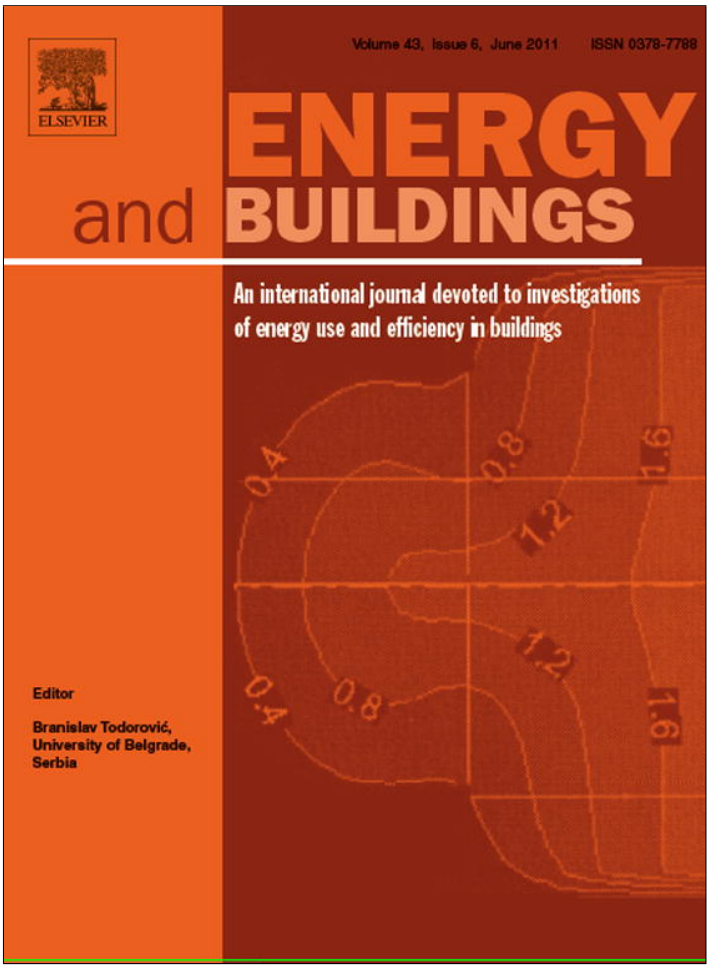
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Exergetic modeling and experimental performance assessment of a novel desiccant cooling system

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ABSTRACT

New approaches to space conditioning of buildings are required to resolve economic, environmental, and regulatory issues. One of the alternative systems that is brought to agenda is the desiccant cooling systems, which may provide important advantages in solving air conditioning problems. This study deals with the performance analysis and evaluation of a novel desiccant cooling system using exergy analysis method. The system was designed, constructed and tested in Cukurova University, Adana, Turkey and has been successfully operated since 2008. This system consists of a desiccant wheel, heat exchangers, fans, evaporative cooler, electric heater unit and refrigeration unit. The exergy transports between the components and the destructions in each of the components of the desiccant cooling system are determined for the average measured parameters obtained from the experimental results. Exergy efficiencies of the system components are determined in an attempt to assess their individual performances and the potential for improvements is also presented. The exergetic efficiency values for the whole system on the exergetic product/fuel basis are calculated to range from round 32% to 10% at the varying dead (reference) state temperatures of 0–30 °C.

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1. Introduction

The best thermal comfort conditions in buildings are required for human being. These comfort conditions are provided by air-conditioning and heating-cooling systems. However, in recent years alternatives to these systems have been needed because of the increasing energy requirements, some economic problems and harmful effects of refrigerants used in conventional air conditioning systems. One of the alternative systems that is brought to agenda is the desiccant cooling systems. In these systems, a desiccant removes moisture from the fresh air, which releases heat and increases the air temperature. The dry air is then cooled using either evaporative cooling or the cooling coils of a conventional vapor-compression refrigeration cycle. The adsorbed moisture in the desiccant is then removed (the desiccant is regenerated to its original dry state) using a secondary warm air flow.

Desiccant based air-conditioning systems are a suitable way to improve indoor air quality due to its superior humidity control. These systems are also alternatives for air conditioning in health care facilities to reduce the airborne disease transmission. The substances used in these processes for removing moisture, removes the bacteria that can cause illnesses and various infections in the air sent to the place being air-conditioned [1,2].

Several review studies performed on the description and operation of desiccant cooling systems by different researchers [3–6]. Systems that use rotary desiccant wheel to dehumidify the air are the most popular desiccant cooling systems and studied by different researchers [7–10]. They showed that desiccant cooling systems are viable alternative to vapor compression systems. Jain et al. [11] investigated a various solid desiccant cycles (the ventilation cycle, the recirculation cycle, the Dunkle cycle and the wet surface heat exchangers cycle) for various outdoor conditions of many cities in India. It was found that amongst ventilation, recirculation and Dunkle cycles, the Dunkle cycle is better for a wide range of outdoor conditions. Subramanyam et al. [12] studied a desiccant wheel integrated air-conditioner for low humidity air-conditioning. They tested a system at various supply airflow rates to evaluate its performance and compare with those of conventional and reheat systems. Their study showed that the proposed system can deliver supply air at much lower dew point temperature compared to the conventional system with a marginal penalty on COP. Jia et al. [13] investigated a high performance rotary solid desiccant cooling system using a novel compound desiccant wheel. Cejudo et al. [14] presented two methodologies for modeling a solid desiccant wheel: physical and neural network models. Experimental values are used to validate the physical model and to calculate the parameters of

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Nomenclature

C	specific heat (kJ/kg K)
$\dot{E}$	energy rate (kW)
$\dot{E}x$	exergy rate (kW)
$I\dot{P}$	improvement potential rate (kW)
P	pressure (kPa)
$\dot{Q}$	heat transfer rate (kW)
T	temperature (K or °C)

Greek letters

$\cos\varphi$	power factor (dimensionless)
ε	exergy (second law) efficiency (dimensionless)
ω	specific humidity ratio (kg_{water}/kg_{air})

Indices

0	reference (dead) state
a	air
act	actual
comp	compressor
con	condenser
dest	destroyed (destruction)
dw	desiccant wheel
ec	evaporative cooler
eh	electric heater
elec	electrical
evap	evaporator
exp	expansion valve
f	fresh air
he	heat exchanger
mech	mechanical
r	regeneration air
ref	refrigerant
s	isentropic process
sys	system
Tot	total
v	water vapor
w	waste air

the neural network model. An important conclusion of this work is that special attention must be paid when calculating moisture content from measured values of dry bulb temperature and relative humidity.

Kabeel [15] investigated the performance of a solar powered desiccant air conditioning system using a rotary honeycomb wheel, which is utilized for the regeneration and absorption processes, at different conditions of inlet air and radiation intensity. Zhang and Niu [16] and Ge et al. [17] studied a two stage rotary desiccant cooling system to investigate its performance. They reported that the required regeneration temperature of system is lower and thermal coefficient of performance is higher than single stage desiccant cooling system. Ge et al. [18] investigated experimentally a one rotor two stage rotary desiccant cooling system (OTSDC), in which two stage dehumidification process is realized by one desiccant wheel. They showed that OTSDC achieved the objective of reducing the size of two stage system with no reduction in system performance. Enteria et al. [19] presented the developed new desiccant cooling system employing double cross-flow heat exchangers to increase the dehumidification performance at lower thermal energy requirement and to maintain the supply air humidity content at the same level as that of the processed air.

Hürdoğan et al. [20] investigated experimentally a novel desiccant based air conditioning system to improve the indoor air quality and reduce energy consumption. In the system studied, the moisture of the fresh air was reduced passing it through a solid desiccant wheel and then its temperature decreased by the "dry coil" of a vapor-compression cycle. They showed that using a heat exchanger for pre-heating the regeneration air with exhaust air was feasible to install. It was also reported that although COP changes between 0.4 and 4 according to the electric heaters switching on or off, the daily mean value was 1.35.

During the past decade, there has been an increasing interest in using exergy as a potential analysis tool for design, analysis and performance evaluation of energy systems [21–26]. For evaluating exergy, the reference environment must be specified. Recently, much attention has been paid to the energy and exergy modeling techniques for energy utilization assessments to minimize energy savings and hence financial savings. Furthermore, exergy analysis is employed to detect and to evaluate quantitatively the causes of the thermodynamic imperfection of the process under consideration.

Exergy analysis has been applied to different types of air-conditioning systems by various researchers [i.e., 27–30]. The studies on exergy analysis (second law analysis) of the desiccant cooling systems, however, are relatively few in numbers. Lavan et al. [31] presented a second law overall system analysis of desiccant air conditioners. A second law analysis of the individual components in these systems gave additional understanding of the feasible performance of desiccant cooling systems. Van den Bulck et al. [32] focused only on the desiccant dehumidifier in their study. The second law efficiency of the dehumidifier was given as a function of operating conditions and the effect of finite transfer coefficients for an actual dehumidifier is discussed. It was shown that operating the rotary dehumidifier at conditions that minimize regeneration energy also yields a local maximum for the second law efficiency. Pons and Kodama [33] evaluated the various terms of internal and external entropy generation in ventilation mode, and expressed relevant formulation with respect to Carnot COP of the system. In the second part of the study, Kodama et al. [34] applied the formulation to an experimental unit and investigated the effects of certain operating parameters on the various terms of entropy generation. Kanoglu et al. [35] developed a procedure for the energy and exergy analyses of open-cycle desiccant cooling systems and it is applied to an experimental unit operating in ventilation mode with natural zeolite as the desiccant. They reported that the unit had a COP of 0.35, a reversible COP of 3.11, and an exergy efficiency of 11.1%. Desiccant wheel had the greatest percentage of total exergy destruction with 33.8% followed by the heating system with 31.2%. Rotary regenerator and evaporative coolers accounted for the remaining exergy destructions. Their analysis showed that an exergy analysis could provide useful information with respect to the theoretical upper limit of the system performance, which cannot be obtained from an energy analysis alone.

Based on the literature survey, there are not any studies on both exergetic and sustainability assessments of desiccant cooling systems to the best of the authors' knowledge. This provided the prima motivation behind doing the present study. In this regard, the authors have extended their previous experimental study on a novel desiccant based air conditioning system [20] to include its exergetic and sustainability aspects. In this study, exergy analysis of the novel desiccant cooling system is performed first. Exergy destructions for each component of the system and whole system are then identified, while the potential for efficiency improvements is presented. Finally, a sustainability assessment is conducted.

2. System description

2.1. Experimental set-up and operation principle

Fig. 1 illustrates a schematic of a desiccant cooling system studied. This system mainly consists of three separate circuits: (i)

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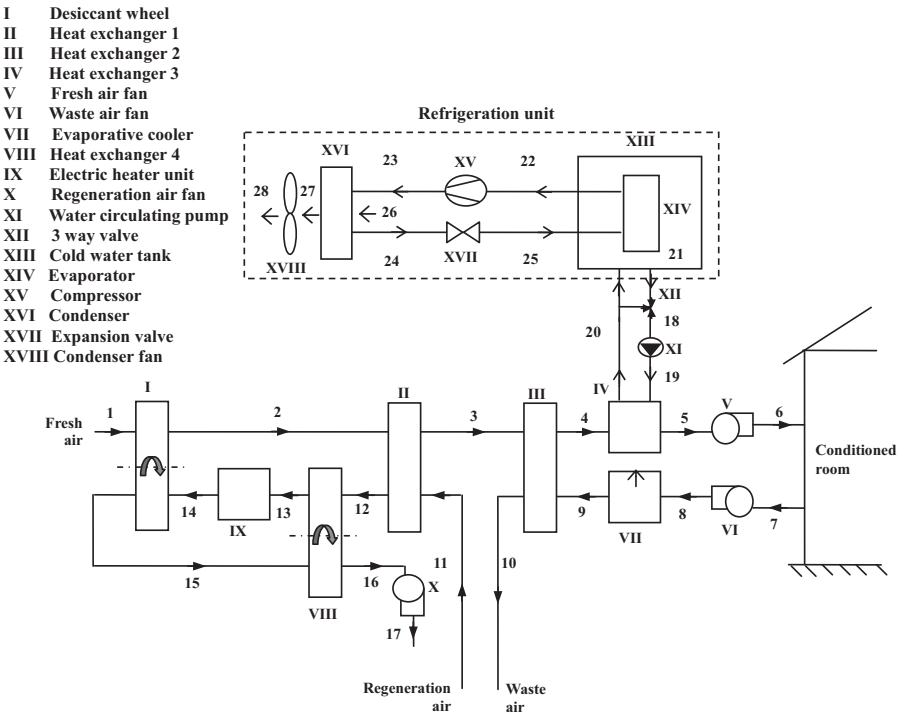


Fig. 1. A schematic view of the desiccant cooling system studied.

Table 1
The main characteristics of the elements of the desiccant cooling system.

Element	Technical specification
Desiccant wheel (I)	Manufacturer: Dehutech AB; model: DT10; desiccant material: silica gel; rotor diameter: 965 mm, rotor depth: 200 mm; 50% process air–50% regeneration air
Heat exchanger 1, 2 (II, III)	Manufacturer: Recuperator S.R.L.; model: BTAC06N060CTTVSC; type: recuperative (air to air, cross flow); material: epoxy coated aluminum; plate spacing: 1.5 mm
Heat exchanger 3 (IV)	Manufacturer: Friterm; model: CW 3228-5R-18T-0600x10-09-RR; type: plate fin tube (air to water); material: copper (tube), aluminum (fin); capacity: 18.6 kW
Heat exchanger 4 (VIII)	Manufacturer: Enventus; model: AK1-XL-W-965-C-200-V-R-C-1; type: rotary (air to air); diameter: 965 mm, depth: 200 mm
Fresh air fan (V)	Manufacturer: Nicotra; model: RDH 280 R; volumetric flow rate: 4000 m^3/h at 1400 Pa pressure loss; power: 3 kW; speed: 2880 rpm
Waste air fan (VI)	Manufacturer: Nicotra; model: RDH 280 R; volumetric flow rate: 4000 m^3/h at 400 Pa pressure loss; power: 1.1 kW; speed: 2880 rpm
Regeneration air fan (X)	Manufacturer: Nicotra; model: RDH 280 R; volumetric flow rate: 4000 m^3/h at 1600 Pa pressure loss; power: 4 kW; speed: 2880 rpm
Evaporative cooler (VII)	Technology: Fogging, nozzle diameter: 0.2 mm, nozzle material: rice
Electric heater unit (IX)	Manufacturer: Baykal; type: finned tubular; material: chrome-nickel
Water circulating pump (XI)	Manufacturer: Wilo, model: WJ203; maximum flow rate: 5 m^3 h^{-1}; maximum pressure head: 43 m of water column; power: 0.75 kW; speed: 2850 rpm
3 way valve (XII)	Manufacturer: Siemens; model: VXP45
Cold water tank (XIII)	Material: chrome; isolation: 5 cm glass wool; capacity: 325 L
Evaporator (XIV)	Type: spiral tube; material: copper; diameter: 22 mm; length: 60 m
Compressor (XV)	Manufacturer: Copeland; model: D3DA 75X; type: semi-hermetic; volumetric flow rate: 32.2 m^3/h; refrigerant: R134a; the rated power of electric motor driving: 7.5 HP (5.6 kW); capacity: 17.1 kW (at evaporating/condensing temperatures of 5/50 °C)
Condenser (XVI)	Manufacturer: Friterm, type: plate fin tube; material: copper (tube), aluminum (fin); heat transfer surface: 120 m^2
Expansion valve (XVII)	Copper capillary tube: 1.5 m long; inside diameter: 1.5 mm

desiccant air-conditioning circuit (air circuit), (ii) the refrigerant circuit and (iii) air-conditioning room. The main characteristics of the elements of the desiccant cooling system are given in Table 1, where the numbers in parentheses correspond to these elements as depicted in Fig. 1 [20].

Humidity of the fresh outside air (state 1) is absorbed by the desiccant material of the wheel (I). The fresh air leaves the process side of the desiccant wheel (state 2) hotter, and much drier than when entering the system. Sensible cooling of the fresh air is carried out (process 4–5) in a cooling coil (IV), which is fed by chilled water from a refrigeration unit (XIV–XVII). However, the fresh air is passed through two recuperative type heat exchangers (II, III) before coming to the cooling coil (heat exchanger 3) for cool recovery. In the recuperators (heat exchangers 1 and 2), only sensible heat is removed from the fresh air, and therefore, humidity ratio of the air remains constant.

The desired blowing temperature of the fresh air (state 6) is obtained in the cooling coil. The surface temperature of the cooling coil is always kept higher than the dew point temperature of the fresh air entering into the cooling coil. Therefore, only sensible cooling is performed in the coil (dry-coil application). Moisture from the fresh air (latent cooling) is removed only in the desiccant wheel and there is no change in the humidity ratio of the fresh air after the desiccant wheel.

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The air sucked from the indoor (state 7) into the waste air duct is evaporatively cooled in an evaporative cooler (VII) before entering into heat exchanger 2 in order to increase the cool recovery. In heat exchanger 2, due to the heat transfer from the fresh air to the waste air, temperature of the fresh air decreases and that of the waste air increases (process 9–10). The waste air leaving the heat exchanger is rejected to the outdoors.

Hot regeneration air is used for removing moisture at the desiccant wheel. In this study, fresh outdoor air is used for regeneration process. The reason of using ambient air is to eliminate the possibility of mixing of the fresh air with the waste air, which may contains harmful bacteria originating from the air-conditioned room. The regeneration air first comes to heat exchanger 1 in which heat transfer from the fresh air to the regeneration air takes place. Temperature of the regeneration air at the exit of the desiccant wheel (state 15) is generally higher than that of the regeneration air leaving heat exchanger 1 (state 12). Therefore, a rotary regenerator (heat exchanger 4, VIII) is used for heat recovery. The regeneration air leaving the desiccant wheel passes through the heat exchanger 4 (process 15–16), in which heat is transferred from the regeneration air that left the desiccant wheel (state 15) to the regeneration air left heat exchanger 1 (state 12). Although the temperature of the regeneration air is increased in heat exchanger 1 and 4 (process 11–13), it is not high enough for dehumidification of the desiccant wheel. Therefore, temperature of the regeneration air should be increased to the desired temperature before entering into the desiccant wheel. Temperature can be increased economically, if a cheap thermal energy source such as solar energy, waste heat or geothermal energy is available. Utilization of such cheap heating energy is very important for the economic operation of desiccant systems. If these systems can be backed up by a cheap thermal energy source, operating cost can be decreased dramatically. In this study, the final temperature of the regeneration air is achieved with the help of electric heaters (IX) to simulate the cheap heat source (process 13–14). Electric heaters are used for adjusting the limits of regeneration temperature accurately, for simplifying the system and also reducing the size of the system. The air removes the humidity of the desiccant wheel (process 14–15) and flows through heat exchanger 4 (process 15–16) before discharged to the outdoors.

2.2. Measurement system and control panel

Temperature, relative humidity, flow rate, electric current and electrical potential difference are measured during the experimental tests. Dry bulb temperature of the air at states 1–17 and 26–28 in Fig. 1 is measured by using Cole-Parmer K-type thermocouples with an accuracy 0.1 °C. Temperature of the water and refrigerant at states 18–25 is also measured by using Cole-Parmer K-type thermocouples. An ice-bath is employed as the reference junction for all the thermocouples used. Relative humidity of the air at states 1, 3, 6, 7, 9 and 16 is measured by using $E+E$ relative humidity transmitters with an accuracy of ±2% (0–90 RH), ±3% (90–100 RH). Dew point temperature of the air at state 4 is measured by using $E+E$ dew point transmitter with an accuracy of ±0.2 °C. Flow rate of the air at states 2, 7 and 15 (fresh, waste and regeneration air channels) is measured by using averaging (blade type) Pitot tubes and Cole-Parmer differential pressure transmitter with an accuracy of ±1%. Flow rate of the water at state 19 is measured by using a rotameter with an accuracy of ±2%. Flow rate of the refrigerant at state 24 is measured by Bass Instruments metal tube flow meter with an accuracy of ±2%. Electric current and electrical potential difference are measured to determine the power consumption of each electrical component in the desiccant cooling system. Monitoring modules are used for this purpose to measure the electric current with an accuracy of ±0.5% and electrical potential difference with an accuracy of ±1.5%. Electrical signal outputs of the thermocouples, transmitters and modules are monitored and recorded with an interval of 30 s by using computer-controlled data acquisition system that have 16 bit resolution and 1 MHz maximum sample rate.

A Programmable Logic Controller (PLC) is used to control the operation of desiccant cooling system. Its functions included

- Controlling the system and operation of its components (start-up and shut-down),
- Adjusting the dry bulb temperature of the air-conditioned room according to the set values by controlling the cooled water temperature with three way valve,
- Adjusting the relative humidity of the air-conditioned room according to the set values by controlling the regeneration temperature with electric heaters,
- Preventing the condensation at cooling coil by controlling the cooled water temperature with three way valve, and
- Protecting the desiccant wheel from the high temperature with the control of electric heaters.

2.3. Uncertainty analysis

Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis is performed using the method described by Holman [36]. In the present study, the temperatures, relative humidities, flow rates, electric currents and electrical potential differences are measured with appropriate instruments explained previously. The total uncertainties of the measurements are estimated to be 0.3 °C for the air, water and refrigerant temperatures, 2.51% for the relative humidities, 1.59% for power inputs to the electrical heaters, compressor and motor of the fans, heat exchangers and circulating pump. The total uncertainty associated with mass flow rates of the air, water and refrigerant was found to be 2.89%, 3.02%, 3.26%, while those with the exergy input/output rates and exergy efficiencies were calculated to be on average 3.20% and 4.52%, respectively.

3. Modeling and analysis

Mass, energy and exergy balances are employed to find the heat input, the rate of exergy destruction, and energy and energy efficiencies. Steady-state, steady-flow processes are assumed. A general mass balance can be expressed in the rate form as:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

where $\dot{m}$ is the mass flow rate, and the subscript "*in*" stands for inlet and "*out*" for outlet.

The general energy balance can be expressed as:

$$\dot{E}_{in} = \dot{E}_{out} \tag{2}$$

where the subscript "*in*" stands for the rate of net energy transfer in by heat, work and mass and "*out*" stands for the rate of net energy transfer out by heat, work and mass. The general energy balance can also written more explicitly as:

$$\dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out} \tag{3}$$

where $\dot{Q} = \dot{Q}_{net,in} = \dot{Q}_{in} - \dot{Q}_{out}$ is the rate of net heat input, $\dot{W} = \dot{W}_{net,out} = \dot{W}_{out} - \dot{W}_{in}$ is the rate of net work output, and h is the enthalpy per unit mass.

The general exergy balance can be expressed in the rate form as:

$$\dot{E}x_{in} - \dot{E}x_{out} = \dot{E}x_{dest} \tag{4a}$$

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where $\dot{E}x_{in} - \dot{E}x_{out}$ stands for the rate of net exergy transfer by heat, work and mass and $\dot{E}x_{dest}$ stands for the rate of net exergy destruction. The general exergy balance can also written as:

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

Using Eq. (4b), the rate of formation of the general exergy balance can also be written as:

$$\sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k - \dot{W} + \sum \dot{m}_{in}\psi_{in} - \sum \dot{m}_{out}\psi_{out} = \dot{E}x_{dest} \tag{5}$$

where $\dot{Q}_k$ is the heat transfer rate through the boundary at temperature T_k at location k, $\dot{W}$ is the work rate, ψ is the flow (specific) exergy, h is the enthalpy and the subscript zero indicates properties at the reference (dead) state of P_0 and T_0.

The specific flow exergy of refrigerant or water is evaluated as:

$$\psi_{ref,water} = (h - h_0) - T_0(s - s_0) \tag{6}$$

The total flow exergy of air is determined as [21]:

$$\psi_a = (C_{p,a} + \omega C_{p,v})T_0 \left[\left(\frac{T}{T_0}\right) - 1 - \ln\left(\frac{T}{T_0}\right)\right] + (1 + 1.6078\omega)R_a T_0 \ln\left(\frac{P}{P_0}\right) + R_a T_0 \left\{(1 + 1.6078\omega) \ln\left[\frac{1 + 1.6078\omega_0}{1 + 1.6078\omega}\right] + 1.6078\omega \ln\left(\frac{\omega}{\omega_0}\right)\right\} \tag{7}$$

where s is the entropy and the specific humidity ratio is:

$$\omega = \frac{\dot{m}_v}{\dot{m}_a} \tag{8}$$

The exergy rate is determined as:

$$\dot{E}x = \dot{m}\psi \tag{9}$$

Exergy efficiency may be defined in various ways, namely universal exergy efficiency (ε_U) and functional exergy efficiency (ε_F), as given below, respectively:

$$\varepsilon_U = \frac{\dot{E}x_{out}}{\dot{E}x_{in}} \tag{10}$$

where "*out*" refers to total exergy output and "*in*" refers to total exergy input.

$$\varepsilon_F = \frac{\text{Exergetic Product}}{\text{Exergetic Fuel}} = \frac{\dot{P}}{\dot{F}} \tag{11}$$

Van Gool's improvement potential on a rate basis, denoted $I\dot{P}$, is expressible as:

$$I\dot{P} = (1 - \varepsilon)(\dot{E}x_{in} - \dot{E}x_{out}) \tag{12}$$

The relative irreversibility (RI) is evaluated as:

$$\text{RI} = \frac{\dot{E}x_{dest,i}}{\dot{E}x_{dest,tot}} = \frac{\dot{I}_i}{\dot{I}_{Tot}} \tag{13}$$

where the subscript "*i*" denotes the *i*th device.

Sustainability index (SI) is a factor, which is directly related to the exergetic efficiency [37]. It aims at supplying clean and affordable energy resources to be used. The efficient use of these resources is also of great importance. SI, which has been recently used by some investigators [i.e., 38,39], can be formulated to be [37]:

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

4. Application of energy and exergy analysis methods to the system investigated

The following assumptions are made during the energy and exergy analyses:

(a) All processes are steady-state and steady-flow with negligible potential and kinetic energy effects and no chemical or nuclear reactions.
(b) Heat transfer to the system and work transfer from the system are positive.
(c) Heat transfer and refrigerant pressure drops in the tubing connecting the components are neglected since their lengths are short.
(d) The circulating pump mechanical ($\eta_{\text{pump,mech}}$) and the circulating pump electrical ($\eta_{\text{pump,elec}}$) efficiencies are 82% and 88%, respectively.
(e) Motor electrical efficiency of desiccant wheel ($\eta_{\text{dw,elec}}$) and heat exchanger 4 ($\eta_{\text{he4,elec}}$) is assumed to be 88%.
(f) The condenser fan mechanical ($\eta_{\text{fan,con,mech}}$) and the condenser fan electrical ($\eta_{\text{fan,con,elec}}$) efficiencies are 80% and 88%, respectively.

Mass and energy balances as well as exergy destructions obtained from exergy balances for each of the components illustrated in Fig. 1 can be expressed as follows:

Desiccant wheel (I):

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

$$\dot{Q}_{dw} = \dot{m}_f(h_2 - h_1); \quad \dot{Q}_{dw} = \dot{m}_r(h_{14} - h_{15}) \tag{14b}$$

$$\dot{E}x_{dest,dw} = \dot{m}_f(\psi_1 - \psi_2) + \dot{m}_r(\psi_{14} - \psi_{15}) + \dot{W}_{dw,elec} \tag{14c}$$

$$\dot{W}_{dw,elec} = \frac{\dot{W}_{dw}}{\eta_{dw,elec}} \tag{14d}$$

Heat exchanger 1 (II):

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_f; \quad \dot{m}_{11} = \dot{m}_{12} = \dot{m}_r \tag{15a}$$

$$\dot{Q}_{he1} = \dot{m}_f(h_2 - h_3); \quad \dot{Q}_{he1} = \dot{m}_r(h_{12} - h_{11}) \tag{15b}$$

$$\dot{E}x_{dest,he1} = \dot{m}_f(\psi_2 - \psi_3) + \dot{m}_r(\psi_{11} - \psi_{12}) \tag{15c}$$

Heat exchanger 2 (III):

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_f; \quad \dot{m}_9 = \dot{m}_{10} = \dot{m}_w \tag{16a}$$

$$\dot{Q}_{he2} = \dot{m}_f(h_3 - h_4); \quad \dot{Q}_{he2} = \dot{m}_w(h_{10} - h_9) \tag{16b}$$

$$\dot{E}x_{dest,he2} = \dot{m}_f(\psi_3 - \psi_4) + \dot{m}_w(\psi_9 - \psi_{10}) \tag{16c}$$

Heat exchanger 3 (IV):

$$\dot{m}_4 = \dot{m}_5 = \dot{m}_f; \quad \dot{m}_{18} = \dot{m}_{19} = \dot{m}_{water} \tag{17a}$$

$$\dot{Q}_{he3} = \dot{m}_f(h_4 - h_5); \quad \dot{Q}_{he3} = \dot{m}_{water}c_{p,water}(T_{20} - T_{19}) \tag{17b}$$

$$\dot{E}x_{dest,he3} = \dot{m}_f(\psi_4 - \psi_5) + \dot{m}_{water}(\psi_{19} - \psi_{20}) \tag{17c}$$

Fresh air fan (V):

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_f \tag{18a}$$

$$\dot{W}_{fan,f} = \dot{m}_f(h_6 - h_5) \tag{18b}$$

$$\dot{E}x_{dest,fan,f} = \dot{m}_f(\psi_5 - \psi_6) + \dot{W}_{fan,f,elec} \tag{18c}$$

Waste air fan (VI):

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_w \tag{19a}$$

$$\dot{W}_{fan,w} = \dot{m}_w(h_8 - h_7) \tag{19b}$$

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$$\dot{E}x_{dest,fan,w} = \dot{m}_w(\psi_7 - \psi_8) + \dot{W}_{fan,w,elec} \quad (19c)$$

Evaporative cooler (VII):

$$\dot{m}_8 = \dot{m}_9 = \dot{m}_w \quad (20a)$$

$$\dot{m}_w h_8 + \dot{m}_{water,ec} h_{water,ec} = \dot{m}_w h_9 \quad (20b)$$

$$\dot{E}x_{dest,ec} = \dot{m}_w(\psi_8 - \psi_9) + \dot{m}_{water,ec}(\psi_{water,ec}) \quad (20c)$$

Heat exchanger 4 (VIII):

$$\dot{m}_{12} = \dot{m}_{13} = \dot{m}_{15} = \dot{m}_{16} = \dot{m}_r \quad (21a)$$

$$\dot{Q}_{he4} = \dot{m}_r(h_{13} - h_{12}) = \dot{m}_r(h_{15} - h_{16}) \quad (21b)$$

$$\dot{E}x_{dest,he4} = \dot{m}_r(\psi_{12} - \psi_{13}) + \dot{m}_r(\psi_{15} - \psi_{16}) + \dot{W}_{he4,elec} \quad (21c)$$

$$\dot{W}_{he4,elec} = \frac{\dot{W}_{he4}}{\eta_{he4,elec}} \quad (21d)$$

Electric heater unit (IX):

$$\dot{m}_{13} = \dot{m}_{14} = \dot{m}_r \quad (22a)$$

$$\dot{W}_{eh} = \dot{m}_r(h_{14} - h_{13}) \quad (22b)$$

$$\dot{E}x_{dest,eh} = \dot{m}_r(\psi_{13} - \psi_{14}) + \dot{W}_{eh} \quad (22c)$$

Regeneration air fan (X):

$$\dot{m}_{16} = \dot{m}_{17} = \dot{m}_r \quad (23a)$$

$$\dot{W}_{fan,r} = \dot{m}_r(h_{17} - h_{16}) \quad (23b)$$

$$\dot{E}x_{dest,fan,r} = \dot{m}_r(\psi_{16} - \psi_{17}) + \dot{W}_{fan,r,elec} \quad (23c)$$

Water circulating pump (XI):

$$\dot{m}_{18} = \dot{m}_{19s} = \dot{m}_{19,act} = \dot{m}_{water} \quad (24a)$$

$$\dot{W}_{pump} = \dot{m}_{water}(h_{19,act} - h_{18}) \quad (24b)$$

$$\dot{W}_{pump,elec} = \frac{\dot{W}_{pump}}{\eta_{pump,elec}\eta_{pump,mech}} \quad (24c)$$

$$\dot{E}x_{dest,pump} = \dot{m}_{water}(\psi_{18} - \psi_{19,act}) + \dot{W}_{pump,elec} \quad (24d)$$

where interactions with the environment are neglected.

Evaporator (XIV):

$$\dot{m}_{22} = \dot{m}_{25} = \dot{m}_{ref} \quad (25a)$$

$$\dot{Q}_{evap} = \dot{m}_{ref}(h_{22} - h_{25}) \quad (25b)$$

$$\dot{E}x_{dest,evap} = \dot{m}_{ref}(\psi_{25} - \psi_{22}) - \dot{Q}_{evap}\left(\frac{T_0}{T_{21}} - 1\right) \quad (25c)$$

Compressor (XV):

$$\dot{m}_{22} = \dot{m}_{23,s} = \dot{m}_{23,act} = \dot{m}_{ref} \quad (26a)$$

$$\dot{W}_{comp} = \dot{m}_{ref}(h_{23,act} - h_{22}) \quad (26b)$$

$$\dot{W}_{comp,elec} = \frac{\dot{W}_{comp}}{\eta_{comp,elec}\eta_{comp,mech}} \quad (26c)$$

$$\dot{W}_{com,elec} = \sqrt{3}\, V_{comp} I_{comp} \cos\varphi \quad (26d)$$

$$\dot{E}x_{dest,comp} = \dot{m}_{ref}(\psi_{22} - \psi_{23,act}) + \dot{W}_{comp,elec} \quad (26e)$$

where heat interactions with the environment are neglected.

Condenser (XVI):

$$\dot{m}_{23} = \dot{m}_{24} = \dot{m}_{ref}; \quad \dot{m}_{26} = \dot{m}_{27} = \dot{m}_{fan,cond} \quad (27a)$$

$$\dot{Q}_{cond} = \dot{m}_{ref}(h_{23,act} - h_{24}); \quad \dot{Q}_{cond} = \dot{m}_{fan,cond} c_{p,a}(T_{27} - T_{26}) \quad (27b)$$

$$\dot{E}x_{dest,cond} = \dot{m}_{ref}(\psi_{23,act} - \psi_{24}) + \dot{m}_{fan,con}(\psi_{26} - \psi_{27}) \quad (27c)$$

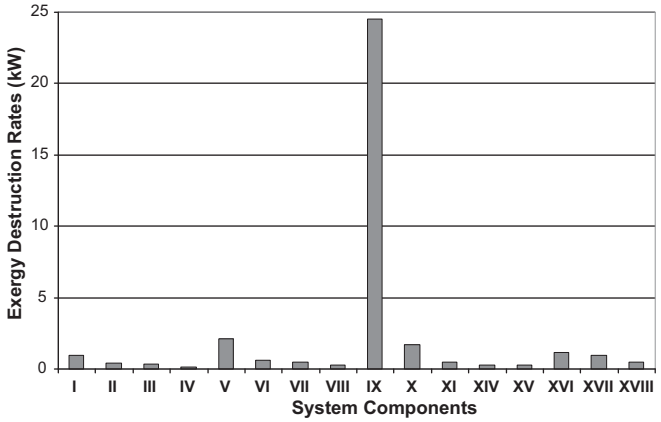


Fig. 2. Exergy destruction rate values of the components used in the system.

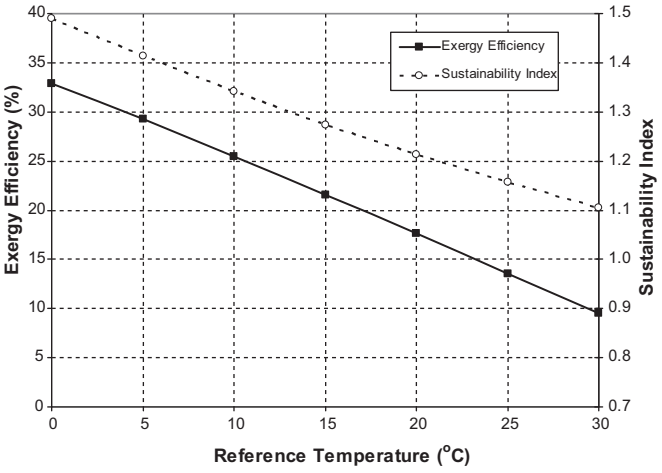


Fig. 3. Variation of exergy efficiency and sustainability index values with various reference temperatures.

Expansion valve (XVII):

$$\dot{m}_{24} = \dot{m}_{25} = \dot{m}_{ref} \quad (28a)$$

$$h_{24} = h_{25} \quad (28b)$$

$$\dot{E}x_{dest,\exp} = \dot{m}_{ref}(\psi_{24} - \psi_{25}) \quad (28c)$$

Condenser fan (XVIII):

$$\dot{m}_{27} = \dot{m}_{28} = \dot{m}_{con} \quad (29a)$$

$$\dot{W}_{fan,con} = \dot{m}_{con}\left[(h_{27} - h_{28}) + \frac{V^2}{2}\right] \quad (29b)$$

$$\dot{W}_{fan,con,elec} = \frac{\dot{W}_{fan}}{\eta_{fan,con,elec}\eta_{fan,con,mech}} \quad (29c)$$

$$\dot{E}x_{dest,fan,con} = \dot{m}_{con}(\psi_{28} - \psi_{27}) + \dot{W}_{fan,con,elec} \quad (29d)$$

Exergy efficiencies of the refrigeration unit, desiccant cooling system and its components are evaluated as follows:

The refrigeration unit (XIV–XVII):

$$\varepsilon_{U,ref} = \frac{\dot{E}x_{cool}}{\dot{W}_{comp,elec}} = \frac{\dot{E}x_{in,evap} - \dot{E}x_{out,evap}}{\dot{W}_{comp,elec}} = \frac{\dot{E}x_{25} - \dot{E}x_{22}}{\dot{W}_{comp,elec}} \quad (30a)$$

$$\varepsilon_{F,ref} = \frac{\sum \dot{P}_{ref}}{\sum \dot{F}_{ref}} \quad (30b)$$

Table 2
Exergy analysis results of the novel desiccant cooling system.

State no.	Description	Fluid	Phase	Temperature (°C)	Pressure (kPa)	Specific humidity ratio (kg water/kg dry air)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg K)	Mass flow rate (kg/s)	Specific exergy (kJ/kg)	Exergy rate (kW)
0	–	Moist air	Dead state	15	101.325	0.0158	–	–	–	0	0
0	–	Water	Dead state	15	101.325	–	63.00	0.224	–	0	0
0		Refrigerant (R-134a)	Dead state	15	101.325	–	267.90	1.076	–	0	0
1, 11	Outdoor	Air	Gas	31.94	101.325	0.0169	75.32	–	1.240	0.502	0.622
2	Dehumidifier outlet/Heat ex. 1 inlet	Air	Gas	52.10	101.325	0.0099	78.18	–	1.240	2.419	2.999
3	Heat ex. 1 outlet/Heat ex. 2 inlet	Air	Gas	41.30	101.325	0.0099	67.12	–	1.240	1.324	1.642
4	Heat ex. 2 outlet/Heat ex. 3 inlet	Air	Gas	26.87	101.325	0.0099	52.31	–	1.240	0.409	0.507
5	Heat ex. 3 outlet/F. fan inlet	Air	Gas	20.94	101.325	0.0099	46.37	–	1.240	0.227	0.282
6	F. fan outlet/Conditioned room inlet	Air	Gas	21.98	101.325	0.0099	47.33	–	1.240	0.251	0.311
7	Conditioned room outlet/W. fan inlet	Air	Gas	26.07	101.325	0.0110	54.27	–	1.240	0.319	0.395
8	W. fan outlet/Evaporative cooler inlet	Air	Gas	26.61	101.325	0.0110	54.82	–	1.240	0.340	0.421
9	Evaporative cooler outlet/Heat ex. 2 inlet	Air	Gas	20.47	101.325	0.0140	56.01	–	1.240	0.067	0.083
10	Heat ex. 2 outlet	Air	Gas	34.80	101.325	0.0140	70.78	–	1.240	0.685	0.849
12	Heat ex. 1 outlet/Heat ex. 4 inlet	Air	Gas	42.36	101.325	0.0169	86.11	–	1.240	1.272	1.577
13	Heat ex. 4 outlet/Elec. heater unit inlet	Air	Gas	48.28	101.325	0.0169	92.25	–	1.240	1.856	2.301
14	Elec. heater unit outlet/Dehumidifier inlet	Air	Gas	68.95	101.325	0.0169	113.68	–	1.240	4.667	5.787
15	Dehumidifier outlet/Heat ex. 4 inlet	Air	Gas	48.90	101.325	0.0238	110.86	–	1.240	2.168	2.688
16	Heat ex. 4 outlet/R. fan inlet	Air	Gas	43.09	101.325	0.0238	104.77	–	1.240	1.576	1.954
17	R. fan outlet	Air	Gas	44.02	101.325	0.0238	105.75	–	1.240	1.664	2.063
18	Cold water tank outlet/Water pump inlet	Water	Liquid	17.34	101.325	–	72.80	0.258	0.900	−0.026	−0.023
19	Water pump outlet/Heat ex. 3 inlet	Water	Liquid	17.44	101.325	–	73.20	0.259	0.900	−0.009	−0.008
20	Heat ex. 3 outlet/Cold water tank inlet	Water	Liquid	19.41	101.325	–	81.50	0.288	0.900	0.058	0.053
21	Cold water tank	Water	Liquid	8.18	101.325	–	34.5	0.124	–	–	–
22	Evaporator outlet/Compressor inlet	Refrigerant	Super heated vapor	10.90	339	–	406.97	1.746	0.147	−53.991	−7.937
23	Compressor outlet/Condenser inlet	Refrigerant	Super heated vapor	75.25	1522	–	448.90	1.774	0.147	−20.129	−2.959
24	Condenser outlet/Expansion valve inlet	Refrigerant	Liquid	51.90	1497	–	274.46	1.245	0.147	−42.137	−6.194
25	Expansion valve outlet/Evaporator inlet	Refrigerant	Mixture	4.96	349	–	274.46	1.268	0.147	−48.765	−7.168
26	Condenser inlet	Air	Gas	31.94	101.325	0.0169	–	–	1.670	0.502	0.622
27	Condenser outlet/Condenser fan inlet	Air	Gas	47.11	101.325	0.0169	–	–	1.670	1.732	2.893
28	Condenser fan outlet	Air	Gas	42.96	101.325	0.0169	–	–	1.670	1.326	2.215
	Evaporative cooler inlet	Water	Liquid	20.00	6000		89.5	0.295	0.018	6.04	0.1087

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Table 3
Exergy, improvement potential rate ($I\dot{P}$) and relative irreversibility (RI) data for representative components of the whole system.

Item number	Component	Exergetic product (exergy output) rate $\dot{P}$ (kW)	Exergetic fuel (exergy input) rate $\dot{F}$ (kW)	Exergy destruction rate $\dot{E}x_{dest}$ (kW)	Exergy efficiency ε (%)	Exergetic improvement potential rate $I\dot{P}$ (kW)	Relative irreversibility RI (%)		
							Refrigeration unit	Heat exchangers	Overall system
I	Dehumidifier	2.377	3.299	0.922	72.05	0.258	–	–	2.62
II	Heat exchanger 1	0.955	1.357	0.403	70.38	0.119	–	33.70	1.15
III	Heat exchanger 2	0.766	1.135	0.368	67.49	0.120	–	30.77	1.05
IV	Heat exchanger 3	0.061	0.225	0.165	27.11	0.120	–	13.80	0.47
V	Fresh air fan	0.029	2.152	2.121	1.35	2.092	–	–	6.03
VI	Waste air fan	0.026	0.640	0.614	4.06	0.589	–	–	1.75
VII	Evaporative cooler	0.083	0.529	0.447	15.67	0.377	–	–	1.27
VIII	Heat exchanger 4	0.724	0.984	0.260	73.58	0.069	–	21.73	0.74
IX	Electric heater unit	3.486	27.98	24.494	12.46	21.442	–	–	69.63
X	Regeneration air fan	0.109	1.798	1.689	6.06	1.587	–	–	4.80
XI	Pump	0.015	0.498	0.483	3.01	0.469	–	–	1.37
XIV	Evaporator	0.472	0.769	0.296	61.38	0.114	9.21	–	0.84
XV	Compressor	4.978	5.280	0.301	94.28	0.017	9.37	–	0.86
XVI	Condenser	2.055	3.235	1.181	63.52	0.352	36.75	–	3.36
XVII	Expansion valve	−7.168	−6.194	0.974	86.41	0.132	30.30	–	2.77
XVIII	Condenser fan	0.678	1.14	0.462	59.47	0.187	14.37	–	1.31
XIV–XVIII	Refrigeration unit	1.015	4.230	3.214	23.99	2.443	100	–	–
II–IV, VIII	Heat exchangers	2.506	3.701	1.196	67.71	0.386		100	–
I–XVIII	Overall system	9.646	44.827	35.18	21.23	28.19			100

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Overall desiccant cooling system (I–XVII):

$$\varepsilon_{F,sys} = \frac{\sum \dot{P}_{sys}}{\sum \dot{F}_{sys}} \quad (31)$$

Desiccant wheel (I):

$$\varepsilon_{F,dw} = \frac{\dot{E}x_2 - \dot{E}x_1}{\dot{E}x_{14} - \dot{E}x_{15} + \dot{W}_{dw,elec}} \quad (32)$$

Heat exchanger 1 (II):

$$\varepsilon_{F,he1} = \frac{\dot{E}x_{12} - \dot{E}x_{11}}{\dot{E}x_2 - \dot{E}x_3} \quad (33)$$

Heat exchanger 2 (III):

$$\varepsilon_{F,he2} = \frac{\dot{E}x_{10} - \dot{E}x_9}{\dot{E}x_3 - \dot{E}x_4} \quad (34)$$

Heat exchanger 3 (IV):

$$\varepsilon_{F,he3} = \frac{\dot{E}x_{20} - \dot{E}x_{19}}{\dot{E}x_4 - \dot{E}x_5} \quad (35)$$

Fresh air fan (V):

$$\varepsilon_{F,fan,f} = \frac{\dot{E}x_6 - \dot{E}x_5}{\dot{W}_{fan,f,elec}} \quad (36)$$

Waste air fan (VI):

$$\varepsilon_{F,fan,w} = \frac{\dot{E}x_8 - \dot{E}x_7}{\dot{W}_{fan,w,elec}} \quad (37)$$

Evaporative Cooler (VII):

$$\varepsilon_{U,ec} = \frac{\dot{E}x_9}{\dot{E}x_8 + \dot{E}x_{water,ec}} \quad (38)$$

Heat exchanger 4 (VIII):

$$\varepsilon_{F,he4} = \frac{\dot{E}x_{13} - \dot{E}x_{12}}{\dot{E}x_{15} - \dot{E}x_{16} + \dot{W}_{he4,elec}} \quad (39)$$

Electric heater unit (IX):

$$\varepsilon_{F,eh} = \frac{\dot{E}x_{14} - \dot{E}x_{13}}{\dot{W}_{eh}} \quad (40)$$

Regeneration air fan (X):

$$\varepsilon_{F,fan,r} = \frac{\dot{E}x_{17} - \dot{E}x_{16}}{\dot{W}_{fan,r,elec}} \quad (41)$$

Water circulating pump (XI):

$$\varepsilon_{F,pump} = \frac{\dot{E}x_{19,act} - \dot{E}x_{18}}{\dot{W}_{pump,elec}} \quad (42)$$

Evaporator (XIV):

$$\varepsilon_{F,evap} = \frac{\dot{Q}_{evap}((T_0/T_{21}) - 1)}{\dot{E}x_{25} - \dot{E}x_{22}} \quad (43)$$

Compressor (XV):

$$\varepsilon_{F,comp} = \frac{\dot{E}x_{23,a} - \dot{E}x_{22}}{\dot{W}_{comp,elec}} \quad (44)$$

Condenser (XVI):

$$\varepsilon_{F,con} = \frac{\dot{E}x_{27} - \dot{E}x_{26}}{\dot{E}x_{23} - \dot{E}x_{24}} \quad (45)$$

Expansion valve (XVII):

$$\varepsilon_{U,\exp} = \frac{\dot{E}x_{25}}{\dot{E}x_{24}} \quad (46)$$

Condenser fan (XVIII):

$$\varepsilon_{F,fan,con} = \frac{\dot{E}x_{27} - \dot{E}x_{28}}{\dot{W}_{fan,con,elec}} \quad (47)$$

5. Results and discussion

A series of experiments were performed during the cooling season of 2008–2010 to determine the performance characteristics of the system investigated. All the experiments were carried out during continuous 10 h, from 9.00 a.m. to 7.00 p.m. In the present study, the results obtained from the experiments on 31 August 2009 at 17:00, which were typical, are given and discussed. During the experiments, flow rates of the air streams (fresh, waste and regeneration) are kept constant at 4000 m^3/h. Temperature and relative humidity of the air-conditioned room were adjusted to 26 °C and 50%, respectively according to ASHRAE comfort zone [40]. In the calculations, the dead (reference) state values were considered to be 15 °C and 101.325 kPa. The value for the dead state humidity ratio was taken to be daily mean value of ambient air humidity ratio. The outdoor air temperature is almost 32 °C, while the temperature of 0 °C is also called technical dead state temperature. So, a parametric study was conducted at the dead state temperatures ranging from 0 to 30 °C in order to investigate the effect of the varying dead state temperatures on the exergy efficiency of the system. The thermodynamic properties of water and R134a were found by using Engineering Equation Solver (EES) software package program.

Temperature, pressure and mass flow rate data for air, water and refrigerant R134a are given in Table 2 according to their state numbers specified in Fig. 1. The exergy rates are also calculated for each state as presented in Table 2, while exergy destruction, exergy efficiency, improvement potential rate ($I\dot{P}$) and relative irreversibility (RI) data for representative components of the whole system are given in Table 3.

As can be seen from Table 3 and Fig. 2, the greatest exergy destruction on the system basis occurs in the electric heater unit, followed by the fresh air fan, regeneration air fan, condenser and the other components. It is clear from Table 3 that the highest irreversibility occurs in sub-regions XVI, II and IX with the relative irreversibility of 36.75%, 33.70% and 69.63% for the refrigeration unit, heat exchangers and the whole system, respectively.

The exergy efficiency of the system and the compressor, which is maximum, are calculated to be 21.23% and 94.28%, respectively.

Van Gool's improvement potential on the rate basis ($I\dot{P}$) given in Eq. (12) is calculated for the each component of the system using the values listed in Table 3. It is found that the electric heater unit has the highest $I\dot{P}$ value with 21.442 kW.

As far as a variation of exergetic efficiency of the system with the varying dead (reference) state temperatures of 0–30 °C is concerned (Fig. 3), it is observed that for increasing the dead state temperature, the exergetic efficiencies decrease from round 32% to 10% for the whole system. Fig. 3 also shows the effect of the reference temperature on the SI. The SI value decreases from 1.5 to 1.1 with the increase in the reference temperature.

6. Conclusions

This paper has undertaken a study through both energy and exergy analysis methods along with the sustainability assessment of a novel desiccant cooling system, which was designed and tested

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by the authors. In the analyses, the experimental values were utilized. Some concluding remarks obtained from the results of the present study may be listed as follows:

- Exergy efficiency of the whole system on the exergetic product/fuel basis is calculated to be 21.23% at a reference state temperature of 15 °C.
- On the refrigeration unit basis, the condenser has the maximum exergy destruction rate, followed by the expansion valve, the condenser fan, the compressor and the evaporator.
- The exergetic efficiency values for the whole system are calculated to range from round 32% to 10% at the varying dead (reference) state temperatures of 0–30 °C.
- The SI value decreases from 1.5 to 1.1 with increasing the reference temperature.
- For a future work, an exergoeconomic analysis is recommended to cover both the exergetic and the economical analyses.

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