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Exergy analysis of a novel configuration of desiccant based evaporative air conditioning system

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ABSTRACT

In this work, a process is developed for exergy analyses of a novel configuration of desiccant based an evaporative air conditioning system. The exergy transfer and destruction between the components of the system are defined for the average measured variables obtained from the experimental results. The exergy formulations are carried out to the experimental system using the data collected during a typical operation of the system. The exergy output, specific flow exergy, exergy destruction, exergy input and exergy efficiency are determined. Furthermore, the sustainability assessment and relative irreversibility of components are obtained. It is found that the exergy efficiency of the entire experimental unit is 40.7% at a reference temperature of 15 °C. It is also observed that the exergy efficiencies of the entire system varies between 56% and 25% for reference temperature of 0–30 °C, respectively. The effects of reference temperature on the performance of the studied system are investigated. Based on the investigation, it is seen that an exergy analysis can provide beneficial knowledge with respect to the theoretical upper limit of the system performance.

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1. Introduction

Desiccant air conditioning systems have been presented as interesting alternatives to conventional vapor compression systems owing to their advantages of utilizing low temperature energy. Generally, desiccant air conditioning systems are a composition of desiccant dehumidification, additional compression refrigeration, evaporative cooling and the regeneration of the desiccant [1]. In a desiccant based air conditioning system, outdoor air passes firstly through in solid desiccant dehumidifier that is main component of this system. After dehumidification, air that is dried can be firstly cooled by a sensible heat exchanger and for further cooling an evaporative cooler are used. Evaporative air cooling has been used for thousands of years in various forms for comfort cooling and is still in common use around the world because of its simplicity, low cost and effectiveness. On the other hand, evaporative cooling system cannot be used alone in humid climate due to raises the humidity of process air. So, this system should be combined with desiccant air conditioning system in hot and humid climate.

Evaporative cooling technologies were either utilized as direct, indirect or direct/indirect. A schematic diagram of two stage indirect–direct evaporative cooling system can be seen in Fig. 1. In evaporative cooling systems, air is drawn through evaporative cooler and its sensible heat energy evaporates water; the heat and mass transfer between the air and water decreases the air dry bulb temperature and increases the humidity at a constant wet-bulb temperature. Main components of basic evaporative cooling system occur from pumps (P), fans (F) and evaporative cooler that can be direct (DEC) or indirect (IEC).

To analyze, design and improve the solid desiccant air conditioning systems a number of works are given in the literature. Some of these works are focus on the optimization and analysis of operating parameters for the whole system [2,3]. Bourdoukan et al. [4] studied a desiccant air handling unit powered by vacuum-tube solar collectors and they first worked the performance of the components under various operating conditions. Then overall performance of the installation evaluated by authors.

Desiccant wheel is one of important components in desiccant air conditioning systems and many investigations have been performed on it [5–7]. Jia et al. [8] carried out experiments on a hybrid desiccant air conditioning system. The system is a combination of two units that are a solid desiccant wheel and a vapor compression air conditioning system. They found that the system reduces 37.5% electricity powers when the process air temperature and relative humidity are held at 30 °C and 55%, respectively.

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Nomenclatures

c_p	specific heat (kJ/kg K)
ex	specific exergy (kJ/kg)
$\dot{E}x$	exergy rate (kW)
f_i	exergetic factor
h	specific enthalpy (kJ/kg)
$I\dot{P}$	rate of improvement potential (kW)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}_j$	the heat transfer rate at location j (kW)
s	specific entropy (kJ/kg K)
T_j	boundary temperature at location j (K)
W	rate of work (kW)

Greek symbol

ψ	specific flow exergy (kJ/kg),
ε	efficiency
δ	fuel depletion ratio
ξ	productivity lack
ω	specific humidity ratio (kg_{water}/kg_{air})

Subscripts

a	air
$dest$	destruction
f	fresh air
i	the ith device
in	input
out	output
R	regeneration air
v	vapor
w	waste air
0	dead reference state
1, 2, . . . , 17	state numbers

Superscripts

Ch	chemical energy
Ke	kinetic energy
Pe	potential energy
Ph	physical energy

Fatouh et al. [9] presented experimental performance data of a solid desiccant based hybrid air conditioning system. The system occurs a solid desiccant wheel combined with a R407C vapor compression air conditioning unit. They analyzed the effect of different air flow rates and regeneration temperatures on reactivation of the desiccant. Ruivo et al. [10] derived new correlations to calculate effectiveness parameters of a solid desiccant dehumidifier. Eicker et al. [11] studied experimental investigations on several commercially available and newly produced rotors. They found that the optimum rotation speed is lower for lithium chloride or compound rotors than that of silica gel rotors.

Exergy which is the maximum useful work possible during a process is now being increasingly recognized in the area of air conditioning system. To optimize and analyzes these types of energy systems, exergy analysis is one of the most effective methods. The detailed information about exergy analysis can be found in Refs. [12,13]. Hürdoğan et al. [14] investigated a desiccant air conditioning system that had a novel configuration and they defined the exergy transfer between the system components. Alpuche et al. [15] analyzed the effect of desiccant air conditioning systems on reaching thermal comfort conditions in the interior of a building. They carried out exergy analysis for the different design temperatures and relative humidity conditions. Kanoğlu et al. [16] established a desiccant cooling system and developed a method for the exergy and energy analyses. Taufiq et al. [17] defined the optimization and modeling analysis for cooling system in the building. La et al. [18] investigated the effects of each components of a desiccant cooling system. They derived appropriate methodology for thermodynamic assessment and developed a generalized model independent of the connection of components.

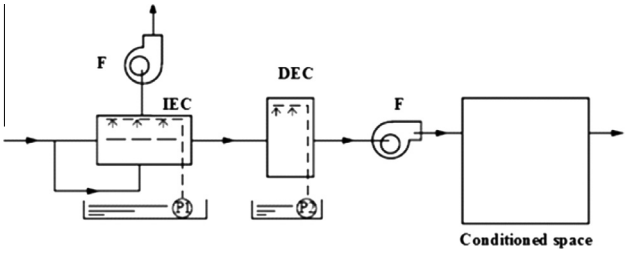


Fig. 1. A schematic diagram of two stage indirect–direct evaporative cooling system.

In the literature survey, many theoretical and experimental studies performed on the first law of desiccant air conditioning systems but very few studies were performed on the exergy analysis of these type systems.

The system described in this paper has a new configuration based on fabrication/assembly. The objective of this study is to perform exergy analyses of desiccant based evaporative air conditioning system [19] with a complex configuration which consists of evaporative coolers, rotary solid desiccant wheel, heat exchangers, electric heater unit, filters, pumps, control units, channels and fans. It is obvious that without detailed calculations it is not possible to predict energy consumption, so, exergy analysis of this type system should be done. In order to gain a more detailed insight to the origin of the energy analysis in this system, exergy analysis of each component has been calculated by means of energy and exergy balance. The effects of the individual irreversible processes in each component on the thermodynamic performance of the system are also analyzed. Moreover, specific flow exergy, exergy efficiency sustainability assessment and relative irreversibility of components are obtained.

2. System description

2.1. Experimental setup and operation

Fig. 2 shows experimentally investigated desiccant air conditioning system that has a novel configuration which consists of dehumidification, heat and cool recovery and evaporative cooling. Experimental setup includes two direct evaporative coolers, one rotary solid desiccant wheel, three heat exchangers, electric heater unit, filters, pumps, control units, channels and fans.

Based on the literature survey, it is seen that waste air from conditioned space is generally used for regeneration in the desiccant conditioning systems. The desiccant based air conditioning design needs new configuration for optimizing the operation of the system. In this study, at the beginning of the system design, an analysis is performed in order to maximize the performance of the system. It is found that outdoor air must be used for regeneration air to decrease consumption of the regeneration heat and

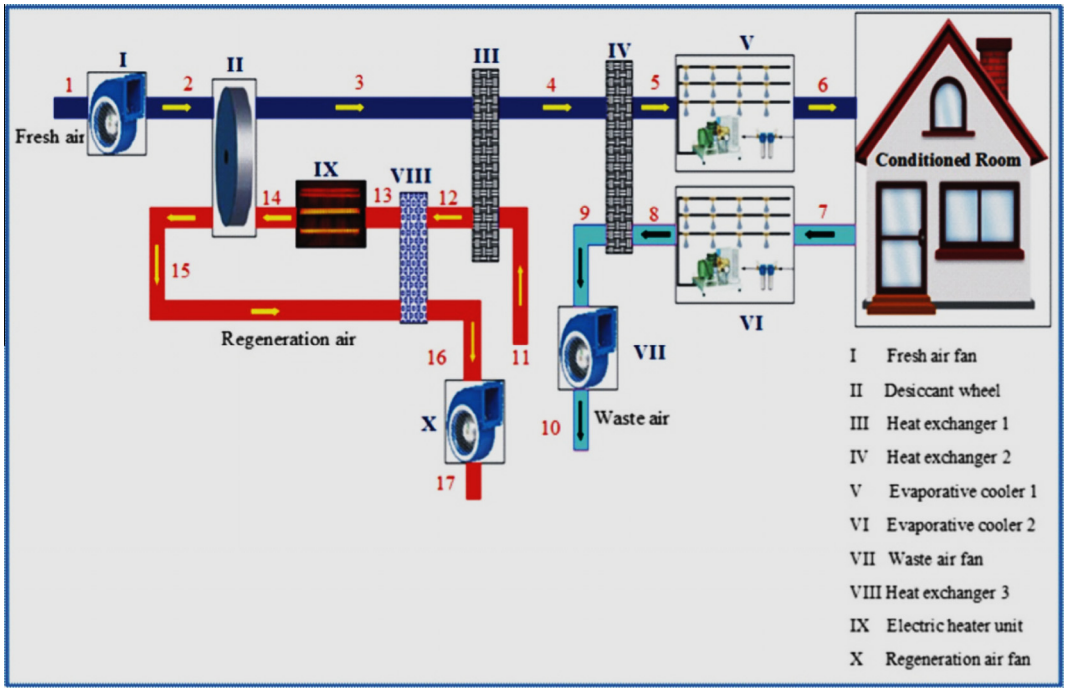


Fig. 2. A schematic view of desiccant based evaporative air conditioning system.

to increase the capacity of desiccant wheel due to lower inlet absolute humidity and higher ambient air temperature. The system has a novel design with regards to both air channels and heat exchangers used. Regeneration air is taken from outdoor and a rotary regenerative type heat exchanger, which is not common to this type of systems, is used for pre-heating the regeneration air with exhaust air. Even the places of the ventilators are carefully selected.

In this system, the diameter and depth of desiccant wheel are main parameters whose values are 965 mm and 200 mm, respectively. The rate of regeneration area to process area in dehumidifier is equal to each other (50–50%). In this system three sensible heat exchangers are used. One of these heat exchangers is rotary regenerator type, the others are recuperative type. The depth and diameter of the rotary regenerator are 200 mm and 965 mm, respectively. The working principle of the components and system were well described in the [19].

This system has three main air channels that are illustrated in Fig. 2. The fresh air is taken from outdoor air by fresh air channel and it is sent into the air-conditioned room. The exhaust air is evacuated from the air conditioning room by a waste air fan in the second channel. For regeneration air stream, the third channel is utilized. A control unit is used to control the air flow rates and regeneration temperature. Filters are used at the inlet of regeneration air channel and fresh air channel. Dehumidified air firstly passes through the heat exchanger 1(HE1) and the heat exchanger 2(HE2) used for further pre-cooling. After dehumidified air passes through the HE2, it is sent into evaporative cooler to cool the fresh air to the supply temperature.

2.2. Uncertainty analysis

The uncertainty analysis is a useful method for identification of corrective actions while validating and performing experiments [20]. In this study, the root of the sum squares is used for the calculation of uncertainty [21]. The temperature, relative humidity, flow rate, electric current and electric potential differences are measured with convenient equipment. The total uncertainties of the measurements are calculated to be ±2.51% for the relative humidity, ±2.89 for the mass flow rates, ±0.3 °C for the air temperatures, ±1.59% for the power inputs to the system, while those with the exergy input/output rates and exergy efficiencies are calculated to be on average 3.51% and 4.83%, respectively.

3. Model analysis

The work potential of a system at a given state may be evaluated by letting the system proceed towards and actually reach a stable equilibrium with the environment. In fact, when the system and the environment are in equilibrium, no further change of state can occur spontaneously and hence no further work can be produced. When such a condition occurs, the system is said to be in the dead state [22]. At the dead state, the conditions of thermal, mechanical and chemical balance between the environment and system are satisfied: the temperature, pressure, and chemical potentials of the system equal those of the environment, respectively. The dead state temperature is taken as 15 °C and for selection of dead state one can see Refs. [23,25,29]. The value of dead state specific humidity ratio is considered as daily mean value of ambient air.

Energy and mass balances as well as exergy destructions obtained from exergy balances for each of the desiccant based evaporative air conditioning system components shown in Fig. 2 are derived as follows. To deeply investigation of derivations of equations related to exergy analysis, one can see Ref. [24].

In a system, the total exergy $\dot{E}x$ consist of four parameters that are potential exergy $\dot{E}x^{pt}$ physical exergy $\dot{E}x^{ph}$, chemical exergy $\dot{E}x^{ch}$ and kinetic exergy $\dot{E}x^{kn}$ [25].

$$\dot{E}x = \dot{E}x^{ph} + \dot{E}x^{kn} + \dot{E}x^{pt} + \dot{E}x^{ch} \quad (1)$$

In a system, the total specific exergy on a mass basis can be given as follows,

$$ex = ex^{ph} + ex^{kn} + ex^{pt} + ex^{ch} \quad (2)$$

The general exergy balance may be given as follows:

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{dest} \tag{3}$$

where $\dot{E}x_{in} - \dot{E}x_{out}$ indicates the amount of net exergy transfer by work, heat and mass and $\dot{E}x_{dest}$ indicates the amount of net exergy destruction. The general exergy balance may also present as:

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{dest} \tag{4}$$

where;

$$\dot{E}x_{heat} = \sum \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j, \tag{5}$$

$$\dot{E}x_{work} = \dot{W}, \tag{6}$$

$$\dot{E}x_{mass,in} = \sum \dot{m}_{in}\psi_{in}, \tag{7}$$

$$\dot{E}x_{mass,out} = \sum \dot{m}_{out}\psi_{out}, \tag{8}$$

where $\dot{Q}_j$ is the heat transfer rate through the boundary at temperature T_j at location j

$\dot{W}$ is the work rate, ψ is the specific flow exergy.

The specific flow exergy of water may be defined as follows;

$$\psi = (h - h_0) - T_0(s - s_0) \tag{9}$$

and the total flow exergy of air may be given as [26,27];

$$\begin{aligned}\psi_a = {} & (C_{p,a} + \omega C_{p,v})T_0\left[\left(\frac{T}{T_0}\right) - 1 - ln\left(\frac{T}{T_0}\right)\right] \\ & + (1 + 1.6078\omega)R_aT_0 ln\left(\frac{P}{P_0}\right) \\ & + R_aT_o\left\{(1 + 1.6078\omega)ln\left[\frac{1 + 1.6078\omega_0}{1 + 1.6078\omega}\right] + 1.6078\omega ln\left(\frac{\omega}{\omega_0}\right)\right\}\end{aligned} \tag{10}$$

where the subscript "0" describe properties at the reference (dead) state of P_0 and T_0, h is the enthalpy, ω is the specific humidity ratio of supply inlet air, s is the entropy and $c_{p,v}$ is the specific heat of water vapor.

The specific humidity ratio is;

$$\omega = \frac{\dot{m}_v}{\dot{m}_a} \tag{11}$$

Exergy efficiency (ε) can be written as:

$$\varepsilon = \frac{\text{Exergetic product}}{\text{Exergetic fuel}} = \frac{\dot{P}}{\dot{F}} \tag{12}$$

where exergetic fuel is the exergy of the ingoing energy flows that can be considered to be necessary for making the product in the process. Depletion of fuel is also conceived as loses of ingoing exergy in the process. Further, there are some other thermodynamic parameters, which are useful for the assessment of the system. One of these parameters is Van Gool's improvement potential. Van Gool [28] has suggested that maximum improvement in the exergy efficiency for a process or system is obtained when the exergy loss or irreversibility $\left(\dot{E}x_{in} - \dot{E}x_{out}\right)$ is minimized. Consequently, he proposed that it is useful to employ the concept of an exergetic improvement potential $I\dot{P}$ when analyzing different processes or sectors of the economy. It is may be written as;

$$I\dot{P} = (1 - \varepsilon)\left(\dot{E}x_{in} - \dot{E}x_{out}\right) \tag{13}$$

The other parameters are sustainability index, fuel depletion ratio, relative irreversibility, exergetic factor and productivity lack. These factors are a proportional magnitude and dimensionless.

To improve the performance of a system, exergy analysis provides an opportunity. Sustainability index (SI) is another parameter which gives some indications how exergy efficiency affects the sustainability of the resources or systems. It is directly related to exergy efficiency. So, it represents how exergy methods are essential in improving efficiency that allows society to maximize the benefits [29].

$$SI = \frac{1}{1 - \varepsilon} \tag{14}$$

The relative irreversibility, denoted RI, performs the assessment of the ith component exergy destruction with the total process exergy destruction and may be written as:

$$RI = \frac{\dot{E}x_{dest,i}}{\dot{E}x_{dest,T}} \tag{15}$$

The fuel depletion ratio is defined as the ratio of the exergy destruction ith component to the total fuel exergy rate entering the system and can be calculated as;

$$\delta = \frac{Ex_{dest,i}}{\dot{F}_{Tot}} \tag{16}$$

The productivity lack is expressed as the ratio of the exergy destruction of ith component to the exergy of useful products. It can be presented by;

$$\xi = \frac{Ex_{dest,i}}{\dot{P}_{Tot}} \tag{17}$$

The exergetic factor deals with the fuel exergy rate and it is calculated as the ratio of the fuel exergy rate of ith component to the exergy rate of the total process.

$$f_i = \frac{\dot{F}_i}{\dot{F}_{tot}} \tag{18}$$

where the subscript 'i' denotes the ith device.

In this study some assumptions are performed through the analyses; (a) the effects of kinetic and potential energy are considered to be negligible and no chemical reactions, (b) the system is assumed to be a steady-state and steady flow, (c) work transfer from the system and heat transfer to the system are considered as positive, (d) air is accepted as an ideal gas with a constant specific heat, (e) motor electrical efficiency of heat exchanger 3($\eta_{he3,elec}$) and desiccant dehumidifier ($\eta_{dew,elec}$) areconsidered to be 88%. Energy, exergy and mass balances for each of the system components can be expressed as follows:

Fresh air fan (I):

Mass balance:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_f \tag{19}$$

Energy balance:

$$\dot{W}_{ffan} = \dot{m}_f(h_2 - h_1) \tag{20}$$

Exergy balance:

$$\dot{E}x_{dest,ffan} = \dot{m}_f(\psi_1 - \psi_2) + \dot{W}_{ffan,elec} \tag{21}$$

$$ffan = \frac{\dot{E}x_2 - \dot{E}x_1}{\dot{W}_{ffan,elec}} = \frac{\dot{m}_f(\psi_2 - \psi_1)}{\dot{W}_{ffan,elec}} \tag{22}$$

Desiccant wheel (II):

Desiccant wheel as shown in Fig. 2 is one of the most important part of the desiccant based evaporative air conditioning system. Air is circulated through a bed of adsorptive material that is silica gel. As the moist air passes through the bed, it gives up water vapor to the desiccant. Then the saturated desiccant is heated and this releases moisture to a different air stream that regeneration air

stream in this work. The exergy of this part degrades due to the irreversibility and is calculated by;

Mass balance:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_f; \dot{m}_{14} = \dot{m}_{15} = \dot{m}_r \tag{23}$$

Energy balance:

$$\dot{Q}_{dew} = \dot{m}_f(h_3 - h_2) = \dot{m}_r(h_{14} - h_{15}) \tag{24}$$

$$\dot{W}_{dew,elec} = \frac{\dot{W}_{dew}}{\eta_{dew,elec}} \tag{25}$$

Exergy balance:

$$\dot{E}x_{dest,dew} = \dot{m}_f(\psi_2 - \psi_3) + \dot{m}_r(\psi_{14} - \psi_{15}) + \dot{W}_{dew,elec} \tag{26}$$

$$dew, ex = \frac{\dot{E}x_3 - \dot{E}x_2}{\dot{E}x_{14} - \dot{E}x_{15} + \dot{W}_{dew,elec}} = \frac{\dot{m}_f(\psi_3 - \psi_2)}{\dot{m}_r(\psi_{14} - \psi_{15}) + \dot{W}_{dew,elec}} \tag{27}$$

Heat Exchanger 1 (III):

A heat exchanger is used to pass energy from one fluid, to another one. The energy is passed in the form of heat, which flow from one to the other spontaneously due to the temperature difference between both fluids. This energy transfer carries some exergy in the meanwhile the hot fluid releases exergy to the cold fluid [30].

The exergy balance of this part can be calculated by;

Mass balance:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_f; \ \dot{m}_{11} = \dot{m}_{12} = \dot{m}_r \tag{28}$$

Energy balance:

$$\dot{Q}_{he1} = \dot{m}_f(h_3 - h_4) = \dot{m}_r(h_{12} - h_{11}) \tag{29}$$

Exergy balance:

$$\dot{E}x_{dest,he1} = \dot{m}_f(\psi_3 - \psi_4) + \dot{m}_r(\psi_{11} - \psi_{12}) \tag{30}$$

$$he1, ex = \frac{\dot{E}x_{12} - \dot{E}x_{11}}{\dot{E}x_3 - \dot{E}x_4} = \frac{\dot{m}_r(\psi_{12} - \psi_{11})}{\dot{m}_f(\psi_3 - \psi_4)} \tag{31}$$

Heat Exchanger 2 (IV):

Mass Balance

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_f; \ \dot{m}_8 = \dot{m}_9 = \dot{m}_w \tag{32}$$

Energy balance:

$$\dot{Q}_{he2} = \dot{m}_f(h_4 - h_5) = \dot{m}_w(h_9 - h_8) \tag{33}$$

Exergy balance:

$$\dot{E}x_{dest,he1} = \dot{m}_f(\psi_4 - \psi_5) + \dot{m}_r(\psi_8 - \psi_9) \tag{34}$$

$$he2, ex = \frac{\dot{E}x_9 - \dot{E}x_8}{\dot{E}x_4 - \dot{E}x_5} = \frac{\dot{m}_w(\psi_9 - \psi_8)}{\dot{m}_f(\psi_4 - \psi_5)} \tag{35}$$

Evaporative cooler 1 (V):

The main principle of evaporative cooling is the conversion of sensible heat to latent heat. When hot air passes through evaporative cooler, heat is absorbed by water sprayed through the sprinklers to bring down the temperature of the moist air. So, the air leaves the system at a lower temperature [16]. Consequently, the humidification process in the evaporative coolers involves irreversibility caused by concentration difference and mass transfer. The methodology of exergy and energy of the evaporative cooler is presented as:

Mass balance:

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_f \tag{36}$$

Energy balance:

$$\dot{m}_f h_5 + \dot{m}_{water,ec1} h_{water,ec1} = \dot{m}_f h_6 \tag{37}$$

Exergy balance:

$$\dot{E}x_{dest,ec1} = \dot{m}_f(\psi_5 - \psi_6) + \dot{m}_{water,ec1}\psi_{water,ec1} \tag{38}$$

$$ec1, ex = \frac{\dot{E}x_6}{\dot{E}x_5 + \dot{E}x_{water,ec1}} = \frac{\dot{m}_f\psi_6}{\dot{m}_f\psi_5 + \dot{m}_{water,ec1}\psi_{water,ec1}} \tag{39}$$

Evaporative cooler 2 (VI):

Mass balance:

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_w \tag{40}$$

Energy balance:

$$\dot{m}_w h_7 + \dot{m}_{water,ec2} h_{water,ec2} = \dot{m}_w h_8 \tag{41}$$

Exergy balance:

$$\dot{E}x_{dest,ec2} = \dot{m}_w(\psi_7 - \psi_8) + \dot{m}_{water,ec2}\psi_{water,ec2} \tag{42}$$

$$ec2, ex = \frac{\dot{E}x_8}{\dot{E}x_7 + \dot{E}x_{water,ec2}} = \frac{\dot{m}_w\psi_8}{\dot{m}_w\psi_7 + \dot{m}_{water,ec2}\psi_{water,ec2}} \tag{43}$$

Waste air fan (VII):

Mass balance:

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_w \tag{44}$$

Energy balance:

$$\dot{W}_{wfan} = \dot{m}_w(h_{10} - h_9) \tag{45}$$

Exergy balance:

$$\dot{E}x_{dest,wfan} = \dot{m}_w(\psi_9 - \psi_{10}) + \dot{W}_{wfan,elec} \tag{46}$$

$$wfan, ex = \frac{\dot{E}x_{10} - \dot{E}x_9}{\dot{W}_{wfan,elec}} = \frac{\dot{m}_w(\psi_{10} - \psi_9)}{\dot{W}_{wfan,elec}} \tag{47}$$

Heat exchanger 3 (VIII):

Mass balance:

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{15} = \dot{m}_{16} = \dot{m}_r \tag{48}$$

Energy balance:

$$\dot{Q}_{he3} = \dot{m}_r(h_{13} - h_{12}) = \dot{m}_r(h_{15} - h_{16}) \tag{49}$$

$$\dot{W}_{he4,elec} = \frac{\dot{W}_{he4}}{\eta_{he4,elec}} \tag{50}$$

Exergy balance:

$$\dot{E}x_{dest,he3} = \dot{m}_r(\psi_{12} - \psi_{13}) + \dot{m}_r(\psi_{15} - \psi_{16}) + \dot{W}_{he4,elec} \tag{51}$$

$$he3 = \frac{\dot{E}x_{12} - \dot{E}x_{13}}{\dot{E}x_{15} - \dot{E}x_{16} + \dot{W}_{he3,elec}} = \frac{\dot{m}_r(\psi_{12} - \psi_{13})}{\dot{m}_r(\psi_{15} - \psi_{16}) + \dot{W}_{he3,elec}} \tag{52}$$

Electric heater unit (IX):

Mass balance:

$$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_r \tag{53}$$

Energy balance:

$$\dot{Q}_{EH} = \dot{m}_r(h_{14} - h_{13}) \tag{54}$$

Exergy balance:

$$\dot{E}x_{dest,EH} = \dot{m}_r(\psi_{13} - \psi_{14}) + \dot{W}_{EH} \tag{55}$$

$$EH = \frac{\dot{E}x_{14} - \dot{E}x_{13}}{\dot{W}_{EH}} = \frac{\dot{m}_r(\psi_{14} - \psi_{13})}{\dot{W}_{EH}} \tag{56}$$

Regeneration air fan (X):

Mass balance:

$$\dot{m}_{16} = \dot{m}_{17} = \dot{m}_r \tag{57}$$

Energy balance:

$$\dot{W}_{rfan} = \dot{m}_r(h_{17} - h_{16}) \tag{58}$$

Exergy balance:

$$\dot{E}x_{dest,rfan} = \dot{m}_r(\psi_{16} - \psi_{17}) + \dot{W}_{rfan,elec} \tag{59}$$

$$rfan = \frac{\dot{E}x_{17} - \dot{E}x_{16}}{\dot{W}_{rfan,elec}} = \frac{\dot{m}_w(\psi_{17} - \psi_{16})}{\dot{W}_{rfan,elec}} \tag{60}$$

4. Results and discussion

In this study, a desiccant air conditioning system with a novel configuration is comprehensively analyzed using exergy formulations derived in this paper and the analysis results is given in Table 1.

The regeneration, waste and fresh air streams have equal volume flow rates of 2000 m^3/h in this work. The regeneration temperature is adjusted to a fixed temperature of 90 °C during the experiments. The measurement taken between the hours 8:30 and 18:30 is given in the paper as a typical example to the experiments done. In this work, the dead state temperature and outdoor air temperature is assumed as 15 °C and 32 °C, respectively. A parametric study is performed at the dead state temperatures ranging from 0 °C to 30 °C to investigate the effect of the varying dead state temperatures on the exergy efficiency of the system. According to their state numbers specified in Fig. 2, temperature, pressure and mass flow rate data for air and water are given in Table 1. The exergy rates and specific exergy are also calculated for each state as presented in Table 1. The other key parameters on the performance of the proposed desiccant based evaporative cooling system are given in Table 2. These parameters are the exergy destruction rate ($\dot{E}x_{dest}$), improvement potential rate ($I\dot{P}$), product exergy (P), fuel exergy (F), fuel depletion rate (δ), productivity lack (ζ), exergetic factor (f).

As can be seen from Table 2, the electric heater unit has the greatest exergy destruction rate (15.899 kW). The major reason causing the exergy destruction in this system depends on heat input. The experimental system uses electrical energy as the heat source since it is the most convenient method of heat input for the experiments and easy to control. As expected the regeneration temperature is an important parameter for the desiccant dehumidification system. So, the electric heater unit is responsible for the high percentage of the total exergy destruction. The same trend can be seen in Enteria [31], which shows that system thermal source is the main contributors to exergy destruction. After the electric heater, the other important component causing exergy

Table 1
Exergy analysis results of the desiccant based evaporative air conditioning system.

State no	Description	Fluid	Phase	Temperature T (°C)	Pressure P (kPa)	Specific humidity ratio ω (kg_{water}/kg_{dryair})	Specific enthalpy h (kJ/kg)	Mass flow rate $\dot{m}$ (kg/s)	Specific exergy ψ (kJ/kg)	Exergy rate $\dot{E}x$ (kW)
0	–	Moist air	Dead state	15	101.325	0.0154	–	–	0	0
0	–	Water	Dead state	15	101.325	–	63	–	0	0
1	Fresh air fan inlet (Outdoor)	Air	Gas	32.02	101.325	0.01526	71.28	0.64	0.01	0.310
2	Fresh air fan outlet, Dehumidifier inlet	Air	Gas	33.71	101.325	0.01526	73.02	0.64	0.025	0.373
3	Dehumidifier outlet, Heat ex.1 inlet	Air	Gas	63.79	101.325	0.00783	84.66	0.64	2.133	2.544
4	Heat ex.1outlet, Heat ex.2 inlet	Air	Gas	41.50	101.325	0.00783	61.92	0.64	0.556	0.922
5	Heat ex.2 outlet, Evaporative cooler1 inlet	Air	Gas	27.64	101.325	0.00783	47.78	0.64	0.348	0.368
6	EC1 outlet, Air conditioned room inlet	Air	Gas	17.80	101.325	0.01259	49.79	0.64	0.308	0.037
7	Air conditioned room outlet, EC2 inlet	Air	Gas	27.37	101.325	0.01409	63.46	0.64	0.025	0.172
8	EC2 outlet, Heat ex.2 inlet	Air	Gas	22.32	101.325	0.01696	65.56	0.64	0.108	0.062
9	Heat ex.2 outlet, Waste air fan inlet	Air	Gas	32.82	101.325	0.01696	76.44	0.64	0.019	0.342
10	Waste air fan outlet	Air	Gas	33.32	101.325	0.01696	76.96	0.64	0.024	0.361
11	Heat ex.1 inlet (Outdoor),	Air	Gas	32.02	101.325	0.01527	71.28	0.64	0.008	0.310
12	Heat ex.2 outlet, Heat ex. 3 inlet	Air	Gas	49.12	101.325	0.01527	88.95	0.64	0.601	1.198
13	Heat ex.3 outlet, Electrical heater inlet	Air	Gas	50.42	101.325	0.01527	90.29	0.64	0.683	1.288
14	Electrical heater outlet, Dehumidifier inlet	Air	Gas	90.37	101.325	0.01527	131.58	0.64	5.525	5.397
15	Dehumidifier outlet, Heat ex.3 inlet	Air	Gas	47.18	101.325	0.02222	104.92	0.64	0.649	1.175
16	Heat ex.3 outlet, Regeneration fan inlet	Air	Gas	47.35	101.325	0.02222	105.10	0.64	0.659	1.186
17	Regeneration air fan outlet	Air	Gas	48.13	101.325	0.02222	105.91	0.64	0.704	1.237

Table 2
Exergetic analysis of each components.

Device number	Device	Exergy destruction rate, $\dot{E}_{x_{dest}}$ (kW)	Product exergy, P (kW)	Fuel exergy, F (kW)	Exergy improvement potential rate, $I\dot{P}$	Fuel depletion rate, δ (%)	Productivity lack, ζ (%)	Exergetic factor, f (%)
I	Fresh air fan	0.486	0.063	0.549	0.430	1.479	3.630	1.670
II	Dehumidifier	4.339	3.719	5.949	0.836	13.193	32.389	18.089
III	Heat exchanger 1	0.733	2.120	2.854	0.188	2.230	5.474	8.677
IV	Heat exchanger 2	0.274	0.711	0.984	0.076	0.832	2.043	2.993
V	Evaporative cooler 1	1.109	0.037	0.477	0.406	3.373	8.280	1.450
VI	Evaporative cooler 2	0.219	0.062	0.281	0.171	0.666	1.634	0.854
VII	Waste air fan	0.081	0.019	0.100	0.066	0.246	0.604	0.304
VIII	Heat exchanger 3	0.119	2.474	2.593	0.005	0.361	0.887	7.883
IX	Electrical heater unit	15.899	4.109	18.434	11.132	48.342	118.681	56.050
X	Regeneration air fan	0.616	0.083	0.667	0.512	1.873	4.600	2.029
I–X	Overall system	23.875	13.396	32.888	11.552	72.595	178.222	100

destruction is the desiccant wheel (4.339 kW). Van Den Bulck et al. [32] identified the causes of exergy destruction for the desiccant wheel as the mixing of process and regeneration air streams, the transfer of energy and mass across finite temperature, and vapor pressure differences between the desiccant matrix and the regeneration air stream. The same trend is also shown by Kodama [33]. Exergy destruction of the other component is smaller than that of desiccant wheel followed EC1, HE1, regeneration air fan, fresh air fan, HE2, EC2, HE3 and waste air fan, respectively. As illustrated in Table. 2 the exergy destruction of the fans is very small when compared with the electric heater unit and the exergy destruction of the regeneration air fan is higher than that of the fresh air fan and waste air fan. The main reason of this is that the fan power requirement caused exergy destruction increases as the frictional resistance increases in the air channel to maintain the air flow rate. The exergy destruction of evaporative coolers is very low and exergy destruction of EC1 is higher than that of EC2. The cause of this is that there is a greater rate of water evaporation in EC1 resulting in higher irreversibility.

One of the other performance data for the experimental system is Van Gool's improvement potential ($I\dot{P}$) given in Eq. (13). This is calculated for each component of the system using the values listed in Table 2. As seen in Table 2, the electric heater unit has the highest $I\dot{P}$ value of 11.132 kW. The electric heater unit has the most potential for improvement and it is followed by the dehumidifier, fresh air fan, EC, HE1. The improvement potential of the other components is very small. In this system, the regenerating heat can be improved by using solar energy, waste heat and natural gas instead of electrical energy.

Table 2 also shows the thermodynamic parameters such as fuel depletion rate, productivity lack and exergetic factor. According to this table, the electric heater unit has the largest fuel depletion rate with 48.342%, while the dehumidifier, EC1, HE1, regeneration air fan, and fresh air fan are 13.193%, 3.373%, 2.230%, 1.873%, 1.479%, respectively. The rate of other units is smaller than 1%.

Fig. 3 illustrates the relative irreversibility of components in the whole system. The results show that the maximum relative irreversibility occurs in the electric heater unit (66.59%) and in the desiccant wheel (18.17%). Because of more exergy destruction of electrical heater unit, improvement is necessary for it. If renewable energy such as solar energy can be used, the overall system efficiency would increase and relative irreversibility of heating unit will decrease. On a less important level, the relative irreversibility occurs in the EC1 (4.65%), HE1 (3.07) and regeneration air fan (2.58%). The rate of relative irreversibility in the other units is approximately smaller than 2%.

The performances of the desiccant air conditioning system components based on exergy efficiency are depicted in Fig. 4. As can be seen from this figure, HE3 has the maximum exergy efficiency with 93.40% and the exergy efficiency of this heat exchanger is higher than that of the other heat exchangers due to rotary type heat exchangers. It is also observed that fans have little exergy efficiency in this system and these values are calculated as 11.48%, 18.93% and 12.45% for fresh air fan, waste air fan and regeneration air fan, respectively. The exergy efficiency of the whole system is calculated as 40.7%. This value of the exergy efficiency is very important when it is compared with previous researches. Several research papers have been dedicated to explore issues about exergy of desiccant cooling systems such as [16,18]. Exergy efficiency of these researches varies between 8.6% and 11.1%. Taking into consideration these results, the exergy efficiency of this system is more meaningful when the system is compared with former researches.

Fig. 5 shows the variation of exergy efficiency with various reference temperatures. As can be seen from this figure, the exergy efficiency varies inversely with the reference temperature. The exergy efficiency of the system decreases from 56% to 25% when the reference temperature is increased from 0 °C to 30 °C. Fig. 5 illustrates also the effect of the reference temperature on the sustainability index (SI). The SI value decreases from 2.29 to 1.34 by increasing the reference temperature.

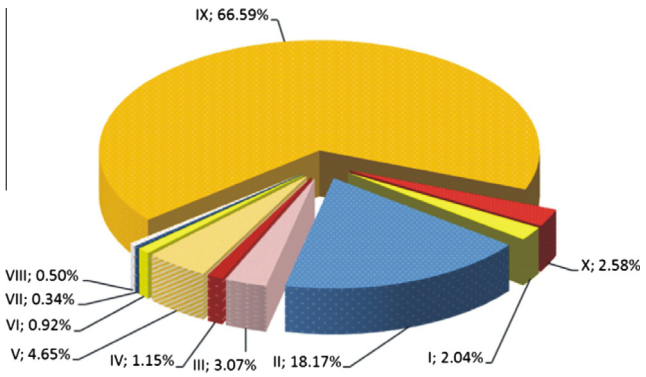


Fig. 3. Relative irreversibility of components in the whole system.

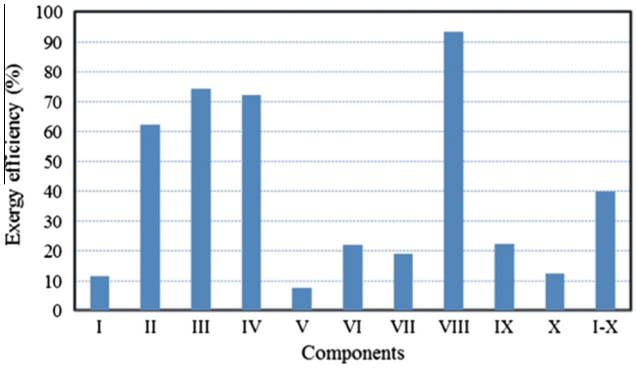


Fig. 4. Performances of the desiccant air conditioning system components based on exergy efficiency.

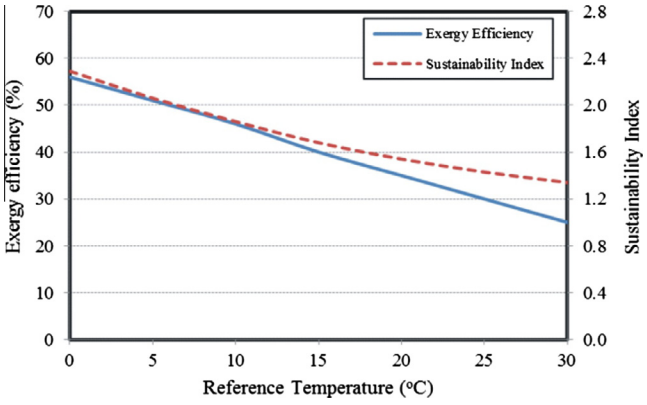


Fig. 5. Variation of exergy efficiency and sustainability index values with various reference temperatures.

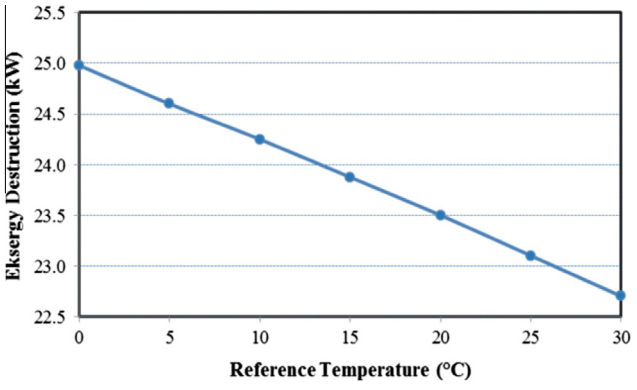
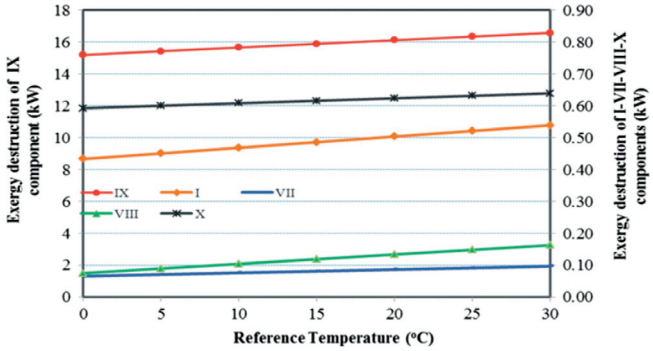


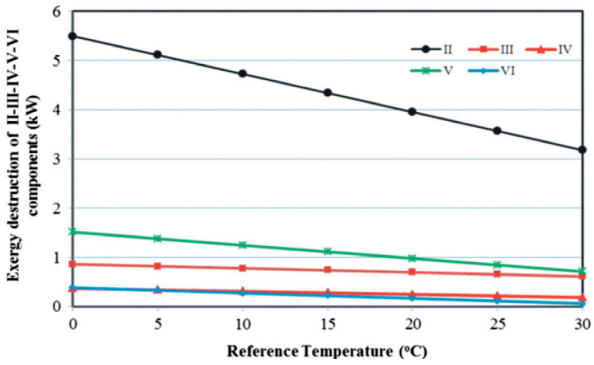
Fig. 6. Variation of exergy destruction of the whole system with various reference temperatures.

Fig. 6 illustrates the exergy destruction of the whole system with increasing the reference temperature. It is observed that the exergy destruction is inversely proportional to the reference temperatures for this system. The maximum and minimum exergy destruction rates of this system are found to be 24.978 kW and 22.764 kW for the reference temperatures of 0 °C and 30 °C, respectively.

The effect of the reference temperature on the components is depicted in Fig. 7. As seen in this figure, the effect of the reference temperature on the components is separated into two parts one of which shows positive effect (Fig. 7a) and the other illustrates negative effect (Fig. 7b). It is observed in Fig. 7a that the exergy



(a) Effect of reference temperature on exergy destruction of I-VII-VIII-IX and X components



(b) Effect of reference temperature on exergy destruction of II-III-IV-V and VI components

Fig. 7. Effect of the reference temperature on the components.

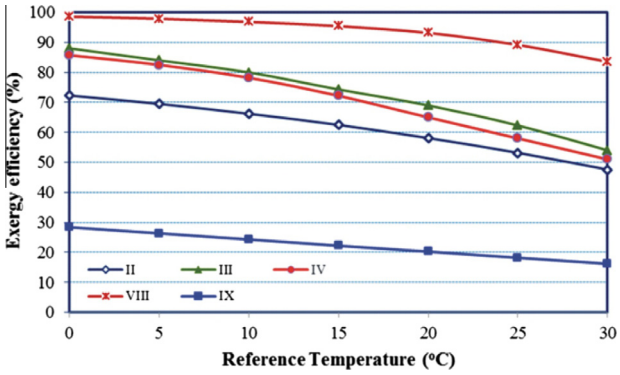


Fig. 8. Effect of reference temperature on the exergy efficiency of the major components (II–III–IV–VIII and IX).

destruction increases with the increase of reference temperature in some components that are I–VII–VIII–IX–X. It is also seen from this figure that the maximum increase of the exergy destruction occurs in the electric heater unit. When the reference temperature is increased from 0 °C to 30 °C, the exergy destruction of this component increases from 15.2 kW to 16.6 kW. In the other components, the increasing of exergy destruction ranged from very small values between 0.1 and 0.9 kW. It is seen from Fig. 7b. that the exergy destruction decreases with the increase of the reference temperature in the components II–III–IV–V–VI. As the reference temperature increases, the desiccant wheel is affected more than the other components. When the reference temperature is increased from 0 °C to 30 °C, the exergy destruction of this component decreases from 5.5 kW to 3.2 kW.

The effect of reference temperature on the exergy efficiency of the major components (II–III–IV–VIII and IX) is illustrated in Fig. 8. As can be seen in this figure, when the reference temperature is increased, the exergy efficiency of the major components decreases. The maximum decrease of the exergy efficiency occurs in the heat exchanger 3. When the reference temperature is increased from 0 °C to 30 °C, the exergy efficiency of this component decreases from 85% to 51%.

5. Conclusions

The performance of a novel configuration of a desiccant based evaporative air conditioning system is evaluated by using exergy analysis, which aims at better determining process efficiencies. The effects of the individual irreversible processes in each component on the thermodynamic performance of a desiccant based evaporative air conditioning system have been studied in this paper and indexes for thermodynamic evaluation have been derived and analyzed. The main conclusions that may be drawn from the present study are listed below.

- The exergy efficiency of the system is obtained to be 40.7% at a reference temperature of 15 °C. Exergy efficiency is also inversely proportional to the reference temperature. The maximum and minimum exergy efficiencies of the whole system are calculated to be 56% and 25% for the reference temperature of 0 °C and 30 °C, respectively.
- The exergy destruction of the whole system decreases by increasing the reference temperature. Minimization of exergy destruction is essential for increasing exergy and exergy efficiencies. The maximum exergy destruction of this system occurs in electric heater unit with 15.899 kW at a reference temperature of 15 °C.
- The highest irreversibility in the system occurs in the electric heater unit with 66.59%, followed by the desiccant wheel, EC1, HE1, regeneration air fan, fresh air fan and the others.
- The SI value decreases from 2.29 to 1.34 by increasing the reference temperature.
- Finally, it is observed that the electric heater unit used as a thermal energy source has a major impact on the system exergy performance. Instead of regenerating thermal energy, solar energy, waste heat and natural gas can be used to increase the exergy performance of the system.

The analysis shows that an exergy analysis can provide some useful information with respect to the theoretical upper limit of the system performance, which cannot be obtained from an energy analysis alone. The analysis allows also the determination of the places of the exergy destruction and shows the direction for the minimization of exergy destruction. Consequently, it is important to consider the exergy destruction of the components to improve the system performance.

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