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Enhanced thermodynamic assessments of the novel desiccant air cooling system for sustainable energy future

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ABSTRACT

In this study, a newly developed desiccant air cooling system with desiccant wheel, sensible heat wheel and evaporative air cooler is assessed. The main purpose of the study is to investigate the developed desiccant air cooling system and compare its performance with the previous studies to show reliability of the developed system. The energy, exergy and sustainability analyses methods are used to assess the system. The data are obtained at 35 °C and 101.3 kPa environment condition. According to the exergy analysis, this system can be largely improved. The electrically and thermally driven energy coefficient of performance of the overall system (6.71 and 0.77) are higher than the corresponding exergy coefficient of performance (0.198 and 0.463). Energy coefficient of performance shows that the performance of the overall system is high, however electrically driven exergy coefficient of performance shows that some modifications can be realized in order to improve the final performance of the system. Among the components, desiccant wheel should be improved, because its exergy destruction ratio is 42.87% and the exergy efficiency is about 38.35%. On the other hand, sensible heat wheel exergy destruction ratio is computed as 10.35%, and its exergy efficiency is determined to be 57.69%. Among the components, desiccant wheel has less sustainability than others considering a combination work together. The effectiveness of the components from maximum to minimum are as sensible heat wheel, regenerative evaporative cooler and desiccant wheel, respectively. For better effectiveness, the component devices should be working around similar high level. So, the urgent improvements for the devices are in the order of desiccant wheel, regenerative evaporative cooler and sensible heat wheel, respectively.

1. Introduction

Energy conservation is a major issue in the world due to the fast growth of energy consumption and environmental degradation (Zhu and Chen, 2014). Electrical household appliances (such as Heating, Ventilation and Air Conditioning systems - HVAC) are responsible for one of the highest electricity consumption rate in buildings (Aridhi et al., 2017). Indeed, maintaining the desired indoor conditions requires a huge amount of energy, particularly in humid/hot climate. Energy consumption of HVAC systems is also directly proportional to the population. People generally stay in buildings that require indoor thermal comfort, and this situation influences the HVAC energy consumption (Allouhi et al., 2015; Nunes et al., 2016).

The energy consumption of HVAC systems could represent about 50% of the total electrical consumption in buildings (Curto et al., 2018). Effective utilization of energy coming from air could represent an important opportunity to decrease the environmental impact. From an energy conservation point of view, desiccant air conditioning systems are alternative to HVAC systems due to their eligibility in using different energy sources (electricity, thermal source, renewable source, etc) (Choudhury et al., 2013; Enteria et al., 2016). They are widely known nowadays due to technological, economic and social developments in the field of energy and the growing attention toward the environment. Desiccant air conditioning systems can extract moisture from humid air with less energy consumption than traditional systems (Peng et al., 2017). These systems can control latent and sensible thermal contents, chemical contents and microorganism (viruses/bacteria) of the air

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Nomenclature

$\dot{E}n$	energy rate (kW)
ex	specific exergy (kJ/kg)
$\dot{E}x$	exergy rate (kW)
c_p	specific heat (kJ/kgK)
h	specific enthalpy (kJ/kg)
m	mass flow rate (kg/s)
$\dot{Q}$	heat rate (kW)
R_a	universal gas constant (kJ/kgK)
s	specific entropy (kJ/kgK)
$\dot{S}$	entropy generation rate (kW/K)
SI	sustainability index
T	temperature (^{o}C or K)
$\dot{W}_{tot}$	fan power (kW)

Greek symbols

ϵ	effectiveness (−)
Ψ	exergy efficiency (%)
ω	humidity ratio (kg_w/kg_{da})
$\overline{\omega}$	mole fraction ratio (−)

Subscripts

0	reference environment (dead state)
air	air
av	average
ch	chemical
cooling	cooling capacity
dest	destruction
dp	dew point
dw	desiccant wheel
el	electrical
ex	exergy
i	ith component
in	input
loss	loss
out	output
rec	regenerative evaporative cooler
reg	regeneration
shw	sensible heat wheel
SI	supply inlet
SO	supply outlet
sys	system
tot	total
th	thermal
wb	wet-bulb

Abbreviations

COP	coefficient of the performance
HVAC	heating, ventilating and air conditioning
REC	regenerative evaporative cooler

(Jani et al., 2017). Desiccant air conditioning systems generally include desiccant & heat wheels and direct evaporative coolers (Panaras et a., 2011; Enteria et al., 2013a,b).

The lowest energy demand to operate a desiccant air conditioning system is determined by the availability analysis. In thermodynamic terms, exergy is the other name of the availability. Considering only the first law of thermodynamics is not enough to completely characterize the desiccant air conditioning. So, the system performance evaluation considering the second law of thermodynamics is necessary (Bejan, 2006).

In the literature, there are some studies on desiccant air cooling systems with regenerative evaporative cooler (Chung and Lee, 2011; Kashif Shahzad et al., 2018; Choi et al., 2014; La et al., 2012; Chen et al., 2018; Hwang et al., 2017; Jani et al., 2016a,b). In addition, there are a few study about advanced thermodynamics assessments of the desiccant air cooling systems that may be related to this study as follows: Zhu and Chen (2014) studied on the novel marine rotary desiccant air-conditioning system. It was found that the *COP* and exergy efficiency of the system were inversely proportional to the regeneration (desiccant drying) temperature. The regeneration air heater, desiccant wheel and regeneration air leaving the desiccant wheel had highest exergy losses. Xiong et al. (2010) applied exergy analysis to the proposed two stage liquid desiccant dehumidification process assisted by calcium chloride ($CaCl_2$) solution. It was found that the proposed system increased the concentration variance and the pre-dehumidification of $CaCl_2$ comparing to basic liquid desiccant dehumidification system. The proposed system increased the exergy efficiency from 6.8% to 23.0%, and the thermal coefficient of performance (*COP*) from 0.24 to 0.73. Tu et al. (2015a) worked on the desiccant dehumidification and cooling systems; the proposed model consisted of desiccant wheel, heat recovery exchanger and single-stage heat pump. It was determined that the evaporative cooler in the processed air side was changed with a sensible heat exchanger to increase the performance. Moreover, if the electrical heater is changed with a heat pump system, the performance of the system could be increased. Uckan et al. (2014) applied exergy analysis on the desiccant based evaporative air conditioning system. The exergy input, output, destruction and efficiency rates were obtained. The computed overall system's exergy efficiency was about 40.7% at the reference temperature of 15 °C. The reference temperature was theoretically changed in the range 0 °C and 30 °C; as a result, the overall exergy efficiency of the system varied from 56% to 25%, respectively. Khalid Ahmed et al. (1998) worked on the hybrid air-conditioning cycle with emphasis on a partly closed solar regenerator. It was assessed that high levels of vapor pressure in the environment caused high irreversibility, low exergy efficiency and low *COP* rates for the system. Enteria et al. (2015) performed exergoeconomic analysis to the desiccant evaporative air-conditioning system. The exergy efficiency, exergy destruction ratio, cost rate and exergoeconomic factors were determined. The maximum exergoeconomic factors were found for the exit air fan, outdoor air fan and secondary heat exchanger. Peng et al. (2017) simulated a liquid desiccant evaporative cooling system. It was assessed that the performance of the system was affected by the hot water temperature, the air/solution flow rate, and relative humidity of the air. The exergy efficiencies of the exchanger, dehumidifier and regenerator were the lowest in the system. Enteria et al. (2013a) performed first and second laws of thermodynamics on the solar assisted desiccant cooling unit which consisted of solar collector, water storage tank, heating coil, desiccant wheel, heat exchanger, direct evaporative cooler and fan. It was found that the solar collector had the highest exergy destruction and loss rates. Also, the electrically driven energy and exergy overall *COP* rates are determined as 1.238 and 0.084, respectively. Tu et al. (2015b) studied the influence of the number of stages on the heat source temperature of the desiccant wheel dehumidification system; in particular, they considered two different cooling and heating media: a general refrigerant and water. When the water was used, the number of stages increased and the heat source temperature did not decrease continuously. When the refrigerant was used, the number of stages increased and the heat source temperature decreased. Jani et al. (2015) applied energy analysis on a hybrid air conditioning system including solid desiccant wheel and vapor compression unit. The cooling capacity was arranged as 1.8 kW and the system was modeled by TRNSYS program for the cooling season. The humidity ratio and ambient temperatures effects on the cooling capacity were also investigated. It was found that the maximum *COP* (around 4.75) is obtained with the following conditions: ambient temperature equal to 35 °C and ambient humidity equal to 23 g/kg. Qi et al. (2016) applied titanium dioxide superhydrophilic coating on the dehumidifier surface to enhance the performance of the cooling. The coating materials were mainly considered as SUS304, SUS316, and SUS410 which were self-cleaning and activated with ultraviolet light. It was found that this coating increased the heat and mass transfer performance of the desiccant dehumidification. Asim et al. (2015) studied on the solid waste of agricultural products as chemical desiccants to make green desiccant cooling systems. The physical and chemical properties of the agricultural waste were investigated and the proposed green desiccant cooling system was compared with other desiccant cooling systems.

The aim of this study is to assess the newly designed desiccant air cooling systems with regenerative evaporative cooler from an energy, exergy and sustainability point of view. The advantage of this novel developed system is to obtain cool air under 20 °C with almost 50% relative humidity. This application can be used when the regeneration temperature is 60 °C. This study differs from the previously conducted ones as follows: (i) exergy analysis is applied to the newly developed desiccant air cooling system with regenerative evaporative cooler using actual working data, (ii) systems' and components' entropy production rates are determined, (iii) sustainability of the system and components are calculated, (iv) All four *COP* rates (electrically driven energy and exergy *COP* and thermally driven energy and exergy *COP*) are investigated. It is possible to make different designs by using the components of the system to increase the overall performance. The main novelty of this study is to create a novel desiccant air cooling system with a new design by using sensible heat wheel, desiccant wheel, heating coil, regenerative evaporative cooler, air filters and fan at new locations/combinations in the system.

2. System description

The system mainly consists of sensible heat wheel, desiccant wheel, heating coil, regenerative evaporative cooler, air filters and fans. The schematic layout of the system is illustrated in Fig. 1. The innovation with new configuration is achieved by the design of the components. As is seen in Fig. 1, there are two channels in the system in which there are two wheels (desiccant wheel and sensible heat wheel) connected with both channels. At the bottom channel, a filter is used before the desiccant wheel. After the desiccant wheel, a fan is used to transfer the air to the sensible heat wheel. Then, a regenerative evaporative cooler is used to cool the air. At the upper channel, a filter is used before the sensible heat wheel which is connected with the bottom channel. Heating coil is then used for heating the air that passes through the desiccant wheel which is connected with the bottom channel. Finally, a fan is used to transfer air to the atmosphere.

There are fourteen points to assess the system in every step. The working process in these points can be explained as follows:

(1 + 7’)-(2): Air (Return air + outdoor make up air) enters to the

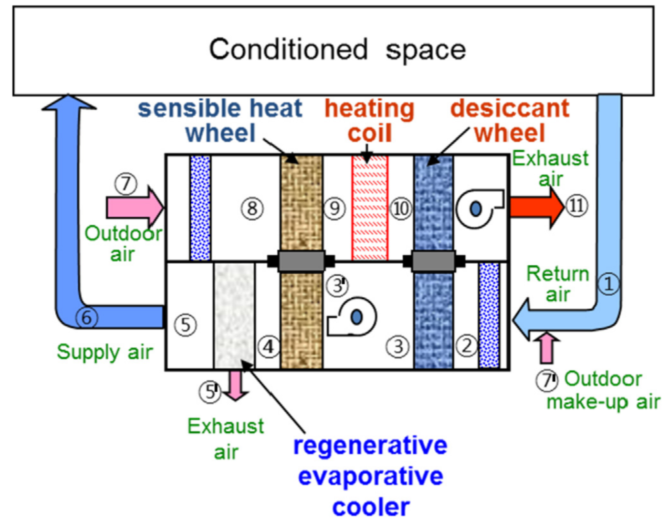

Fig. 1. Schematic layout of the system.

system through filter (Temperature increases, relative humidity and pressure decrease)

(2)–(3): Air goes through desiccant wheel by taking heat from other channel (Temperature increases, relative humidity and pressure decrease)

(3)-(3′): Air is sent to the sensible heat wheel by fan (Pressure increases)

(3′)-(4): Air goes through sensible heat wheel by giving its heat to the other channel (Temperature and pressure decrease, relative humidity increases)

(4)–(5): Air goes through regenerative evaporative cooler (a kind of heat and mass exchanger) (Temperature and pressure decrease, relative humidity increases)

(5)=(6): Air is sent to the conditioned space

(7)–(8): Outdoor air enters to the system through filter (Pressure decreases)

(8)–(9): Outdoor air goes through sensible heat wheel by taking heat from other channel (Temperature increases, pressure and relative humidity decrease)

(9)–(10): Outdoor air goes through heating coil by taking heat (Temperature increases, pressure and relative humidity decrease)

(10)–(11): Outdoor air goes through desiccant wheel to the environment by giving its heat to the other channel (Temperature and pressure decrease, relative humidity increases).

The fan power of the process channel at the bottom side of the system (2-3-3’-4-5) is 1.25 kW, while regenerative channel at the upper side of the system (8-9-10) has 0.82 kW fan power. Furthermore, regenerative evaporative cooler fan power is 0.06 kW. The air enters to the system at 27 °C and 49.6% relative humidity. After the desiccant wheel, the temperature of the air increases to 42.3 °C and the relative humidity decreases to 15.45%. Then, the temperature of the air decreases to 38.2 °C and the relative humidity increases to 19.21% after the sensible heat wheel. By the utilization of the regenerative evaporative cooler, air temperature decreases to 19.8 °C, relative humidity reaches to 55.76% and then it is sent to the conditioned space. The data of the system is tabulated in Table 1. The system is developed and tested in the Korea Institute of Science and Technology (KIST) laboratory. The experimental data are obtained under the environmental conditions of 35 °C temperature and 39.75% relative humidity considering the summer days of Republic of Korea. After the verification of the data by the KIST, the analyses are started to assess the newly developed solid desiccant cooling system. The environment temperature, pressure,

Table 1
Data of the system.

Parameter	Units	1	2	3	3′	4	5	5′	6	7	7′	8	9	10	11
Pressure	kPa	101.3	101.2	101.1	101.6	101.4	101.3	101.3	101.3	101.3	101.3	101.2	101.1	101	100.8
Dry bulb temperature	°C	27	29.4	42.3	42.3	38.2	19.8	30.6	19.8	35	35	35	40.9	60	41.3
Humid ratio	g_v/kg_{da}	11.1	12	8.04	8.04	8.04	8.04	28.2	8.04	14.1	14.1	14.1	14.1	14.1	19.8
Relative humidity	%	49.6	46.6	15.4	15.4	19.2	55.8	99.6	55.8	39.8	39.8	39.8	28.9	11.2	39.4
Wet bulb temperature	°C	19.5	20.9	21.8	21.8	20.6	14.4	30.5	14.4	23.9	23.9	23.9	25.4	29.8	28.7
Dew point temperature	°C	15.6	16.8	10.8	10.7	10.7	10.7	30.5	10.7	19.3	19.3	19.3	19.3	19.3	24.7
Saturation temperature	°C	19.4	20.8	21.7	21.6	20.4	14.3	30.5	14.3	23.7	23.7	23.7	25.2	29.5	28.5
Enthalpy	kJ/kg	55.5	60.2	63.3	63.3	59.1	40.3	102.9	40.3	71.4	71.4	71.4	77.4	97.1	92.5
Volumetric flow rate	dm^3/s	823	1176	1176	1176	1176	823	353	823	823	353	823	823	823	823
Mass flow rate	g/s	943	1348	1348	1348	1348	943	405	943	943	405	943	943	943	943
Vapor pressure	Pa	1777	1918	1293	1293	1293	1293	4395	1293	2246	2246	2246	2246	2246	3126
Saturated vapor pressure	Pa	3567	4102	8338	8338	6704	2310	4394	2310	5628	5628	5628	7745	19944	7911
Specific heat of air	J/kgK	1023	1024	1025	1025	1023	1018	1038	1018	1028	1028	1028	1029	1035	1034
Specific heat of water	J/kgK	1864	1865	1868	1868	1867	1863	1865	1863	1866	1866	1866	1868	1874	1868

relative humidity, humidity ratio and saturated water vapor pressure are measured as 35 °C, 101.325 kPa, 39.75%, 0.01373 kg/kg and 5.628 kPa, respectively.

3. Analysis

The solid desiccant cooling system is analyzed from an energy, exergy and sustainability point of view. In this regard, desiccant wheel, sensible heat wheel, regenerative evaporative cooler and overall system are investigated.

3.1. Energy analysis

The general energy balance of the system is written as follows:

$$\sum \dot{E}n_{in} = \sum \dot{E}n_{out} \tag{1}$$

where "$\dot{E}n_{in}$" is energy input rate and "$\dot{E}n_{out}$" is energy output rate.

The energy rate of a fluid ($\dot{E}n_i$) (where i corresponds to the working fluid - i.e. air) can be calculated as follows:

$$\dot{E}n_i = \dot{m}_i\, h_i \tag{2}$$

where "$\dot{m}$" is mass flow rate and "h" is enthalpy of fluid.

The thermally driven energy *COP* of the system (COP_{th}) is determined by;

$$COP_{th} = \frac{\dot{Q}_{cooling}}{\dot{Q}_{reg}} \tag{3}$$

where "$\dot{Q}_{cooling}$" is the cooling capacity rate and "$\dot{Q}_{reg}$" is the regeneration heat rate of the system as follows:

$$\dot{Q}_{cooling} = \dot{m}_1 (h_1 - h_6) \tag{4}$$

$$\dot{Q}_{reg} = \dot{m}_9 (h_{10} - h_9) \tag{5}$$

The electrically driven energy *COP* (COP_{el}) of the system is calculated as;

$$COP_{el} = \frac{\dot{Q}_{cooling}}{\dot{W}_{tot}} \tag{6}$$

where "$\dot{W}_{tot}$" is the total fan power of the system which is measured as 2.13 kW during experiment.

The effectiveness of the air cooling systems is generally explained with the wet-bulb effectiveness (ϵ_{wb}), which is the ratio of the device temperature depression to the potential wet-bulb depression as follows (Caliskan et al., 2011a,b):

$$\epsilon_{wb} = \frac{T_{SI} - T_{SO}}{T_{SI} - T_{wb,SI}} \tag{7}$$

where "T_{SI}" is the supply inlet dry-bulb temperature of the air, "T_{SO}" is the supply outlet dry-bulb temperature of the air and "$T_{wb,SI}$" is the supply inlet wet-bulb temperature of the air.

The dew point effectiveness (ϵ_{dp}) is also another method to measure the effectiveness of the desiccant air cooling system. The dew point effectiveness is the ratio of temperature depression to the potential dew point depression as follows (Caliskan et al., 2012a,b):

$$\epsilon_{dp} = \frac{T_{SI} - T_{SO}}{T_{SI} - T_{dp,SI}} \tag{8}$$

where "$T_{dp,SI}$" is the supply inlet dew point temperature of air.

Let's move to characterize each single component present in this system:

3.1.1. Desiccant wheel (DW)

The energy balance of the desiccant wheel is written as follows:

$$\dot{E}n_2 + \dot{E}n_{10} = \dot{E}n_3 + \dot{E}n_{11} + \dot{Q}_{loss,dw} \tag{9}$$

where "$\dot{Q}_{loss,dw}$" is the heat loss rate of the desiccant wheel.

The general effectiveness of the desiccant wheel (ϵ_{dw}) can be calculated as follows:

$$\epsilon_{dw} = \frac{\dot{m}_2 (T_3 - T_2)}{\dot{m}_{10} (T_{10} - T_2)} \tag{10}$$

The specific humidity based effectiveness of the desiccant wheel ($\epsilon_{dw,\omega}$) is computed as (Dincer and Rosen, 2013);

$$\epsilon_{dw,\omega} = \frac{\omega_2 - \omega_3}{\omega_2 - \omega_{3,ideal}} \tag{11}$$

where "ω" is the specific humidity and "$\omega_{3,ideal}$" is the specific humidity at the desiccant wheel exit in the ideal case. $\omega_{3,ideal}$ can be taken as zero. When it reaches this value it means that the ideal desiccant wheel completely dehumidifies the air.

3.1.2. Sensible heat wheel (SHW)

The energy balance of the sensible heat wheel is determined by;

$$\dot{E}n_{3'} + \dot{E}n_8 = \dot{E}n_4 + \dot{E}n_9 + \dot{Q}_{loss,shw} \tag{12}$$

where "$\dot{Q}_{loss,shw}$" is the heat loss rate of the sensible heat wheel.

The effectiveness of the sensible heat wheel (ϵ_{shw}) is computed from

$$\epsilon_{shw} = \frac{\dot{m}_4(T_{3'} - T_4)}{\dot{m}_8(T_{3'} - T_8)} \tag{13}$$

3.1.3. Regenerative evaporative cooler (REC)

The dew point effectiveness of the regenerative evaporative cooler ($\epsilon_{dp,rec}$) is written as follows:

$$\epsilon_{dp,rec} = \frac{T_4 - T_5}{T_4 - T_{4,dp}} \tag{14}$$

3.2. Exergy analysis

The general exergy balance of the system can be written as follows:

$$\sum \dot{E}x_{in} = \sum \dot{E}x_{out} + \dot{E}x_{dest} \tag{15}$$

where "$\dot{E}x_{in}$" is exergy input rate, "$\dot{E}x_{out}$" is exergy output rate and "$\dot{E}x_{dest}$" is exergy destruction rate.

The maximum work obtainable from a fluid moving from its current state to the "dead" state ($\dot{E}x_i$) is generally calculated as:

$$\begin{aligned}\dot{E}x_i &= \dot{m}_i\,[(h_i - h_0) - T_0(s_i - s_0)] \\ &= \dot{m}_i\, c_{p,i}\left[(T_i - T_0) - T_0 \ln\left(\frac{T_i}{T_0}\right)\right]\end{aligned} \tag{16}$$

where "T" is temperature, "s" is entropy, "c_p" is specific heat of the fluid. Subscript "0" shows the reference environment condition (dead state). The reference environment temperature and pressure are 35 °C and 1 atm, respectively.

However, exergy of a humid air includes thermal exergy, chemical exergy, and mechanical exergy. The mechanical exergy can be generally neglected for desiccant air cooling systems. So, exergy of air can be determined as follows (Shukuya and Hammache, 2002; Dincer and Rosen, 2013):

$$\begin{aligned}\dot{E}x_{air} &= \dot{E}x_{ch,air} + \dot{E}x_{th,air} = \dot{m}_{air}ex_{ch,air} + \dot{m}_{air}ex_{th,air} \\ &= \dot{m}_{air}(ex_{ch,air} + ex_{th,air}) = \dot{m}_{air}ex_{tot,air}\end{aligned} \tag{17}$$

where "$\dot{E}x_{ch,air}$" is chemical exergy of air, "$\dot{E}x_{th,air}$" is thermal exergy of air, "$ex_{ch,air}$" is specific chemical exergy of air, "$ex_{th,air}$" is specific thermal exergy of air and "$ex_{tot,air}$" is total specific exergy of air ($ex_{ch,air} + ex_{th,air}$).

$$\dot{E}x_{ch,air} = \dot{m}_{air}ex_{ch,air} = \dot{m}_{air}\left[R_a T_0\left[(1+\overline{\omega})\ln\frac{(1+\overline{\omega}_0)}{(1+\overline{\omega})} + \overline{\omega}\ln\frac{\overline{\omega}}{\overline{\omega}_0}\right]\right] \tag{18}$$

$$\dot{E}x_{th,air} = \dot{m}_{air}ex_{th,air} = \dot{m}_{air}\left[(c_{p,air} + \omega\, c_{p,v})\ \left[T - T_0 - T_0 \ln\frac{T}{T_0}\right]\right] \tag{19}$$

where "R_a" is general gas constant (0.287 kJ/kgK), "ω" is the humidity ratio of air, "$c_{p,v}$" is the specific heat of water vapor, and "$\overline{\omega}$"is the mole fraction ratio of air calculated as in (Caliskan et al., 2011a).

$$\overline{\omega} \cong (1.608)\omega \tag{20}$$

The thermally driven exergy COP ($COP_{ex,th}$) of the system is determined by;

$$COP_{ex,th} = \frac{\dot{E}x_{cooling}}{\dot{E}x_{reg}} \tag{21}$$

where "$\dot{E}x_{cooling}$"and "$\dot{E}x_{reg}$" are the exergy cooling capacity rate and regeneration exergy rate of the system, respectively. By considering the heat transfer interaction, they can be explained as follows (Dincer et al., 2014; Eicker, 2014):

$$\dot{E}x_{cooling} = \dot{Q}_{cooling}\left|1 - \frac{T_0}{T_{cooling,av}}\right| \tag{22}$$

$$T_{cooling,av} = \frac{T_1 + T_6}{2} \tag{23}$$

$$\dot{E}x_{reg} = \dot{Q}_{reg}\left|1 - \frac{T_0}{T_{reg,av}}\right| \tag{24}$$

$$T_{reg,av} = \frac{T_9 + T_{10}}{2} \tag{25}$$

Another definitions of the "$\dot{E}x_{cooling}$"and "$\dot{E}x_{reg}$" are based on the fluid (air) flow principle as follows:

$$\dot{E}x_{cooling} = \dot{E}x_6 - \dot{E}x_1 \tag{26}$$

$$\dot{E}x_{reg} = \dot{E}x_{10} - \dot{E}x_9 \tag{27}$$

where the points of "6" and "1" are the supply and return (room) air conditions, while points "9" and "10" are the inlet and outlet air conditions of the regeneration heat unit in the desiccant cooling system as shown in Fig. 1, respectively. In this paper, we use Eqs. (26) and (27) as the exergy cooling capacity rate and regeneration exergy rate of the system, respectively.

The electrically driven exergy COP ($COP_{ex,el}$) of the system is computed as:

$$COP_{ex,el} = \frac{\dot{E}x_{cooling}}{\dot{W}_{tot}} \tag{28}$$

The exergy efficiency of the desiccant air cooling system (Ψ_{sys}) is calculated as

$$\Psi_{sys} = \frac{\dot{E}x_{cooling}}{\dot{E}x_{reg} + \dot{W}_{tot}} \tag{29}$$

Let's characterize each component:

3.2.1. Desiccant wheel (DW)

The exergy balance of the desiccant wheel can be written as;

$$\dot{E}x_2 + \dot{E}x_{10} = \dot{E}x_3 + \dot{E}x_{11} + \dot{E}x_{loss,dw} + \dot{E}x_{dest,dw} \tag{30}$$

where "$\dot{E}x_{loss,dw}$" and "$\dot{E}x_{dest,dw}$" are the exergy loss rate and exergy destruction rate of the desiccant wheel.

$$\dot{E}x_{loss,dw} = \dot{Q}_{loss,dw}\left(1 - \frac{T_0}{T_{dw}}\right) \tag{31}$$

where "T_{dw}" is the average temperature of the desiccant wheel.

$$T_{dw} = \frac{T_{10} + T_{11} + T_2 + T_3}{4} \tag{32}$$

The entropy production rate of the desiccant wheel ($\dot{S}_{dw}$) is calculated from

$$\dot{S}_{dw} = \frac{\dot{E}x_{dest,dw}}{T_0} \tag{33}$$

The exergy efficiency of the desiccant wheel (Ψ_{dw}) is determined from

$$\Psi_{dw} = \frac{\dot{E}x_3 - \dot{E}x_2}{\dot{E}x_{10} - \dot{E}x_{11}} \tag{34}$$

3.2.2. Sensible heat wheel (SHW)

The exergy balance of the sensible heat wheel is computed by:

$$\dot{E}x_{3'} + \dot{E}x_8 = \dot{E}x_4 + \dot{E}x_9 + \dot{E}x_{loss,shw} + \dot{E}x_{dest,shw} \tag{35}$$

where "$\dot{E}x_{loss,shw}$" and "$\dot{E}x_{dest,shw}$" are the exergy loss rate and exergy destruction rate of the sensible heat wheel.

$$\dot{E}x_{loss,shw} = \dot{Q}_{loss,shw}\left(1 - \frac{T_0}{T_{shw}}\right) \tag{36}$$

where "T_{dw}" is the average temperature of the desiccant wheel.

$$T_{shw} = \frac{T_8 + T_9 + T_4 + T_{3'}}{4} \tag{37}$$

The entropy production rate of the sensible heat wheel ($\dot{S}_{shw}$) is found from

$$\dot{S}_{shw} = \frac{\dot{E}x_{dest,shw}}{T_0} \tag{38}$$

The exergy efficiency of the sensible heat wheel (Ψ_{shw}) can be determined by

$$\Psi_{shw} = \frac{\dot{E}x_9 - \dot{E}x_8}{\dot{E}x_{3'} - \dot{E}x_4} \tag{39}$$

3.3. Sustainability analysis

Sustainability assessment is done by Sustainability Index (*SI*) parameter. It gives information about how exergy efficiency is connected with the sustainability. If the exergy efficiency increases, the sustainability index increases too (For detailed information, see Ref. (Dincer and Rosen, 2013)).

$$SI = \frac{1}{1 - \Psi} \tag{40}$$

4. Results and discussion

The solid desiccant cooling system is analyzed point by point through advanced thermodynamics analyses. The energy and exergy analyses results of the system points are given in Table 2 while the energy and exergy rates of the system points are illustrated in Fig. 2.

Detailed information about the results obtained from the energy and exergy analyses are explained below:

(1 + 7')-(2): Return air and outdoor make up air enter to the system with 52.308 kW and 28.868 kW energy rates, respectively (totally 81.176 kW). After the filter, the energy rate of the total air reduces to 81.160 kW. On the other hand, the exergy rate of the total air in the entrance is 0.1398 kW, and it reduced to 0.094 kW after the filter of the system. Hence, small amounts of energy and exergy of the air are lost in the filter.

(2)–(3): Air takes 4.107 kW energy rate and 0.2861 kW exergy rate through desiccant wheel by using the outdoor air in the counter channel. As a result, the energy and exergy rates of the air reach to 85.267 kW and 0.3801 kW, respectively.

(3)-(3′): Air is pushed by a fan to the sensible heat wheel and the energy and exergy rates are constant.

(3′)-(4): Air goes through sensible heat wheel and its energy and exergy rates are reduced to 79.630 kW and 0.2838 kW, respectively. This phenomenon occurs because the air (working fluid) releases part of its energy to the external air through the sensible heat wheel channels.

(4)–(5): Air goes through regenerative evaporative cooler which is a kind of heat and mass exchanger. The energy rate of the air is reduced to 38.020 kW and the corresponding exergy rate reaches to 0.5603 kW. The change in energy and exergy occur because of the heat transfer in the exchanger.

(5)=(6): Air is sent to the conditioned space and both the energy and exergy rates remain constant.

(7)–(8): Outdoor air enters into the second channel of the system through a filter with an energy rate of 67.304 kW and exergy rate of 0.0007 kW.

(8)–(9): Outdoor air goes through sensible heat wheel by taking 5.738 kW energy and 0.0555 kW exergy rates from other channel through wheel. The energy and exergy rates of the outdoor air reach to 73.042 kW and 0.0562 kW, respectively.

(9)–(10): Outdoor air enters to the heating coil and takes 18.578 kW energy and 0.908 kW exergy rates. As a result, it reaches 91.620 kW energy rate and 0.9646 kW exergy rate.

(10)–(11): Outdoor air exit from the system through desiccant wheel and fan by giving its 4.343 kW energy and 0.7461 kW exergy rates to the other channel.

The total fan power, cooling capacity and regeneration heat are found as 2.13 kW, 14.288 kW and 18.578 kW, respectively. Hence, the electrically driven *COP* (COP_{el}) rate is determined to be 6.71,

Table 2
Energy and exergy analyses results of the system points.

Points	Energy rate (kW)	Exergy rate (kW)
1	52.308	0.1395
2	81.160	0.0940
3	85.267	0.3801
3′	85.267	0.3801
4	79.630	0.2838
5	38.020	0.5603
5′	41.613	0.3398
6	38.020	0.5603
7	67.304	0.0007
7′	28.868	0.0003
8	67.304	0.0007
9	73.042	0.0562
10	91.620	0.9646
11	87.277	0.2185

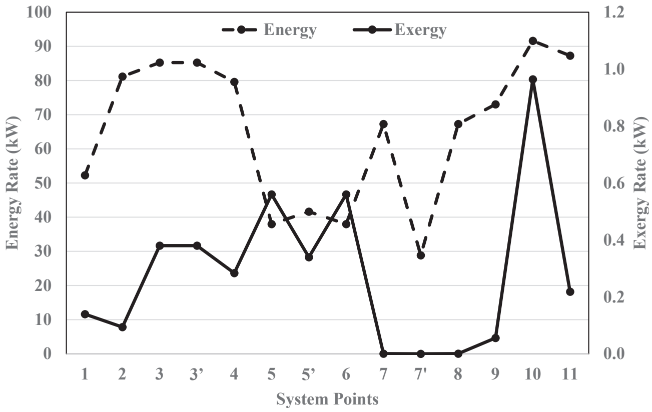

Fig. 2. Energy and exergy rates of the system points.

while thermally driven *COP* (COP_{th}) rate is 0.77. On the other hand, the exergy cooling capacity rate and regeneration exergy rate are calculated as 0.421 kW and 0.908 kW, respectively. The performances of the overall system are given in Fig. 3. The electrically driven exergy *COP* ($COP_{ex,el}$) is about 0.198, while thermally driven exergy *COP* ($COP_{ex,th}$) rate is 0.463. The performance of this system can be considered high if compared with the ones of previous studies. The comparison of the performances with the previous studies is shown in Table 3. The electrically driven *COP* rate of the present study is 0.339 times higher than Tu et al. (2015b), 4.418 times higher than Enteria et al. (2013b) and 0.412 times higher than Jani et al. (2015). The thermally driven *COP* rate of the present study is 0.053 times higher than Xiong et al. (2010). On the other hand, the electrically driven exergy *COP* of the system is 1.357 times higher than Enteria et al. (2013b).

The energy and exergy analyses results of the desiccant wheel and sensible heat wheel are tabulated in Table 4. The heat loss rate of the desiccant wheel and sensible heat wheel are found to be 0.2355 kW and 0.1008 kW, respectively, while the corresponding exergy loss rates are determined to be 0.0061 kW and 0.0013 kW, respectively. The exergy destruction rate of the desiccant wheel (0.4538 kW) is higher than the one of the sensible heat wheel (0.0394 kW). The entropy production of the desiccant wheel (0.00147 kW/K) is almost ten times higher than the sensible heat wheel one (0.00013 kW/K). So, the sensible heat wheel is better than desiccant wheel by thermodynamic point of view. The general effectiveness of the desiccant wheel and sensible heat wheel are found to be 0.6024 and 0.8025, while the exergy efficiencies are 38.35% and 57.69%, respectively. The specific humidity based effectiveness of the desiccant wheel is about 0.33 and it is lower than the general effectiveness of the desiccant wheel. The specific humidity based effectiveness is the best definition for the desiccant wheel because it involves the moisture transfer. On the other hand, the effectiveness of the regenerative evaporative cooler is determined as 0.6695 which is between desiccant wheel and sensible heat wheel rates.

The wet bulb effectiveness and dew point effectiveness of the overall system are calculated to be 0.95 and 0.63, respectively, while exergy efficiency of the overall system is 13.85%. The effectiveness rates of the overall system, desiccant wheel, sensible heat wheel and regenerative evaporative cooler are illustrated in Fig. 4; moreover, exergy efficiencies and sustainability indexes of the overall system, desiccant wheel and sensible heat wheel are shown in Fig. 5. Overall system exergy efficiency and sustainability are lower than the one of the desiccant and sensible heat wheel, while there is no assessment of the heating coil due to the limited data. The sustainability of the overall system, desiccant wheel and sensible heat wheel are found as 1.2255, 1.6220 and 2.3633, respectively. Between the two analyzed components, the desiccant wheel has the minimum exergy efficiency and sustainability.

5. Conclusions

A novel solid desiccant air cooling system with desiccant wheel, sensible heat wheel and evaporative air cooler is designed and assessed by using energy, exergy and sustainability analyses. The data are obtained considering the following environment conditions: 35 °C and 101.3 Pa. The system is different than the previous studies with its design and it is analyzed from an energy, exergy and sustainability point of view. The heating coil data is not taken during the experiment and for this reason its assessment is not presented. Under these circumstances, the following main conclusions can be drawn from this study;

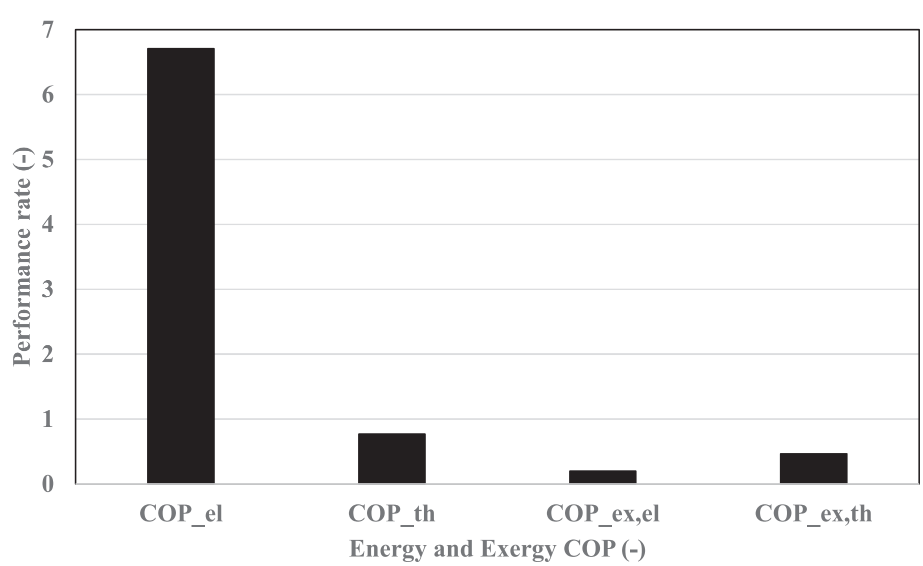

Fig. 3. Performances of the overall system.

Table 3
Comparison of the performances with the previous studies.

	Xiong et al. (2010)	Tu et al. (2015b)	Enteria et al. (2013b)	Jani et al. (2015)	Present study
COP_{th}	0.73	–	–	–	0.769
COP_{el}	–	5.01	1.238	4.75	6.708
$COP_{ex,th}$	–	–	–	–	0.463
$COP_{ex,el}$	–	–	0.084	–	0.198
Desiccant type	Liquid	Solid	Solid	Solid	Solid
Environment temperature	30 °C	33 °C	>30 °C	35	35 °C
Environment humid ratio	18.7 g/kg$_{da}$	19 g/kg$_{da}$	>16 g/kg$_{da}$	23 g/kg$_{da}$	14.1 g/kg$_{da}$

Table 4
Energy and exergy analyses results of the desiccant wheel and sensible heat wheel.

Parameter	Desiccant Wheel (DW)	Sensible Heat Wheel (SHW)
Heat loss rate ($\dot{Q}_{loss}$) (kW)	0.2355	0.1008
Effectiveness (ϵ) (−)	0.6024	0.8025
Effectiveness considering humidity (ϵ_{ω}) (−)	0.33	–
Exergy loss rate ($\dot{E}x_{loss}$) (kW)	0.0061	0.0013
Exergy destruction rate ($\dot{E}x_{dest}$) (kW)	0.4538	0.0394
Exergy efficiency (Ψ) (%)	38.35	57.69
Entropy production rate ($\dot{S}$) (kW/K)	0.00147	0.00013

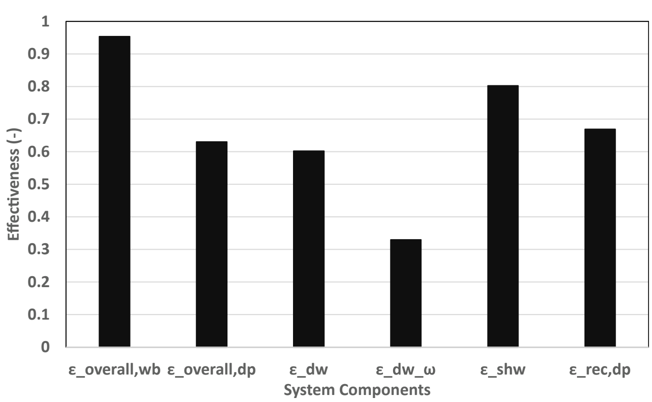

Fig. 4. Effectiveness rates of the overall system (overall), desiccant wheel (dw), sensible heat wheel (shw) and regenerative evaporative cooler (rec).

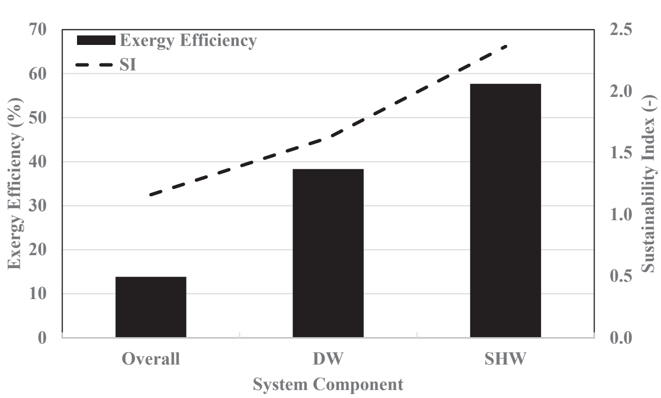

Fig. 5. Exergy efficiencies and sustainability indexes of the overall system (Overall), desiccant wheel (DW) and sensible heat wheel (SHW).

(i) The energy (wet bulb and dew point) effectiveness of the overall system is good enough, but, applying the second law of thermodynamics, the exergy efficiency shows that this system can be largely improved.

(ii) The electrically and thermally driven energy *COP* of the overall system (6.71 and 0.77) are higher than the corresponding exergy *COP* (0.198 and 0.463). Energy *COP* shows that the performance of the overall system is high, however electrically driven exergy *COP* shows that some modifications can be realized in order to improve the final performance of the system.

(iii) Among the components, desiccant wheel should be improved. Because 42.87% of its exergy is destructed (0.4538 kW), while its exergy efficiency is about 38.35%. On the other hand, sensible heat wheel exergy destruction ratio is 10.35%, and its exergy efficiency is 57.69%.

(iv) Entropy production of desiccant wheel (0.00147 kW/K) is ten times higher than the one of sensible heat wheel (0.00013 kW/K). It shows that the desiccant wheel is not working as reliable component as the sensible heat wheel; this fact suggests that it may be necessary to change that component with a newer and more efficient one.

(v) Sustainability of the sensible heat wheel (2.36) is 46% higher than the desiccant wheel (1.62). Among the components, desiccant wheel has less sustainability than others.

(vi) The effectiveness of the components from maximum to minimum are as sensible heat wheel, regenerative evaporative cooler and desiccant wheel, respectively. For better effectiveness, the component devices should be working around similar high level. So, the urgent improvement for the devices are in the order of desiccant wheel, regenerative evaporative cooler and sensible heat wheel, respectively.

Implications for theory and practice: In the theory, there may be many applications to arrange the desiccant air cooling systems. But, it is limited to show their applicability in real life. In this study theory and practice come together. First, a new desiccant air cooling system is designed theoretically and then it is practically operated as a real experimental system. The results show that the performance of this desiccant air cooling system is very promising.

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