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Journal of Architectural Environment & Structural Engineering
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ARTICLE

Feasibility of Desiccant Liquid Cooling (LDC) System under Mediterranean Climate

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ARTICLE INFO

Article history
Received: 6 December 2018
Accepted: 3 January 2019
Published:7 March 2019

Keywords:
Desiccant liquid
Cooling system
Climatic conditions
Operating conditions

ABSTRACT

Contrary to the traditional Heat Ventilation Air Conditioning (HVAC) systems, the Liquid-Desiccant-Cooling (LDC) systems are considered important systems to control indoor air conditions. In addition, the LDC technologies are more adequate for the hot and humid climates. In this paper, we present an analytical analysis to study feasibility of a LDC system under Mediterranean climate. In air streams and desiccant liquid, the mathematical equations including the sensible and latent heat transfer equations are presented. The climatic and operating parameters impacts on the supplied air qualities, moisture removal rate (MRR) and sensible heat ratio (SHR) are evaluated. As a consequence, this study provides a solution to investigate the feasibility of this type of air conditioning technologies under hot and humid climate.

1. Introduction

Desiccant liquid Cooling (LDC) systems are not only industrial and residential systems to offer a comfortable ambiance but also a good factor to reduce the electric consumption and to keep environment safe. In the recent years, a high portion of the primary energy (40-50%) is consumed by the residential sector which caused about 33% gases emission in the world [1]. Further, the traditional HVAC systems have environmental drawbacks such as global warming, air pollution, and acid precipitation [2]. Thus, LDC systems observe fast growth in the research and development moreover several studies are investigated various LDC systems performance which considered an efficient technology to control indoor air conditions. These systems operated by desiccant liquids which have a high affinity to absorb the moisture content in air with possibility to cool and store under no sunshine hours [3].

Min et al [4] introduced two novel configurations of desiccant liquid air conditioning system and investigated their performances. A comparison between these configurations with other desiccant liquid systems to improve indoor air qualities was carried. The simulation results proved that there are efficient under hot and humid climatic conditions.

In other study, Min et al [5] studied a lithium chloride (LiCl) desiccant liquid air conditioning system. A regenerator (dehumidifier) core to regenerate (to dehumidifier) the weak desiccant solution (the ambient air) are simulated. They concluded that the choice of the operating and climatic parameters has the significant impact on the pro-

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posed system performance.

Xiao et al [6,7] studied a LDC system which consists of dehumidifier/regenerator cores and total heat exchanger. To control the global system effectiveness, they proposed control strategies for the ambient air (desiccant solution) inside the dehumidifier (regenerator) core. The influences of total heat exchanger are also investigated. The simulation results concluded that this studied system is more adequate for the hot and humid climates and more stable using the total heat recovery which ameliorate the energy performance by 19.9% to 34.8%.

Various air conditioning systems including both adsorption-absorption cooling and desiccant cooling are researched by Wang et al [8]. A variety of significant parameters such as the operating mechanisms, energy performance, maintenance and economic feasibility are discussed. In addition, they offered a detailed analysis and working instructions of air conditioning systems for residential applications. They concluded that the sorption systems needed more research and improvement to reduce their energy consumption with low cost and good efficiency.

Gommed and Grosman [9, 10] tested a LDC system in hot and humid Mediterranean place which characterized by a high energy consumption for air conditioning sector in the summer days. Air to air heat exchanger is inserted to minimize the heat transfer losses in the LDC system and improve the global system efficiency. Under various operating conditions, they tested the LDC system which is very efficient in the Mediterranean climate. Ma et al [11] examined a hybrid LDC system and they concluded that LDC systems performance is higher than the conventional air conditioning systems by 44.5 %.

Further, numerous studies and experimental measurements carried out to investigate the LDC systems under various climatic and operating conditions to improve the system efficiency [12-26]. However, other studies are carried only to investigate one of principal parts of LDC systems such as the regeneration part or the dehumidification part [27-29].

Most importantly, the mainly studies should present an economic study of operating LDC system to obtain best rapport economic between price performance [30,31]. Thus, the analytical investigations such as the climatic, economic and operating systems play an important factor to install and to choose the specific and effectiveness LDC systems. Furthermore, the choice of a LDC system from the above air conditioning system categories is related by the operating and climatic conditions. Before installing and testing the selected LDC system under real climatic Mediterranean conditions like as Tunisian hot and humid climate, we aims to present an analytical study of the selected LDC system and to evaluate the influence of both operating and climatic conditions on the effectiveness of the proposed LDC system.

2. Description of the LDC System

In this part, Figure 1 presents a schematic description of the selected LDC system. The diverse numbers indicated the air and desiccant solution positions in the selected LDC system operation. This system operates with two cycles which one is for the desiccant solution and the other is for the fresh air. At first and after passing through the Solution-Solution Heat eXchanger (SHX) to be pre-cooled by the weak desiccant solution coming from the dehumidifier core, the desiccant liquid (solution) cooled by the cold water in the water-solution Heat Exchanger (HE2) then pumped to the bottom of dehumidifier. At this moment, the open air cycle begin by enter to the LDC system. After going through the air-air heat exchanger to be pre-dehumidified, the hot/humid air enter to the dehumidifier which makes contact with the strong cold desiccant solution where it is dehumidified and cooled. In other hand, the diluted (weak) desiccant solution passes through the SHX to be pre-heated and then it entered to the water-solution heat exchanger (HE1) where it attain the regeneration temperature. When the regeneration process is finished, the outlet concentrated (strong) desiccant solution begins another cycle.

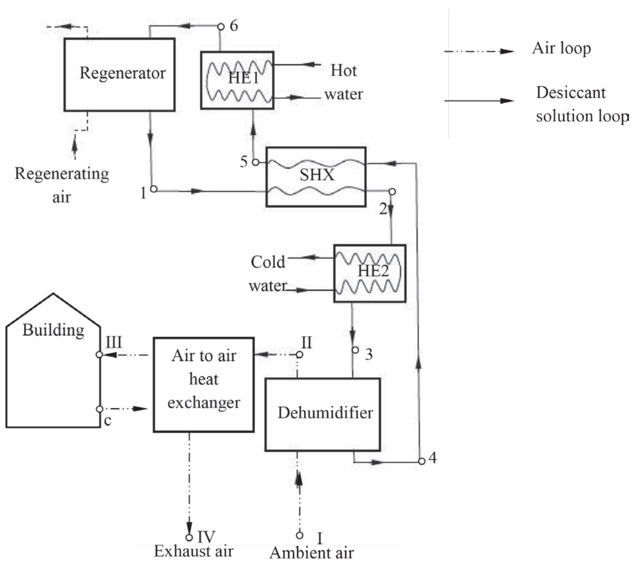


Figure 1. Schematic description of the selected Liquid Desiccant Cooling (LDC) system

3. Choice of Desiccant Liquid Solution

From the frequent investigations carried on the hygro-

  DOI: https://doi.org/10.30564/jaeser.v2i1.387

scopic substances, a large numbers of desiccant liquid solution which used under the various air conditioning systems are found. However, various criteria are taken account to choose the adequate desiccant liquid solution such as the regeneration temperature, crystallization limit, vapor pressure, and the price.

A comparison investigation is made between lithium chloride (LiCl) and lithium bromide (LiBr) solutions which are frequently used by LDC systems. The results prove that the LiCl solution is more efficient than the LiBr solution due to its best absorption ability, high crystallization limit, and less regeneration temperature (60-65°C) as well as it is not a toxic material, does not cause any health danger and lower cost [14].

Pietruschka et al [32] found in experimental analysis that the calcium chloride ($CaCl_2$) solution has a potential to dehumidifier the supply air low than the lithium chloride (LiCl) solution. Further, the LiCl solution offers 40-50% higher dehumidification rates over a wide relative humidity range and it has not causticity with metals.

An investigation carried out by Longo and Gasparella [33] to study the influence of three desiccant liquid solutions (LiCl, LiBr and KCOOH) in both dehumidifier and regenerator cores. Experimental results are shown that the KCOOH solution has a lower dehumidification performance than LiCl and $CaCl_2$ solutions.

A collected works including interpolating equations for the principal proprieties of aqueous LiCl, $CaCl_2$solutions is presented by Conde [34]. These studied proprieties are essential in the design of LDC air conditioning systems that are based on sorption processes.

We can found that from a variety of desiccant liquid solutions, the lithium chloride solution (LiCl) is the most solution used in the LDC system as well as the adequately proprieties for the region climate, consequently for this study.

4. Mathematical Formulation

4.1 Regenerator and Dehumidifier Equations

In this part, a mathematical model is presented to describe the principal equations of the selected LDC system. In this analytical model, the desiccant solution temperature is calculated in different positions which are presented in figure 1. In addition, the desiccant solution concentrations in the regeneration and dehumidification parts of LDC system are changed. Further, an analytical study is established to study the heat and mass transfer rates of different parts of LDC system as well as the operating performance.

The strong desiccant solution is pre-cooled by the return weak desiccant one in the SHX, where its temperature is reduced with unchanged concentration from point 1 to point 2. The weak desiccant solution temperature is calculated and shown in Eqs. (1) and (2), respectively.

$$T_2^{strong} = T_1^{strong} - \eta_1 * C_1^{\min} * \left(T_1^{strong} - T_4^{weak}\right) / C_1^{strong} \tag{1}$$

$$C_1^{\min} = \min\left(C_1^{strong}, C_4^{weak}\right) = \left(c_1^{strong} m_1^{strong}, c_4^{weak} m_4^{weak}\right) \tag{2}$$

After that, the strong desiccant solution passes through the heat exchanger (HE2) to reduce their temperature which is calculated and shown in Eqs. (3) and (4), respectively.

$$T_3^{strong} = T_2^{strong} - \eta_2 * C_2^{\min} * \left(T_2^{strong} - T_{in}^{cold}\right) / C_2 \tag{3}$$

$$C_2^{\min} = \min\left(C_2, C_{cold}\right) = \left(c_2^{strong} m_2^{strong}, c_{in}^{water} \rho_{in}^{cold} V^{cold}\right) \tag{4}$$

The strong desiccant solution is diluted by the inlet fresh air exactly in the dehumidifier core, where its concentration is calculated as follows:

$$\xi^{weak} = \left(m_1^{strong} \xi^{strong}\right) \Big/ \left(m_1^{strong} + m_{abs}\right) \tag{5}$$

Where m_{abs} represents the absorbed moisture from the fresh air which is expressed by:

$$m_{abs} = \frac{\left(p_3^f - p_3^d\right)}{RT\left(\dfrac{1}{k_l^f} + \dfrac{H}{k_l^d}\right)} A_{deh} \tag{6}$$

In the dehumidification process, a sensible heat exchange between the hot/humid fresh air and the strong/cold desiccant solution, and another latent which is released by the condensation process are considered. Therefore, the temperature of outlet desiccant solution can be written by:

$$T_4^{weak} = T_3^{strong} + \left(t_3^f - T_3^{strong}\right) . \left(\left(\frac{1}{k_s^f} + \frac{1}{k_s^d}\right) A_{deh} + m_{abs}\left(2500 - 2.35 * T_3^{strong}\right)\right)^{-1} \Big/ \left(\rho_1^{strong} m_3^{strong}\right) \tag{7}$$

After leaving the dehumidifier core, the diluted (weak) desiccant solution passes through the SHX, where it's pre-heated by the strong desiccant solution coming from the regenerator. Therefore, the desiccant solution temperature is expressed by:

$$T_5^{weak} = T_4^{weak} + \eta_1 * C_1^{\min} * \left(T_1^{strong} - T_4^{weak}\right) / C_4 \tag{8}$$

$$C_1^{\min} = \min\left(C_1, C_4\right) = \left(c_1^{strong} m_1^{strong}, m_4^{weak} c_4^{weak}\right) \tag{9}$$

DOI: https://doi.org/10.30564/jaeser.v2i1.387

From point 5 to point 6, the weak desiccant solution is heated by the hot water in the HE1 to attain the regeneration temperature with unchanged concentration. The weak desiccant solution temperature is calculated and shown in Eqs. (10) and (11), respectively.

$$T_6^{weak} = T_5^{weak} + \eta_3 * C_3^{\min} * \left(T_{in}^{hot} - T_5^{weak}\right) / C_5 \quad (10)$$

$$C^3_{\min} = \min\left(C_5, C_{hot}\right) = \left(c_5^{weak} m_5^{weak}, c_{in}^{hot} \rho_{in}^{hot} V^{hot}\right) \quad (11)$$

Inside the regenerator, it is regenerated after that by the inlet fresh air where the concentration is increased and calculated as follows:

$$\xi^{strong} = \left(m_5^{weak} \xi^{weak}\right) \Big/ \left(m_5^{weak} - m_{des}\right) \quad (12)$$

Where m_{des} represents the desorbed moisture by the fresh air which is expressed by:

$$m_{des} = \frac{\left(p_5^{weak} - p_5^f\right)}{RT\left(\frac{1}{k_l^f} + \frac{H}{k_l^d}\right)} A_{reg} \quad (13)$$

Where p presents the vapor pressure which based on the air temperature as described by [35]:

$$p = 0.61078 \exp\left(\frac{17.269 * t}{273.15 + t}\right) \quad (14)$$

In the regeneration process, as the high weak desiccant solution temperature and specific capacity, the fresh air is assumed to absorb both sensible and latent heat from the weak desiccant solution. Hence, desiccant solution temperature is calculated by:

$$T_1^{strong} = T_6^{weak} + \left(t_6^{weak} - T_6^a\right) \cdot \left(\left(\frac{1}{k_s^f} + \frac{1}{k_s^d}\right) A_{reg} + m_{des}\left(2500 - 2.35 * t_6\right)\right)^{-1} \Big/ \left(\rho_6^{weak} m_6^{weak}\right) \quad (15)$$

In this global LDC system, the desorbed moisture in the regeneration process is assumed the same quantity to the absorbed one in the dehumidification process. Therefore, the inlet desiccant solution concentration at point 1 is assumed strong one. These assumed equations are shown in Eqs. (16) and (17), respectively.

$$m_{des} = m_{abs} \quad (16)$$

$$\xi^{strong} = \xi^{inlet} \quad (17)$$

After the desiccant solution equations are calculated, the air properties such as temperature and humidity ratio are obtained. The temperature and humidity ratio of ambient air are partially reduced when it passes through the air-air heat exchanger. The outlet ambient air temperature and humidity ratio are expressed, respectively:

$$T_{II}^f = T_I^f - \frac{\left(T_I^f - T_{IV}^e\right)}{RT\left(\frac{1}{k_s^f} + \frac{1}{k_s^e}\right) c^a \rho^a V_I^f} A_{AAE} \quad (18)$$

$$W_{II}^f = W_I^f - \frac{\left(p_I^f - p_{IV}^e\right)}{RT\left(\frac{1}{k_s^f} + \frac{\delta}{k_m} + \frac{1}{k_s^e}\right) c^a \rho^a V_I^f} A_{AAE} \quad (19)$$

Further, they are significantly reduced in the dehumidifier core by the strong cold desiccant solution. The temperature and humidity ratio of supply air into the build are calculated by:

$$T_{III}^f = T_{II}^f - \frac{\left(T_{II}^f - T_{III}^s\right)}{RT\left(\frac{1}{k_s^f} + \frac{1}{k_s^e}\right) c^a \rho^a V_{II}^f} A_{deh} \quad (20)$$

$$W_{III}^f = W_{II}^f - \frac{\left(p_{II}^f - p_{III}^s\right)}{RT\left(\frac{1}{k_s^f} + \frac{\delta}{k_m} + \frac{1}{k_s^e}\right) c^a \rho^a V_{II}^f} A_{deh} \quad (21)$$

4.2 Heating and Cooling Systems

In the HE2, the desiccant solution temperature is reduced by cold water coming from a cooling system. Hence, the exchanged heat between them is calculated by the following expression:

$$Q^{cold} = m_2^{strong} * C_2^{strong} * \left(T_2^{strong} - T_3^{strong}\right) = m^{water} * C^{cold} * \left(T_{out}^{cold} - T_{in}^{cold}\right) \quad (22)$$

In the other hand, in the HE1, the desiccant solution is heated by a hot water leaving a heating system to attain the regeneration temperature. The heat transfer is calculated and shown in equation (23):

$$Q^{hot} = m_5^{weak} * C_5^{weak} * \left(T_6^{weak} - T_5^{weak}\right) = m^{water} * C^{hot} * \left(T_{in}^{hot} - T_{out}^{hot}\right) \quad (23)$$

4.3 LDC System Performance

The coefficient of performance (COP) of the global LDC system is calculated by the ratio of producing to input energy of the studied system.

$$COP = \frac{\text{producing\_energy}}{\text{input\_energy}} \quad (24)$$

DOI: https://doi.org/10.30564/jaeser.v2i1.387

Where the producing energy is the reduced energy from the fresh (ambient) air to the supply one which also is described by the cooling capacity (CC) of LDC system as defined by:

$$CC = m^a * c^a * \left(T_f^a - T_{sp}^a\right) \tag{25}$$

Therefore, the COP can be written as the ratio of CC to total energy consumption of the LDC system.

$$COP = \frac{CC}{\left(Q^{cold} + \left(E_{el}/0.3\right)\right) + Q^{hot}} \tag{26}$$

Where E_{el} represents the electrical energy consumption by the LDC system, we assumed that the LDC system consumed an electrical energy similar to the Liu system [36] and 0.3 is assumed the conversion coefficient from electric power to thermal energy [37].

In addition, the impacts of various operating and climatic on global performance are evaluated on the basis of the Moisture Removal Rates (MRR) and Sensible Heat Ratios (SHR).

The MMR (g/s) is the removed moisture from the fresh air as expressed:

$$MRR = m^a * \left(W_{amb} - W_{air,deh,out}\right) \tag{27}$$

The SHR represents the ratio of sensible to total energy as calculated:

$$SHR = \frac{Q_{sen}}{Q_{sen} + Q_{lat}} \tag{28}$$

Where $Q_{sen}$and Q_{lat} represent the sensible and latent energies, respectively.

5. Results and Discussion

5.1 Model Validation

To validate our analytical study, we illustrate a comparison between the experimental measurements and our analytical results of the selected LDC system. A similar input conditions used in the experimental work is proposed in the analytical resolution. In the first test, the operating conditions are: cold water temperature 16°C and flow rate 0.23 kg/s; hot water temperature 63°C and flow rate 0.15 kg/s; fresh and return air flow rates 509 m^3/h; and the fresh air temperature 30°C. Table 1 shows a comparison between the Liu experimental data and analytical cooling capacity values. By varying the inlet humidity values of fresh air from 11.43 g/kg to 16.84 g/kg, both experimental (CC_{exp}) and analytical cooling capacity ($CC_{analytical}$) are increased from 2.83 to 4.14 kW and from 3.24 to 3.8 kW, respectively. The average difference percentage between our results and Liu experimental data is within 10%.

In second case, the cold and hot water temperatures in the Liu experimental measurements are respectively 17°C and 61.5°C. In addition, the cold and hot water flow rates are 0.15 kg/s and 0.16 kg/s, respectively. Both analytical and measured cooling capacity variations are increased from 2.82 kW to 5.53 kW and from 3.19 to 5.35 kW, respectively, when the temperature values of fresh air is varied from 29.5°C to 36.18°C as shown in Table 2. The $CC_{analytical}$ in the Liu measurements increased similarly with an average difference percentage is about 10 %.

In tables 1 and 2, the cooling capacity is mostly affected by change in fresh air proprieties such as temperature and humidity ratio values which are increased as well as the air enthalpy. The highest of CC values can explained by the increasing of enthalpy difference between the fresh and leaving air from the LDC system. The lower variation in the humidity ratio provoked a change in the moisture removal rate MRR as well as the latent enthalpy. Acceptable average difference percentage between our analytical results and measured values is established. According to these results, we can use this analytical study to investigate the effects of the operating and climatic parameters on the selected LDC system.

Table 1. Comparison between the analytical and experimental results of influence of fresh air humidity on the cooling capacity

Temperature (°C)	Humidity ratio (g/kg)	$CC_{analytical}$ (kW)	CC_{exp} [35] (kW)	$\lvert CC_{analytical}-CC_{exp}\rvert / CC_{exp}$(%)
30,47	11.43	2,83	3,24	13
29,9	12.54	3,04	3,47	12
30,3	14.88	3,45	3,62	5
30,61	16.84	4,14	3,8	9

Table 2. Comparison between the analytical and experimental results of influence of fresh air temperature on the cooling capacity

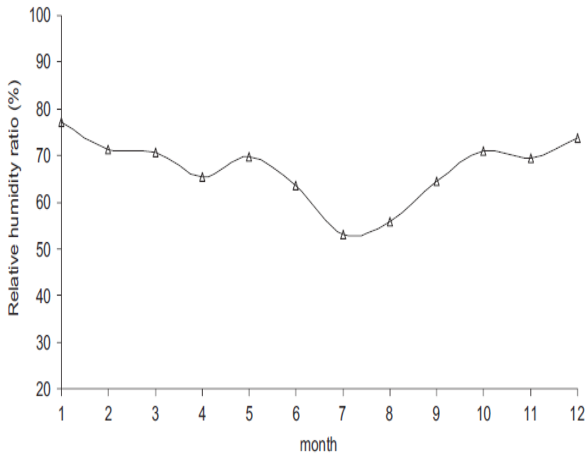
Temperature (°C)	Humidity ratio (g/kg)	$CC_{analytical}$(kW)	CC_{exp} [35] (kW)	$\lvert CC_{analytical}-CC_{exp}\rvert / CC_{exp}$(%)
29.49	15.54	2.828	3.19	11
30.64	15.04	2.854	3.21	12
31.68	13.47	3.202	3.26	2
32.58	14.37	3.577	3.85	7
33.73	14.30	4.149	4.53	9
34.61	14.56	4.952	5.75	14
36.18	15.01	5.538	5.35	3

5.2 Air and Desiccant Solution Cycle Thermal Processes

Majority of Mediterranean regions are usually hot and hu-

 DOI: https://doi.org/10.30564/jaeser.v2i1.387

mid areas. Tunisia located on the Mediterranean coast of North Africa and Tunis is the capital which located at 36° latitude and 10° longitude and is one of this region which characterized by 350 sunny days per year. The climatic data used in this study provided by the National Institute of Meteorology. The daily ambient temperature can attain as high as 46 °C in the summer days, however the minimum temperature values are less than 5 °C in winter season as described in figure 2.a. Figure 2.b presents the mean daily relative air humidity which reached at 53% and 77% in the summer and winter seasons, respectively[38].



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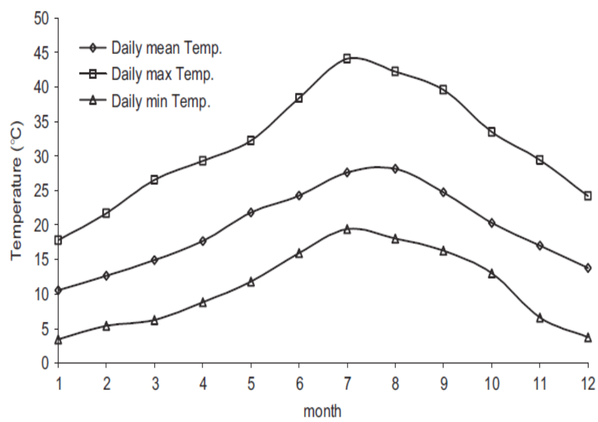

(b)

Figure 2. a. Daily mean, maximum and minimum temperature in Tunis, b. daily mean relative air humidity in Tunis.

The air temperature and humidity ratio entering to the selected LDC system are varied from point to another which air thermal process is presented in the psychometric chart as shown in figure 3 and listed in Table 3. The inlet ambient air is taken under Mediterranean climatic summer which characterized by hot and humid climate. The inlet air temperature ($T_{air,deh,in}$) and humidity ratio ($W_{air,deh,in}$) values taken at 13h are equal to 39.22°C and 23.9 g/kg, respectively. From point I to II, the ambient air is partially pre-cooled and dehumidified by the leaving exhaust air in the air to air heat exchanger, where the air temperature (humidity ratio) decreased by 10°C (6 g/kg). From point II to III, the air enters to the dehumidifier core where it makes contact with cold and strong desiccant solution. Thus, their values (air temperature and humidity ratio) are decreased significantly by 7°C and 10.5 g/kg. After that, the air is supplied into the build. The point c presents the indoor conditions (25°C and 50% in summer). The exhaust air passes through the air to air heat exchanger then exits with 36°C of temperature and 15 g/kg humidity ratio values, respectively, at point IV.

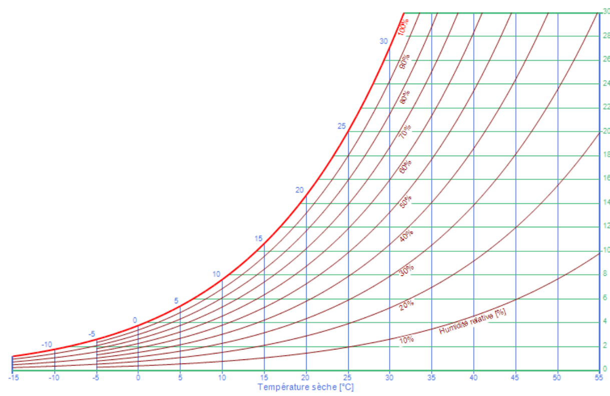

Figure 3. Air thermal process on psychometric chart

Table 3. The temperature and humidity ratio values on the air cycle diagram

Point	T_{air} [°C]	Wair [g/kg]
I	39.22	23.9
II	28.16	18.1
III	21.34	7.3
c	25	9.8
IV	36.38	14.8

Figure 4 depicts the desiccant solution cycle diagram and its process on the Conde diagram of aqueous lithium chloride solution, at the atmospheric pressure [27]. The strong desiccant solution concentration is assumed equal to the inlet desiccant solution concentration at point 1. Before entering the dehumidifier, the strong desiccant solution at point 1 is pre cooled by a return weak desiccant solution, resulting in a lower temperature without changing the concentration value at point 2. Then, desiccant solution passes through the heat exchanger (HE2) where its temperature decreased from point 2 to 3 by the cold water coming from the cooling system. In the dehumidifier core, the strong desiccant solution absorbed the

  DOI: https://doi.org/10.30564/jaeser.v2i1.387

moisture from the ambient air, where both desiccant solution temperature and concentration changed at point 4. On the way to the regeneration process, the weak desiccant solution temperature increased due to absorbing heat from the strong desiccant solution in the solution to solution heat exchanger (SHX). After that to attain the regeneration temperature, the weak desiccant solution heated in heat exchanger (HE1) by the hot water coming from the heating system, where the weak solution temperature increased from point 4 to 5 with unvaried concentration. In the regenerator, the both sensible and latent heat rates are absorbed by the fresh air from the weak desiccant solution (point 5 to 6). The desiccant solution temperature and concentration values on the operating LDC system cycle are presented in Table 4.

Table 4. The temperature and concentration values on the desiccant solution cycle diagram

Point	$T_{desiccant\ solution}$ [°C]	$\xi_{desiccant\ solution}$ [%]
1	55	35
2	33	35
3	22	35
4	29	30
5	51	30
6	65	30

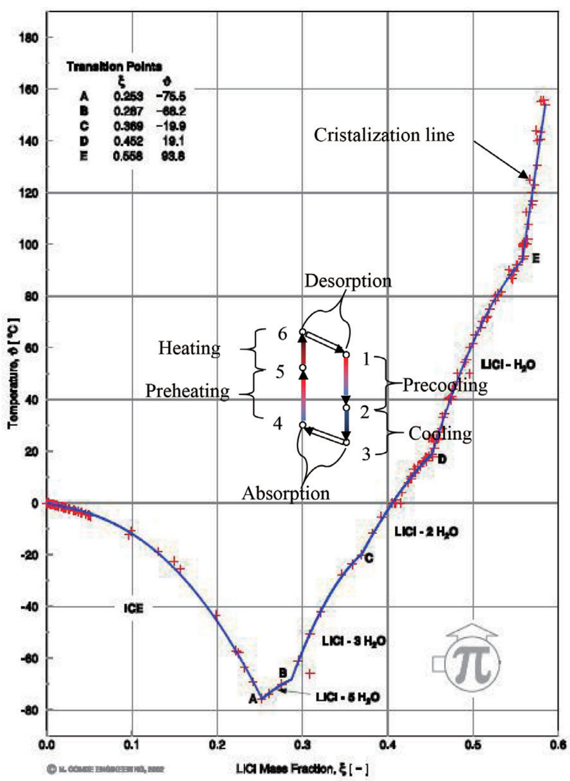

Figure 4. Desiccant solution cycle diagram and its process on the Conde diagram of aqueous lithium chloride solution

5.3 Influencing Factors

Various parameters such as the outdoor climatic parameters and operating conditions have considerable effects on the selected LDC system efficiency. The influencing parameters gap and the reference values evaluated in this study are listed in Table 5.

Table 5. The reference values and variation ranges of the operating and climatic parameters

Parameter	Reference value	Minimum value	Maximum value	Unit
$T_{sol,reg,in}$	45	40	70	°C
$T_{sol,deh,in}$	20	14	26	°C
T_{amb}	38	30	42	°C
HR_{amb}	50	50	80	%
m_{sol}	0.25	-	-	kg/s
m_{air}	0.35	-	-	kg/s

5.3.1 Influence of Ambient Air Temperature (Tamb)

Under a variety of places, the ambient air temperature (T_{amb}) is one of the climatic parameters which varied from place to another and has a significant effect on the LDC performance. By keeping all other parameters such as inlet solution temperature in the regenerator and dehumidifier cores, air and solution flow rates at the reference values, the influence of ambient air temperature on the LDC system are shown in figure 5. By changing ambient air temperature values from 30 to 42°C, the humidity ratio values ($W_{air,deh,out}$) increased by 6 $g.kg^{-1}$ and the outlet air temperature ($T_{air,deh,out}$) gradually changed by 0.7°C as plotted in figure 5.a. As depicted in figure 5.b, the moisture removal rate (MMR) is depending to the ambient humidity ratio which enhanced from 3 to 7$g.kg^{-1}$ with the ambient air temperature. In addition, a high ambient air temperature leads to a great sensible heat transfer values (SHR) which increased from 0.406 to 0.426 as plotted in figure 5.b. In consequence, it can be explained the raise moisture removal rate and the sensible heat transfer rate by the temperature difference between the inlet ambient air and cold strong desiccant solution which minimized hot ambient air temperature and the correlation between the sensible and latent heat variation on the dehumidification process.

Figure 5.c shows the cooling capacity and coefficient of performance of the selected LDC system under various ambient air temperature values. Further, both cooling capacity and coefficient of performance increase by 70% with the raise of the inlet ambient air temperature from 30 to 42°C. This signifies that the cooling capacity as well the performance of LDC system varied with the climatic conditions and the best system operation achieved in the hot climate. We can explained this enhance in the global performance by the air enthalpy difference between the

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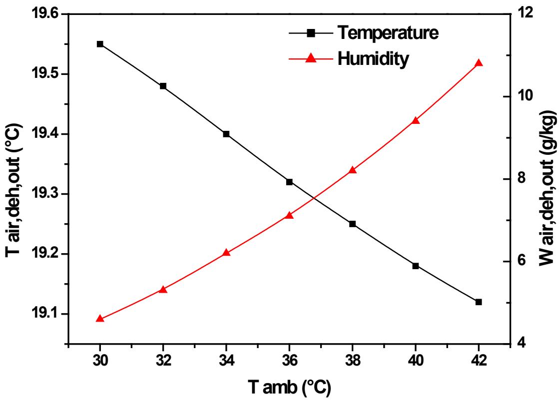
inlet and outlet of LDC system.



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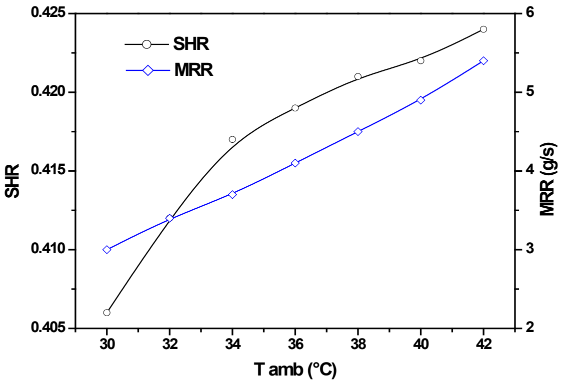

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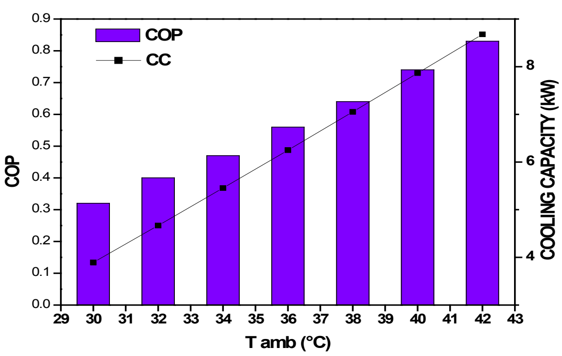

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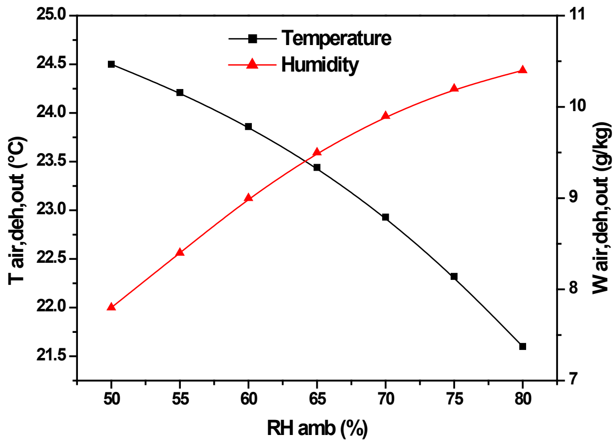
Figure 5. Influence of ambient air temperature (T_{amb}) on: a. outlet air state from the dehumidifier, b. SHR and MRR and c. the performance of the system

5.3.2 Influence of Ambient Air Relative Humidity (RHamb)

By keeping the same inlet operating parameters at the reference values, Figure 6.a describes the ambient relative humidity (RH_{amb}) impact on the outlet air conditions. By changing the relative humidity (RH_{amb}) from 50 to 80%, the air temperature and humidity ratio values leaving the LDC system reduced by 3°C and increased by 2.5g.kg^{-1}, respectively. The ambient relative humidity and air temperature are related by the proportional correlation which explained the same response of the LDC system under different relative humidity values as well as ambient temperature. The moisture removal rate (MRR) enhanced by 2.4 g.s^{-1} however, the sensible heat ratio (SHR) decreased by 11% when the RH_{amb} varied from 50 to 80% as seen in Figure 6.b. It can be explained the raise moisture removal rate by the increase of mass transfer potential between the ambient air and cold desiccant solution which provoked a great latent heat transfer though the reduce sensible heat transfer rate may clarified by the lower temperature exchange between the inlet ambient air and the cold weak desiccant solution which absorbed high moisture quantities. As seen in figure 6.c, the increase of ambient relative humidity (RH_{amb}) leads a high cooling capacity values which increased by 1 kW, in addition the raise of global performance of the LDC system by 8%.



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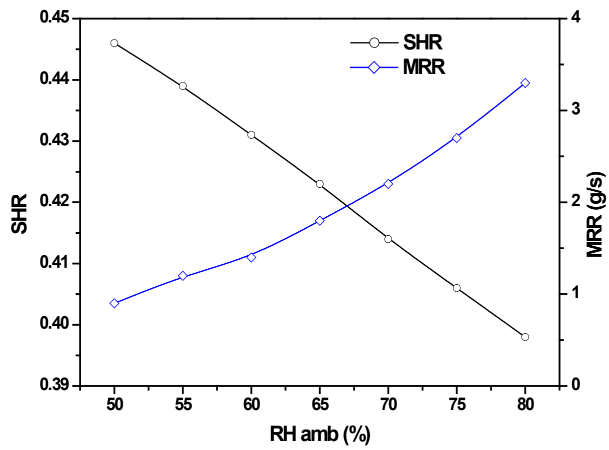

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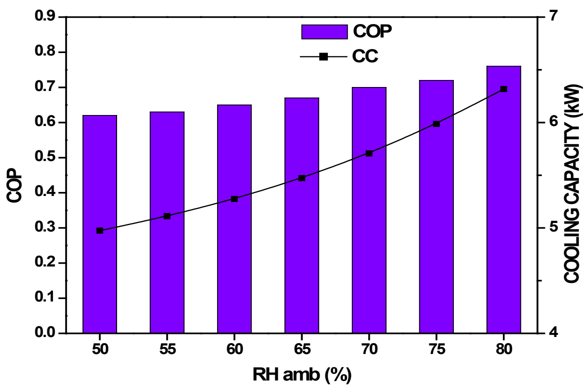

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Figure 6. Influence of ambient air relative humidity (RHamb) on: a. outlet air state, b. SHR and MRR of the dehumidifier and c. the performance of the system

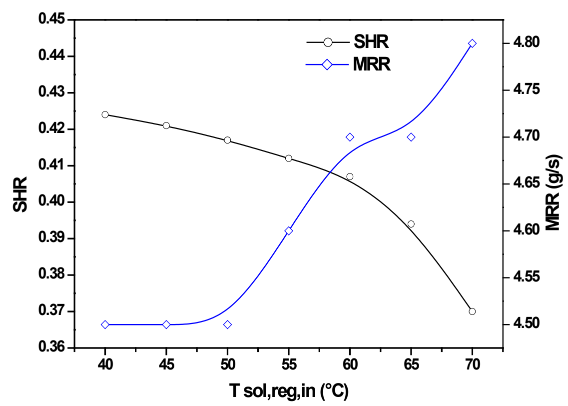
We can concluded from the analysis of the impacts of climatic conditions parameters to choose of the operating region of this type of LDC system has significant effects on the operating LDC system. When both ambient air temperature and humidity ratio are at a high values, the LDC system effectiveness enhances. The daily ambient temperature in Tunisia can arrive at more than 40 °C besides, the relative humidity which attains more than 70%. As consequent, this LDC system type is considerably efficient under hot and humid climate and especially under Tunisia climate with high performance.

5.3.3 Influence of Solution Inlet Temperature to the Regenerator (Tsol,reg,in)

The choice of the best desiccant solution is one important factor to operate the LDC system with high performance. The type of desiccant liquid should be carefully selected which is based on significant proprieties such as crystallization limit, the regeneration temperature. The lithium chloride solution (LiCl) is the good choice in the LDC system for the reason that adequately proprieties for the Tunisia climate consequently for this study [32-34].

By varying the desiccant solution temperature ($T_{sol,reg,in}$) from 40 to 70°C and keeping the ambient air propriety (T_{amb} and RH_{amb}) at the reference values, the moisture removal rate (MRR) increased from 4.5 to 4.8 $g.s^{-1}$ as shown in figure 7.a. However, the sensible heat ratio decreases (SHR) reduced from 0.84 to 0.77. This variation can be explained by changing of the desiccant solution concentration between the regeneration and dehumidification processes. In the regenerator part, when the desiccant solution enters with high temperature value, it has a high vapor pressure [34]. So, the potential to transfer the moisture from this desiccant solution to the air stream becomes greater when it makes contact with the regenerating air.

Figure 7.b shows the effect of the desiccant solution temperature ($T_{sol,reg,in}$) on the cooling capacity and coefficient of performance of the LDC system. By raising the desiccant solution temperature ($T_{sol,reg,in}$) progressively from 40 to 70°C, the cooling capacity of the LDC system increased by 8%. A high regeneration temperature of desiccant solution leads a large potential transfer of moisture to the air stream so; the leaved strong desiccant solution has a good ability to absorb the moisture at the dehumidification process. However, the coefficient of performance decreased by 60%. The explanation of this reduction is due to the high required energy to heat the desiccant solution to attain the regeneration temperature as well as to operate the global LDC system.



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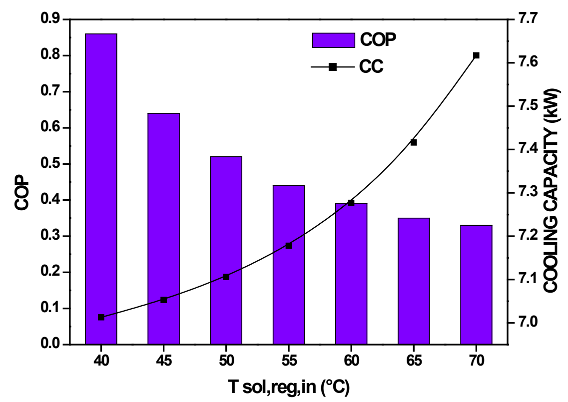

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Figure 7. Influence of inlet desiccant solution temperature to the regenerator (Tsol,reg,in) on: a. SHR and MRR and b. the performance of the system

5.3.4 Influence of Solution Inlet Temperature to the Dehumidifier (Tsol,deh,in)

At the dehumidification process, the inlet desiccant solution temperature ($T_{sol,deh,in}$) is an important factor to obtain a good absorption phenomenon between the ambient air and the desiccant solution. As depicted from figure 8.a, by varying desiccant solution temperature values ($T_{sol,deh,in}$) from 14 to 26°C, the air temperature($T_{air,deh,out}$) and humidity ratio ($W_{air,deh,out}$) enhanced by 2.5°C and13 $g.kg^{-1}$, respectively. In addition, the raise of desiccant solution temperature provides a low moisture removal rate (MRR) sensible heat ratio (SHR) values which varied from 6.9 to 2.5 $g.s^{-1}$ and from 0.46 to 0.38, respectively as described in figure 8.b.

At low temperature values, the desiccant solution has a low vapor pressure [34]. Hence, the potential to absorb the moisture from the ambient air entering the dehumidifier increases significantly so, this is explained the increase of humidity ratio ($W_{air,deh,out}$) and decrease of moisture removal rate (MRR) when the desiccant solution temperature increased. However, the small raise of sensible heat rate (SHR) is due to reduction of latent heat in the absorption phenomenon so, the increase slowly the sensible heat due to the small difference between the ambient air and the desiccant solution.

Figure 8.c shows the variations of cooling capacity and coefficient of performance of the LDC system. As seen that the cooling capacity and coefficient of performance reduced by 0.8 kW and 0.33, respectively, when desiccant solution temperature ($T_{sol,deh,in}$) increased from 14 to 26°C. In the dehumidifier core, the influence of the inlet desiccant solution temperature on the leaving air proprieties from the LDC has a strong effect on the cooling capacity and the global performance. This signifies that the LDC system performance improved at lower desiccant solution temperature ($T_{sol,deh,in}$).

We can conclude from the desiccant solution analysis that in addition to choose of the desiccant solution type, his operating temperature has an influence on the performance of LDC system. When desiccant solution temperature in both regeneration and dehumidification processes is at the adequate values, the LDC system effectiveness enhanced. As consequent, this LDC system type is effectively under Tunisia climate with best choosing of the operating parameters to attain the best performance.

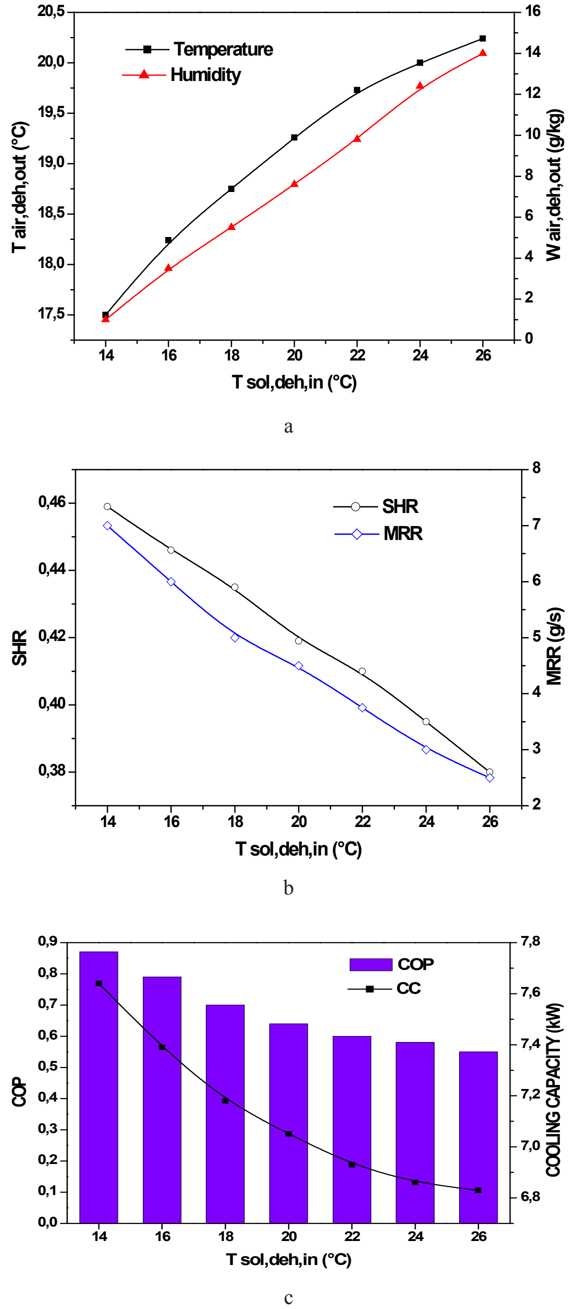

Figure 8. Influence of inlet desiccant solution temperature to the dehumidifier (Tsol,deh,in) on: a. outlet air state from the dehumidifier, b. SHR and MRR and c. the performance of the system

6. Conclusion

Desiccant liquid cooling (LDC) system under various climatic and operating conditions is studied. The analytical equations are developed to describe the sensible and latent heat transfer between the ambient air and desiccant solution and to calculate the energy performance of the select-

DOI: https://doi.org/10.30564/jaeser.v2i1.387

ed LDC system. The agreement between our analytical results and Liu experimental measurements allowed us to validate our analytical model to study the feasibility of the LDC system under Mediterranean place. The influence of the climatic and operating parameters on the LDC system performance is obtained. Therefore, the results can be established:

The impact of the ambient climatic conditions (T_{amb}, RH_{amb}) on the LDC system is important. Both cooling capacity and coefficient of performance increase by 70% with the raise of the inlet ambient air temperature from 30 to 42°C as well as the increase of ambient relative humidity (RH_{amb}) leads a high cooling capacity values which increased by 1 kW, in addition the raise of global performance of the LDC system by 8%.

Inlet desiccant solution temperature in the regenerator core has a strong effect on the supplied air temperature and humidity ratio values when it is sufficiently higher.

Finally, the raise of desiccant solution temperature provides a low moisture removal rate (MRR) sensible heat ratio (SHR) values which varied from 6.9 to 2.5 $g.s^{-1}$ and from 0.46 to 0.38 at the dehumidification process. This can be explained by the high potential to absorb the moisture from the ambient air entering the dehumidifier.

Consequently, this study provides a solution to give best gap of the selected LDC system operation. In addition, this study should be followed by an experimental work under different climatic conditions to validate.

Nomenclature	
A	Area (m^2)
c	specific heat capacity($J.kg^{-1}.K^{-1}$)
C	Mass quantity multiplying specific heat capacity ($J.kg^{-1}.K^{-1}$)
CC	Cooling capacity
COP	Coefficient of performance
E	energy consumption (kW)
H	Henry's low constant
k_s	convective heat transfer coefficient ($W.m^{-2}.K^{-1}$)
k_l	convective mass transfer coefficient ($m.s^{-1}$)
m	Mass quantity (kg)
MRR	Moisture Removal Rate ($g.s^{-1}$)
P	Pressure (Pa)
p	vapor pressure (Pa)
Q	energy (J)
R	Ideal gas constant (8.314 $J.mol^{-1}.K^{-1}$)
RH	Relative Humidity (%)
SHR	Sensible Heat Ratio
t	temperature (°C)
T	Temperature (°K)
V	Volume flow ($m^3.s^{-1}$)
W	humidity ratio ($kg.kg^{-1}$)
Greek letters	
δ	membrane thickness (m)
ξ	Mass desiccant solution concentration (%)
ρ	density ($kg.m^{-3}$)
η_1, η_2, η_3	effectiveness of the HE1, SHX and HE2
Superscripts	
a	air
cold	cold water
d	desiccant solution
e	exhaust air
f	fresh air
hot	hot water
inlet	inlet
min	minimal value
m	membrane
s	supply
strong	strong desiccant solution
water	water
weak	weak desiccant solution
Subscripts	
abs	absorbed
AEE	air to air heat exchanger
amb	ambient air
Deh	dehumidifier
des	desorbed
el	electric energy (W)
f	fresh air
in	inlet
lat	latent heat
out	outlet
Reg	regenerator
sen	sensible heat
sp	supply air
1,2,3,4,5,6	desiccant solution positions
I,II,III,IV	air stream positions

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