the supplied air from the entrance of the absorber to the output of the coiling coil ($Q_{Airtotal}$). This value is divided by the total energy used by the system (Q_{Total}) (see Equation 2).

The $SCOP_{GLOBALNR}$ is calculated in the same way as the $SCOP_{GLOBAL}$, but without including renewable energy (Q_{Solar}) as an expense in the process of regeneration (see Equation 3).

$$SCOP_{LDS} = Q_{Deh}/Q_{Reg} \quad (1)$$

$$SCOP_{GLOBAL} = Q_{Airtotal}/Q_{Total} \quad (2)$$

$$SCOP_{GLOBALNR} = Q_{Airtotal}/(Q_{Total} - Q_{Solar}) \quad (3)$$

where the energy of the regeneration process (Q_{Reg}) corresponds to all of the thermal energy required in the process of regeneration, calculated by adding the energy provided by the solar subsystem (Q_{Solar}) and the energy required by the auxiliary boiler (Q_{Boiler}) (see Equation 4).

The total energy (Q_{Total}) corresponds to the primary energy used in the system, which is calculated according to Dai et al [8], adding the thermal energy required in the system (Q_{Reg}) and the electrical energy required by the chiller ($W_{Chiller}$) divided by a primary energy conversion factor (PECF) that can be taken equal to equivalent to 0.3 (see Equation 5).

$$Q_{Reg} = Q_{Solar} + Q_{Boiler} \quad (4)$$

$$Q_{Total} = Q_{Reg} + W_{Chiller}/PECF \quad (5)$$

In addition the solar installation was evaluated by calculating the seasonal efficiency of the collectors ($SEFI_{COL}$) and the seasonal solar contribution factor (SSF) (see equations 6 and 7).

$$SEFI_{COL} = Q_{Solar}/(G \times A_{COL}) \quad (6)$$

$$SSF = Q_{Solar}/Q_{Reg} \quad (7)$$

The initial simulation was carried out at a regeneration temperature of 70ºC, a solar collector area of 887 m^2, eight hours of accumulation in the solar storage tank, two hours of accumulation in the desiccant tank, and a LiCl mass fraction between 0.350 and 0.414 The results obtained when the various parameters were modified are summarized in Table 5.

The variable with the biggest impact on the seasonal efficiency of the solar collectors ($SEFI_{COL}$) is the regeneration temperature and results are best at low temperatures. For the seasonal solar factor (SSF) results are best when the total solar collector area and the regeneration temperature are higher and when the LiCl mass fraction in the desiccant tank is lower.

The variable that has the most influence on to the $SCOP_{LDS}$ is the regeneration temperature. An increase of between 60 to 70ºC increases this indicator by about 11%. The $SCOP_{GLOBAL}$ and $SCOP_{GLOBALNR}$ are better when regeneration temperature is higher.

Table 5.
Summary of seasonal sensitivity results

Description		$SEFI_{COL}$	SSF	$SCOP_{LDS}$	$SCOP_{GLOBAL}$	$SCOP_{GLOBALNR}$
Sensitivity to regeneration temperature variation						
1.1	Regeneration temperature 60ºC	0.55	54%	0.55	0.61	1.14
1.2	Regeneration temperature 65ºC	0.53	52%	0.57	0.63	1.14
1.3	Regeneration temperature 70ºC	0.51	53%	0.62	0.67	1.23
Sensitivity to solar tank volume variation						
2.1	Solar tank 4h - Desiccant tank 2h	0.50	52%	0.62	0.67	1.22
2.2	Solar tank 6h - Desiccant tank 2h	0.51	53%	0.62	0.67	1.23
2.3	Solar tank 8h - Desiccant tank 2h	0.51	53%	0.62	0.67	1.23
Sensitivity to desiccant tank volume variation						
3.1	Solar tank 6h - Desiccant tank 1h	0.51	52%	0.61	0.66	1.20
3.2	Solar tank 6h - Desiccant tank 2h	0.51	53%	0.62	0.67	1.23
3.3	Solar tank 6h - Desiccant tank 4h	0.51	53%	0.63	0.68	1.26
Sensitivity to maximum LiCl mass fraction variation in the desiccant tank						
4.1	LiCl mass fraction 035 to 0.385	0.51	53%	0.62	0.74	1.46
4.2	LiCl mass fraction 0.35 to 0.414	0.51	47%	0.62	0.67	1.21
4.3	LiCl mass fraction 0.35 to 0.440	0.51	42%	0.61	0.61	1.02
Sensitivity to solar collector area variation						
5.1	Collector area base x 0.25	0.52	11%	0.62	0.66	0.73
5.2	Collector area base x 0.50	0.52	25%	0.62	0.67	0.84
5.3	Collector area base x 1.00	0.51	53%	0.62	0.67	1.23
5.4	Collector area base x 1.50	0.50	79%	0.62	0.68	2.19

The inlet cooling water temperature to the absorber is usually relevant for the liquid desiccant systems because the capability of retaining moisture of a liquid desiccant solution is higher at lower solution temperatures[4]. The proposed HLDS does not permit the control of this temperature because the cooling water is produced by a cooling tower, whose operation conditions depends on the environment temperature and humidity; however the advantage of this system is that the cooling tower minimize the use of electricity.

The LiCl mass fraction in the desiccant tank, has an important influence. When the maximum set-point of the LiCl mass fraction increases from 0.385 to 0.414 and 0.440, the demanded energy by regeneration process also increases about 11.3% and 24.5% respectively; that implies that more energy from the auxiliary boiler must be used in the system, and the $SCOP_{GLOBAL}$ decreases from 0.74 to 0.67 and 0.61, with variations of 9.0% and 21.7% respectively. Therefore the better $SCOP_{GLOBAL}$ and $SCOP_{GLOBALNR}$ are obtained when the system operates with a LiCl mass fraction lower than 0.385. In any case, even decreasing this maxim LiCl mass fraction until 0.385, a suitable supply humidity ratio is obtained during all the year.

Finally, increasing the solar collector surface decreases energy consumption and makes the seasonal solar factor (SSF) bigger. When the SSF is the highest, the obtained $SCOP_{GLOBALNR}$ reaches the maximum values; the analysed system has a $SCOP_{GLOBALNR}$ of 2.19 when the final solar contribution factor is 79%.

5 Conclusions

According to short-term and long-term simulation results the model implemented in TRNSYS for the described HLDS presents an adequate behaviour. Therefore, the proposed method for modelling the absorber and the regenerator based on the performance tables and the interpolation component behaves is a useful option,. Moreover, convergence problems have not been found and simulation time has been reasonable.

The variables that have the greatest impact on the system were identified through a sensitivity analysis, considering the seasonal performance through dynamic long-term simulations. Using these results the control strategy of the system can be optimized.

In the analysed case the most relevant variables are the regeneration temperature, the LiCl mass fraction in the desiccant tank, and the solar collector surface. The increase of the regeneration temperature from 60 to 70ºC improves the $SCOP_{LDS}$ about 11.0%. By reducing the maxim LiCl mass fraction in the desiccant tank from 0.440 to 0.385, the $SCOP_{GLOBAL}$ can be increased about 21.7%. Therefore, the control strategy can be optimized by rsetting the regeneration temperature at 70ºC and the LiCl mass fraction in the desiccant lower than 0.385.

On the other hand the best $SCOP_{GLOBALNR}$ is obtained with a solar collector surface of ** because the highest solar contribution factor is achieved.

Comentado [JP1]: Pon al valor de la superficie, no he sido capaz de encontrarlo en el paper.

Finally, it must be emphasized that, although this paper presents an alternative method to model the absorber and the regenerator for a specific type of HLDS, it can be applied to other configurations of the hybrid air conditioning systems with LDS for dehumidification.

References

[1] L. Mei, Y.J. Dai, A technical review on use of liquid-desiccant dehumidification for air-conditioning application, Renew. Sustain. Energy Rev. 12 (2008) 662–689. doi:10.1016/j.rser.2006.10.006.

[2] D.G. Waugaman, A. Kini, C.F. Kettleborough, A Review of Desiccant Cooling Systems, J. Energy Resour. Technol. 115 (1993) 1. doi:10.1115/1.2905965.

[3] C.S. Khalid Ahmed, P. Gandhidasan, a. a. Al-Farayedhi, Simulation of a hybrid liquid desiccant based air-conditioning system, Appl. Therm. Eng. 17 (1997) 125–134. doi:10.1016/S1359-4311(96)00025-7.

[4] ASHRAE, Sorbents and desiccants, in: Handb. Fundam. Chapter 32, American Society of Heating, Refrigerating & Air Conditioning Engineers, 2009: pp. 32.1–32.6.

[5] V. Öberg, D. Goswani, Advances in solar energy. Chapter 10: A review of liquid desiccant cooling, in: Adv. Sol. Energy, American Solar Energy Society Inc., Boulder (Colorado), 1998.

[6] M. Jaradat, M. Krause, R. Heinzen, L. Mesquita, Task 38: Solar air-conditioning and refrigeration. Section 4: Liquid desiccant sytems, 2010.

[7] H.-M. Henning, M. Motta, D. Mugnier, Solar cooling handbook. A guide to solar assisted cooling and dehumidification processes, 3rd ed., Ambra, Vienna, 2013.

[8] Y.J. Dai, R.Z. Wang, H.F. Zhang, J.D. Yu, Use of liquid desiccant cooling to improve the performance of vapor compression air conditioning, Appl. Therm. Eng. 21 (2001) 1185–1202. doi:10.1016/S1359-4311(01)00002-3.

[9] M. Tu, C.Q. Ren, L. a. Zhang, J.W. Shao, Simulation and analysis of a novel liquid desiccant air-conditioning system, Appl. Therm. Eng. 29 (2009) 2417–2425. doi:10.1016/j.applthermaleng.2008.12.006.

[10] S. Yamaguchi, J. Jeong, K. Saito, H. Miyauchi, M. Harada, Hybrid liquid desiccant air-conditioning system: Experiments and simulations, Appl. Therm. Eng. 31 (2011) 3741–3747. doi:10.1016/j.applthermaleng.2011.04.009.

[11] Y. Luo, H. Yang, L. Lu, R. Qi, A review of the mathematical models for predicting the heat and mass transfer process in the liquid desiccant dehumidifier, Renew. Sustain. Energy Rev. 31 (2014) 587–599. doi:10.1016/j.rser.2013.12.009.

[12] S. Jain, P.K. Bansal, Performance analysis of liquid desiccant dehumidification systems, Int. J. Refrig. 30 (2007) 861–872. doi:10.1016/j.ijrefrig.2006.11.013.

[13] J. Dean, E. Kozubal, L. Herrmann, J. Miller, A. Lowenstein, G. Barker, et al., Solar-powered, liquid-desiccant air conditioner for Low-electricity humidity Control, DOE, Department of Energy of U.S, 2012.

[14] X. Liu, Z. Li, Y. Jiang, B. Lin, Annual performance of liquid desiccant based independent humidity control HVAC system, Appl. Therm. Eng. (2006) 1198–1207.

[15] L. Zhang, C. Dang, E. Hihara, Performance analysis of a no-frost hybrid air conditioning system with integrated liquid desiccant dehumidification, Int. J. Refrig. 33 (2010) 116–124. doi:10.1016/j.ijrefrig.2009.08.007.

[16] C.K. Lee, H.N. Lam, Computer simulation of ground-coupled liquid desiccant air conditioner for sub-tropical regions, Int. J. Therm. Sci. 48 (2009) 2365–2374. doi:10.1016/j.ijthermalsci.2009.05.010.

[17] L. Crofoot, Experimental evaluation and modeling of a solar liquid desiccant air conditioner, Queen’s University, 2012.

[18] R.S. Das, S. Jain, Simulation of potential standalone liquid desiccant cooling cycles, Energy. 81 (2015) 1–10. doi:10.1016/j.energy.2015.01.009.

[19] H.M. Hellmann, G. Grossman, Investigation of an open-cycle dehumidifier-evaporator-regenerator (DER) absorption chiller for low-grade heat utilization, ASHRAE Trans. (1995) 1281–1289. doi:10.1016/0140-7007(95)90314-P.

[20] W. Griffiths, Solar energy assisted air-conditioning apparatus and method, 1979.

[21] A. Chebbah, Analysis and Design of a solar powered Liquid Desiccant air Conditioner for use in Hot and Humid Climate, University of Florida, Gainesville, FL, 1987.

[22] P. Gandhidasan, Performance analysis of an open-cycle liquid desiccant cooling system using solar energy for regeneration, Int. J. Refrig. 17 (1994) 475–480. doi:10.1016/0140-7007(94)90008-6.

[23] K. Gommed, G. Grossman, J. Prieto, J. Ortiga, A. Coronas, Experimental comparison between internally and externally cooled air-solution contactors, Sci. Technol. Built Environ. 21 (2015) 267–274. doi:10.1080/23744731.2015.1015381.

[24] W. Kessling, E. Laevemann, C. Kapfhammer, Energy storage for desiccant cooling systems component development, Sol. Energy. 64 (1998) 209–221. doi:10.1016/S0038-092X(98)00081-4.