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# Experimental exergoeconomic assessment of a desiccant cooling system

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ABSTRACT

Desiccant cooling has become a well established technology in most parts of the world, especially recently in Turkey. The increased growth of the technology was caused by the contribution of refrigerants used in conventional cooling systems to the depletion of the ozone layer. This technology provides a tool to control humidity (moisture) levels in conditioned air spaces. In this study, a desiccant cooling system was designed, constructed and tested in Cukurova University, Adana, Turkey while it has been successfully operated since 2008. Exergy, cost, energy and mass (EXCEM) analysis was applied to this system for the first time to the best of the authors' knowledge. The relations between thermodynamic losses and capital costs were also parametrically investigated and illustrated in figures. Based on the overall system (OS) results, some components of the whole system, namely the electric heater unit, the expansion valve, the pump, the fresh air fan and the condenser fan were obtained to be inefficient. Particularly, the electric heater unit was important as its exergy loss rate ($\dot{R}_{ex}$) value was 29.36 times greater than that of the OS.
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## 1. Introduction

Desiccant cooling systems have been widely used in air-conditioning because of their ability to remove moisture from outdoor ventilation air while allowing conventional air-conditioning systems to deal primarily with control temperature (sensible cooling loads). In these systems, a desiccant removes moisture from the air, which releases heat and increases the air temperature. A combination of heat exchange with ambient air and evaporative or conventional cooling coils then cools the dry air. Temperature and humidity loads are very effectively and efficiently met by separating them in this way. The desiccant is then dried out (regenerated) to complete the cycle using thermal energy. The design and operation of a desiccant system are based on the desiccant material used to accomplish the dehumidification. Desiccant materials are those that attract moisture due to differences in vapor pressure. Most people are familiar with desiccants such as silica gel packages that are included with new electronics or textile products. Desiccants can be in the form of a solid or a liquid. These desiccants have been selected based on their ability to hold large quantities of water, their ability to be reactivated, and cost [1–3].

The literature for desiccant technology is rich, and various review studies are also available [3–6]. Desiccant cooling systems are viable alternative to vapor compression systems. Dai et al. [7] presented a comparative study of a standalone vapor compression system (VCS), the desiccant-associated VCS, and the desiccant and evaporative cooling associated VCS. The authors found that the desiccant-associated VCS had more cooling production than the VCS alone by 20–30%. Subramanyam et al. [8] studied a desiccant wheel integrated air-conditioner for low humidity air-conditioning to evaluate its performance and compare with those of conventional and reheat systems. They showed that the proposed system could deliver supply air at much lower dew point temperature compared to the conventional system with a marginal penalty on COP. Elsayed et al. [9] presented a theoretical investigation on the performance of air cycle refrigerator driving air conditioning system integrated desiccant system. They indicated that the system performance was better than the conventional vapor compression air conditioning system with reheating coil. They also reported that the system had a potential to become a good alternative for the conventional vapor compression air conditioning system with low environmental load.

In the recent years, novel systems including the desiccant technology have been experimentally investigated by different researchers. In this regard, Hürdoğan et al. [10,11] conducted an experimental investigation of a novel desiccant based air conditioning system to improve the indoor air quality and reduce energy consumption. They showed that using a heat exchanger for preheating the regeneration air with exhaust air was feasible to install. It was also reported that although COP changed between 0.4 and 4 according to the electric heaters switching on or off, the daily mean

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**Nomenclature**

| | |
|---|---|
| $\dot{E}$ | energy rate, kW |
| $\dot{E}x$ | exergy rate, kW |
| $K$ | capital cost of system, $/GJ |
| $L$ | thermodynamic loss, GJ/kg |
| $\dot{L}$ | thermodynamic loss rate, kW |
| $\dot{m}$ | mass flow rate, kg/s |
| $R$ | ratio of thermodynamic loss rate to capital cost, GJ/$ |

*Subscripts*

| | |
|---|---|
| *a* | accumulation |
| *C* | creation |
| *con* | consumption |
| *eq* | equipment |
| *en* | energy |
| *ex* | exergy |
| *gen* | generation |
| *in* | input, inlet |
| *M* | maintenance |
| *out* | output, outlet |
| *P* | product |
| *W* | waste |

value was 1.35. La et al. [12] proposed a novel rotary desiccant cooling system incorporating two-stage dehumidification and regenerative evaporative cooling. The objective of their study was to extend the ability for handling sensible heat of the rotary desiccant cooling system. Based on the experimental results, the novel chiller was found to be a good choice for space cooling using low-grade heat source. Uçkan et al. [13] developed a novel configuration of desiccant based evaporative cooling system for air conditioning application. The system has a novel design in terms of both air channels and heat exchangers used. It was shown that indoor air conditions are in the range of thermal comfort zone defined by ASHRAE and expanded comfort zone for evaporative air conditioning applications. Enteria et al. [14] evaluated a new solid desiccant heat pump system based on the different outdoor air typical in the Asia–Pacific Region climate with some conditions of the return air (temperature and relative humidity). The result indicated that the system was good in the reduction of air latent energy content compared to the reduction of air sensible energy content.

A number of investigations are reported in the literature regarding of the hybrid desiccant cooling systems. Fatouh et al. [15] presented experimental performance data of a solid desiccant based hybrid air conditioning system. The system consists of a packed bed solid desiccant integrated with a R407C conventional vapor compression refrigeration system. Their results revealed that solid desiccant based hybrid air conditioning system reduces the compressor electric power and the number of electric unit (kW h) by 10.2%. La et al. [16] investigated experimentally and analyzed theoretically a solar hybrid desiccant air conditioning system, which combines the technologies of two-stage desiccant cooling (TSDC) and air-source vapor compression air-conditioning (VAC) together. They showed that the solar hybrid system is feasible to not only humid weather condition but also temperate and extreme humid weather conditions. Ghali [17] simulated numerically the transient performance of a hybrid desiccant vapor compression air conditioning system for the ambient conditions of Beirut. The main feature of this hybrid system was that the regenerative heat needed by the desiccant wheel is partly supplied by the condenser dissipated heat while the rest is supplied by an auxiliary gas heater. Author showed that the payback period of the hybrid system is less than five years when the initial cost of the hybrid air conditioning system priced an additional 1712.00 USD.

Desiccant based air conditioning systems are cost effective systems when a cheap heat source such as solar energy, waste heat or geothermal energy is used to regenerate the desiccant [18–22]. Hürdoğan et al. [23] developed a model to investigate the utilization of solar energy in the desiccant cooling system. The increase in the regeneration air temperature due to solar energy assistance for the days, on which the experiments were carried out, was calculated using the model. The results obtained from the model and the experiments were compared with each other. They showed that utilization of solar energy in the system increased the coefficient of performance (COP) between 50% and 120%.

In recent years, exergy analysis has been very popular in assessing the performance of various energy-related systems, especially due to permitting many of the shortcomings of energy analysis to be overcome. In this context, it is seen a useful tool in identifying the causes, locations and magnitudes of process inefficiencies [24,25]. Various studies have been undertaken by many researchers [26–31] on exergy analysis of desiccant cooling systems. Hürdoğan et al. [32] presented a study by conducting both energy and exergy analyses along with the sustainability assessment of a novel desiccant cooling system. The exergetic efficiency values for the whole system on the exergetic product/fuel basis were calculated to range from round 32% to 10% at the varying dead (reference) state temperatures of 0–30 °C.

In the analysis and design of energy systems, scientific disciplines (mainly thermodynamics) and economic disciplines (mainly cost accounting) are combined together to achieve optimum designs. For energy conversion devices, cost accounting conventionally considers energy based-unit costs. Many researchers have also strongly recommended that costs are better distributed among outputs if cost accounting is based on exergy, which is also a way to a sustainable development. One rationale for this statement is that exergy is often a consistent measure of economic value. In addition, exergy-based economic-analysis methodologies are also available in the literature (e.g., exergoeconomics, thermoeconomics) [24,33].

In this paper, we have conducted a comprehensive exergoeconomic assessment of a desiccant cooling system. The exergoeconomic analysis method combining exergy analysis with economic analysis was performed using EXCEM analysis to give a better perspective on the energy system considered. It also offers a true picture of the production cost of a process. In the recent years, various exergy-based economic analysis methodologies have been used by many investigators [34–38]. The exergoeconomic analysis in this study is based on the methodology proposed by Rosen and Scott [39] while it was utilized by Rosen and Dincer [35,36]. The methodology provides a comprehensive assessment by accounting for the quantities exergy, cost, energy and mass. The main objective of the EXCEM is to investigate costs and thermodynamic losses for devices in energy systems. According to this methodology, it is possible to assess the cost of desiccant cooling systems.

Additionally, it is necessary to point out that the number of the studies conducted on energy and exergy analyses of a desiccant cooling system is relatively low, but no studies have appeared on exergoeconomic analysis of a desiccant cooling system using EXCEM analysis in the open literature to the best of the authors'

knowledge. This provided a main motivation for performing this contribution.

## 2. Description of the system investigated

### 2.1. Experimental set-up

The system was designed, constructed and tested by the authors in Cukurova University, Adana. Adana is the fifth largest city of Turkey with a population of over two million and located in the Eastern Mediterranean region of Turkey. It has coordinates of 36°59'N, 35°16'E. The weather conditions of Adana are hot and very humid during summer time. Cooling seasons extend from mid-May to mid-October. The outdoor air temperature reaches above 30 °C with humidity ratio of above 15 g/kg in summer [40,41].

The schematic and photographic view of a desiccant cooling system studied [10,11,32] are shown in Figs. 1 and 2, respectively. The system consists of a desiccant wheel, heat exchangers, fans, evaporative cooler, electric heater unit and refrigeration unit. Average life times of the system components considered are given in Table 1.

Humidity of the fresh outside air (state 1) is absorbed by the desiccant material of the wheel (I). Sensible cooling of the fresh air is carried out (processes 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



**Fig. 2.** A photographic view of the desiccant cooling system studied.

**Table 1**
Average life time of the system components.

| Component | Life time (year) |
|---|---|
| Dehumidifier | 10 |
| Heat exchangers | 10 |
| Fans | 5 |
| Evaporative cooler | 5 |
| Electric heater unit | 10 |
| Refrigeration unit | 10 |
| Pump | 5 |

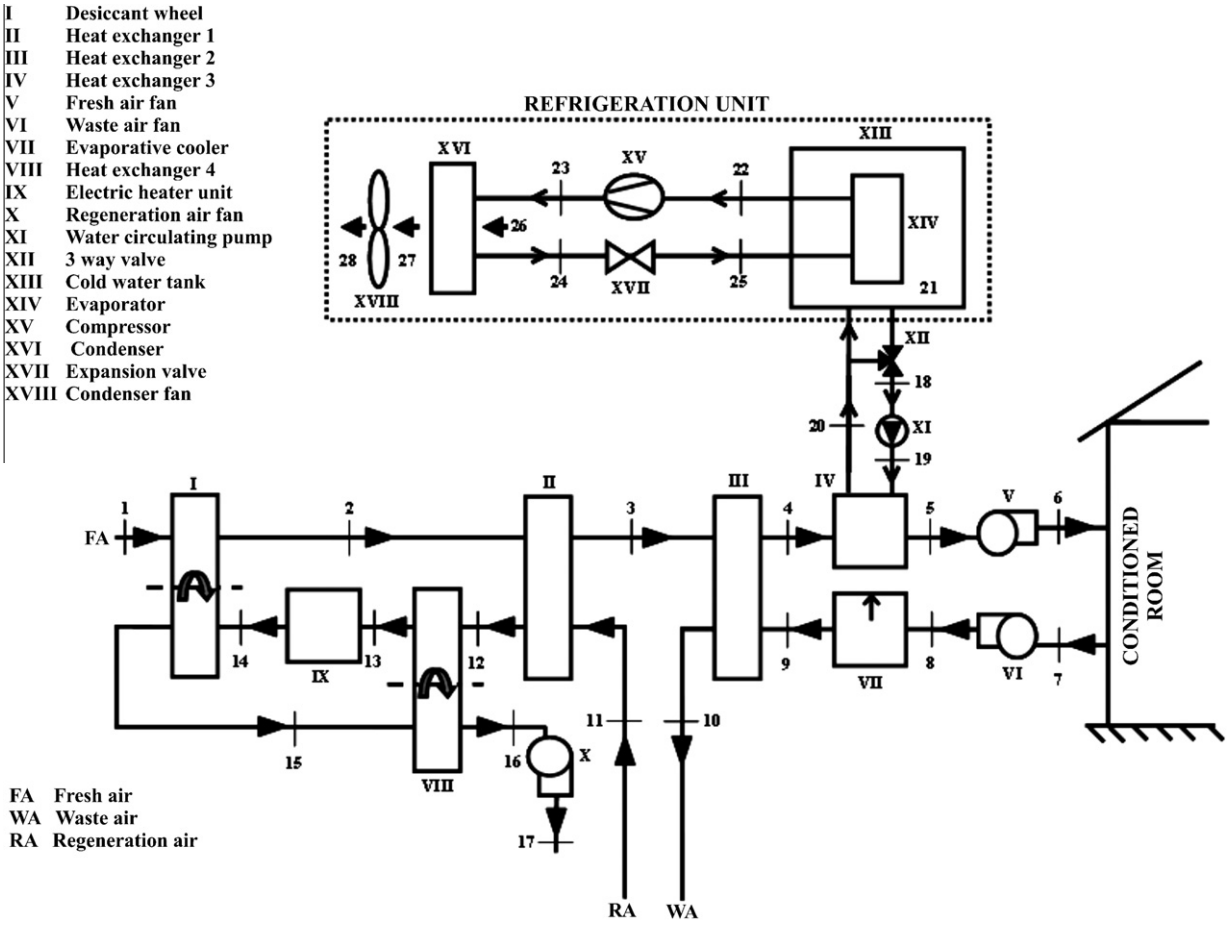



**Fig. 1.** A schematic view of the desiccant cooling system studied.

**Table 2**
Parameters measured during the experiments.

| State no. | Description | Temperature (°C) | Pressure (kPa) | Specific humidity ratio (kg water/kg dry air) |
|---|---|---|---|---|
| 1, 11 | Outdoor | 29.43 | 101.325 | 0.0165 |
| 2 | Dehumidifier outlet/Heat ex. 1 inlet | 51.09 | 101.325 | 0.0102 |
| 3 | Heat ex. 1 outlet/Heat ex. 2 inlet | 39.39 | 101.325 | 0.0102 |
| 4 | Heat ex. 2 outlet/Heat ex. 3 inlet | 25.83 | 101.325 | 0.0102 |
| 5 | Heat ex. 3 outlet/F. fan inlet | 22.13 | 101.325 | 0.0102 |
| 6 | F. fan outlet/Conditioned room inlet | 23.56 | 101.325 | 0.0102 |
| 7 | Conditioned room outlet/W. fan inlet | 26.03 | 101.325 | 0.0105 |
| 8 | W. fan outlet/Evaporative cooler inlet | 26.45 | 101.325 | 0.0105 |
| 9 | Evaporative cooler outlet/Heat ex. 2 inlet | 20.01 | 101.325 | 0.0134 |
| 10 | Heat ex. 2 outlet | 33.48 | 101.325 | 0.0134 |
| 12 | Heat ex. 1 outlet/Heat ex. 4 inlet | 41.02 | 101.325 | 0.0165 |
| 13 | Heat ex. 4 outlet/Elec. heater unit inlet | 46.45 | 101.325 | 0.0165 |
| 14 | Elec. heater unit outlet/Dehumidifier inlet | 69.43 | 101.325 | 0.0165 |
| 15 | Dehumidifier outlet/Heat ex. 4 inlet | 47.65 | 101.325 | 0.0228 |
| 16 | Heat ex. 4 outlet/R. fan inlet | 42.29 | 101.325 | 0.0228 |
| 17 | R. fan outlet | 42.77 | 101.325 | 0.0228 |
| 18 | Cold water tank outlet/Water pump inlet | 19.61 | 101.325 | – |
| 19 | Water pump outlet/Heat ex. 3 inlet | 19.71 | 101.325 | – |
| 20 | Heat ex. 3 outlet/ Cold water tank inlet | 20.93 | 101.325 | – |
| 21 | Cold water tank | 7.40 | 101.325 | – |
| 22 | Evaporator outlet/Compressor inlet | 9.60 | 320 | – |
| 23 | Compressor outlet/Condenser inlet | 76.43 | 1529 | – |
| 24 | Condenser outlet/Expansion valve inlet | 54.49 | 1504 | – |
| 25 | Expansion valve outlet/Evaporator inlet | 3.76 | 335 | – |
| 26 | Condenser inlet (Outdoor) | 29.43 | 101.325 | 0.0165 |
| 27 | Condenser outlet/ Condenser fan inlet | 44.42 | 101.325 | 0.0165 |
| 28 | Condenser fan outlet | 45.61 | 101.325 | 0.0165 |

**Table 3**
Exergy and improvement potential rate data for representative components of the whole system.

| Item number | 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 $\varepsilon$ (%) | Exergetic improvement potential rate $I\dot{P}$ (kW) |
|---|---|---|---|---|---|
| I | 2.330 | 3.570 | 1.240 | 65.27 | 0.431 |
| II | 0.975 | 1.410 | 0.435 | 69.15 | 0.134 |
| III | 0.671 | 0.989 | 0.319 | 67.85 | 0.103 |
| IV | 0.078 | 0.142 | 0.064 | 54.93 | 0.029 |
| V | 0.048 | 2.104 | 2.057 | 2.28 | 2.010 |
| VI | 0.020 | 0.538 | 0.518 | 3.72 | 0.499 |
| VII | 0.069 | 0.512 | 0.442 | 13.48 | 0.382 |
| VIII | 0.632 | 0.906 | 0.274 | 69.76 | 0.083 |
| IX | 3.818 | 29.680 | 25.862 | 12.86 | 22.535 |
| X | 0.054 | 1.770 | 1.716 | 3.05 | 1.664 |
| XI | 0.015 | 0.524 | 0.509 | 2.86 | 0.494 |
| XIV | 0.509 | 0.897 | 0.388 | 56.72 | 0.168 |
| XV | 5.225 | 5.450 | 0.225 | 95.87 | 0.009 |
| XVI | 2.023 | 3.248 | 1.225 | 62.29 | 0.462 |
| XVII | 4.688 | 5.768 | 1.080 | 81.27 | 0.202 |
| XVIII | 0.194 | 1.140 | 0.946 | 17.02 | 0.785 |
| XIV–XVIII | 2.742 | 7.402 | 4.660 | 37.04 | 2.934 |
| II–IV, VIII | 2.356 | 3.447 | 1.092 | 68.35 | 0.346 |
| I–XVIII | 21.348 | 58.647 | 37.301 | 36.40 | 23.723 |

exchangers (II and III) before coming to the cooling coil (heat exchanger 3) for cool recovery.

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. The waste air leaving the heat exchanger is rejected to the outdoors.

The regeneration air first comes to heat exchanger 1 in which heat transfer from the fresh air to the regeneration air takes place. The regeneration air leaving the desiccant wheel passes through the heat exchanger 4 (processes 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 (processes 11–13), it is not high enough for dehumidification of the desiccant wheel. The final temperature of the regeneration air is achieved with the help of electric heaters (IX) to simulate the cheap heat source (processes 13–14). The air removes the humidity of the desiccant wheel (processes 14–15) and flows through heat exchanger 4 (processes 15–16) before discharged to the outdoors.

Temperature, relative humidity, flow rate, electric current and electrical potential difference are measured with appropriate instruments during the experimental tests. More detailed information of these devices and description of Programmable Logic Controller (PLC) used in the control of the system, can be find in the authors previous works [10,11,32].

Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis is performed using the method described by Holman [42]. The total uncertainties of the

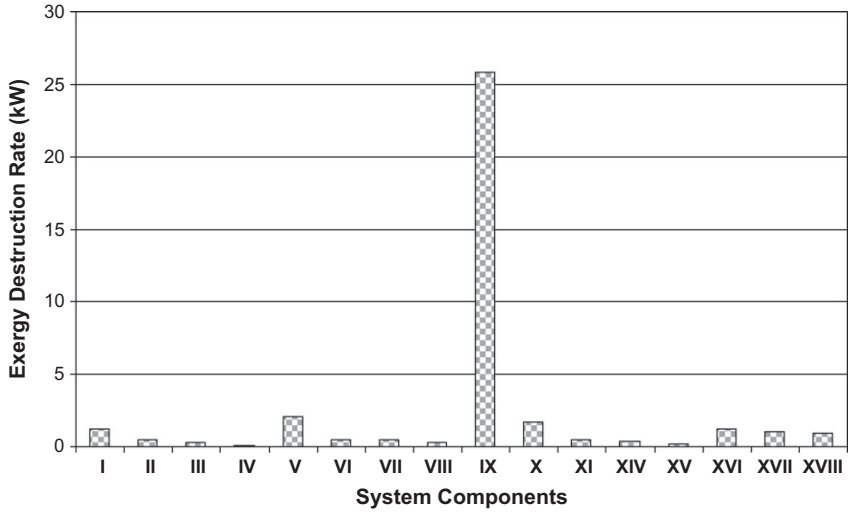



**Fig. 3.** Exergy destruction rate values of the components used in the system.

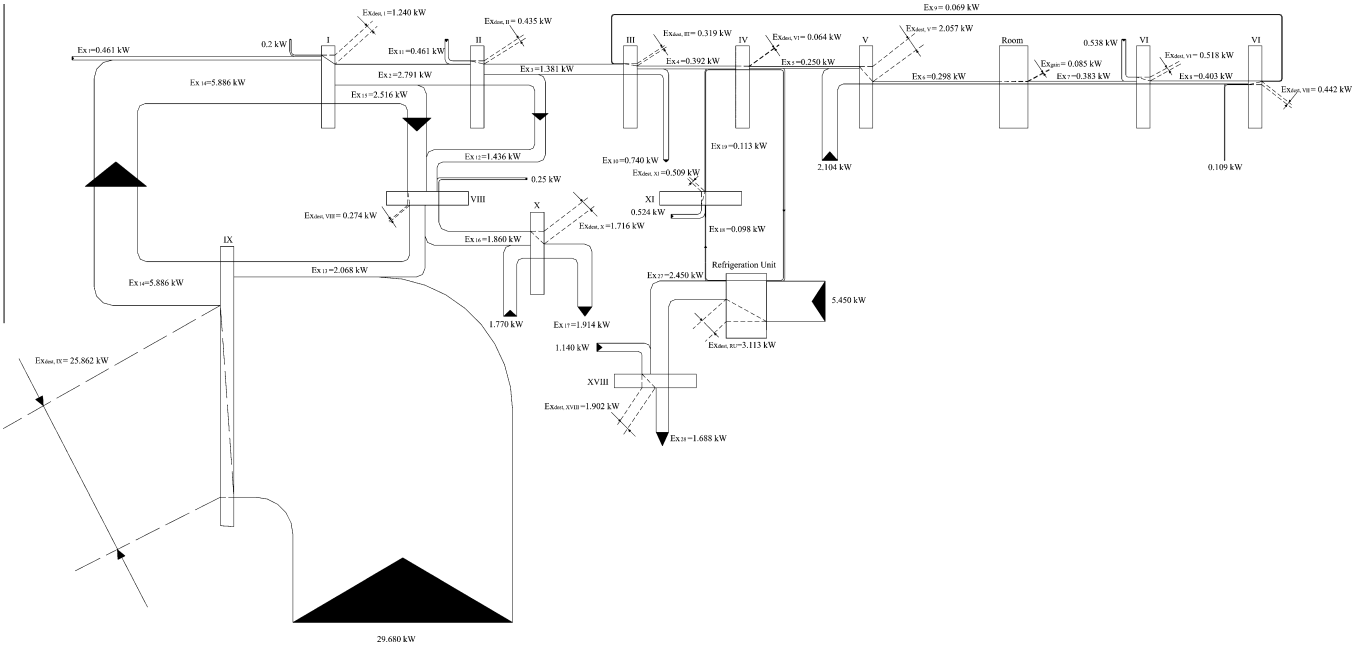



**Fig. 4.** Exergy loss and flow diagram (Grassmann diagram) of the desiccant cooling system.

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 [32].

### 2.2. Modeling and analysis

Here, we have applied the model and assessment methodology outlined earlier by Rosen and Dincer [35,36] to a novel desiccant cooling system. The performance of this system is evaluated using exergy analysis method. The balance equations are written for mass, energy and exergy flows in the system and its components as they are considered steady-state steady-flow control volume system. The appropriate energy and exergy equations are derived for this system and its components. The analysis of the desiccant cooling system is presented in this paper. More detailed description of the system as well as performing energy and exergy calculations can be found in the authors' previous works [10,11,32].

Mass and energy rate balances are subject to first law of thermodynamic (neglecting nuclear reactions), can be neither generated nor consumed.

Consequently, mass flow rate balance for a quantity in a system may be written as:

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

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

Exergy is consumed during the process due to irreversibilities and exergy consumption is proportional to entropy creation [34].

$$\dot{E}x_{in} - \dot{E}x_{out} - \dot{L}_{ex} = \dot{E}x_a \tag{3}$$

**Table 4**
Performance parameters for the desiccant cooling system investigated.

| Item number | Component | $K^a$ (USD) | $\varepsilon$ (%) | $I\dot{P}$ (kW) | $\dot{L}_{ex}$ (kW) | $\dot{R}_{ex}$ (MW/USD) |
|---|---|---|---|---|---|---|
| I | Dehumidifier | 14,720 | 65.27 | 0.431 | 1.240 | 0.084 |
| II | Heat exchanger 1 | 770 | 69.15 | 0.134 | 0.435 | 0.565 |
| III | Heat exchanger 2 | 770 | 67.85 | 0.103 | 0.319 | 0.414 |
| IV | Heat exchanger 3 | 440 | 54.93 | 0.029 | 0.064 | 0.145 |
| V | Fresh air fan | 1000 | 2.28 | 2.010 | 2.057 | 2.057 |
| VI | Waste air fan | 980 | 3.72 | 0.499 | 0.518 | 0.529 |
| VII | Evaporative cooler | 920 | 13.48 | 0.382 | 0.442 | 0.480 |
| VIII | Heat exchanger 4 | 4540 | 69.76 | 0.083 | 0.274 | 0.060 |
| IX | Electric heater unit | 770 | 12.86 | 22.535 | 25.862 | 33.587 |
| X | Regeneration air fan | 1080 | 3.05 | 1.664 | 1.716 | 1.589 |
| XI | Pump | 105 | 2.86 | 0.494 | 0.509 | 4.848 |
| XIV | Evaporator | 1230 | 56.72 | 0.168 | 0.388 | 0.316 |
| XV | Compressor | 3375 | 95.87 | 0.009 | 0.225 | 0.067 |
| XVI | Condenser | 1230 | 62.29 | 0.462 | 1.225 | 0.996 |
| XVII | Expansion valve | 185 | 81.27 | 0.202 | 1.080 | 5.839 |
| XVIII | Condenser fan | 490 | 17.02 | 0.785 | 0.946 | 1.931 |
| XIV–XVIII | Refrigeration unit | 5275 | 76.58 | 0.905 | 3.865 | 0.733 |
| II–IV, VIII | Heat exchangers | 6515 | 68.35 | 0.346 | 1.092 | 0.168 |
| I–XVIII | Overall system | 32,605 | 36.40 | 23.723 | 37.301 | 1.144 |

[a] Exchange rate (selling): 1 USD = 1.6302 Turkish Lira (TL), announced at 3.30 pm on 29 June, 2011 by the Central Bank of Turkey.

Cost is an increasing, nonconserved quantity. The general balance equation can be written for cost as:

$$K_{in} + K_{gen} - K_{out} = K_a \tag{4}$$

where $K_{in}$, $K_{out}$, and $K_a$ represent, respectively, the cost associated with all inputs, outputs and accumulations for the system. $K_{gen}$ corresponds to the appropriate capital and other costs associated with the creation and maintenance of a system.

$$K_{eq} + K_{C,M} = K_{gen} \tag{5}$$

Exergy losses can be identified from the exergy rate balance in Eq. (3). There are two types of exergy losses: the "waste exergy output" which represents the loss associated with exergy that is emitted from the system, and the "exergy consumption" which represents the internal exergy loss due to process irreversibilities. These two exergy losses sum to the total exergy loss. Hence, the loss rate based on exergy, $\dot{L}_{ex}$, is defined as [36]

$$\dot{L}_{ex} = \dot{E}x_{con} + \dot{E}x_{out,W} \tag{6}$$

For a thermal system operating normally in a continuous steady-state steady-flow process mode, the accumulation terms in balance equations are zero. Hence all losses are associated with $\dot{L}_{ex}$. The exergy loss rate can be obtained through the following equations [36]:

$$\dot{L}_{ex} = \sum_{in} \text{Exergy flux rates} - \sum_{p} \text{Exergy flux rates} \tag{7}$$

where the summations are over all input streams and all product output streams.

A parameter, $\dot{R}$ is defined as the ratio of thermodynamic loss rate $\dot{L}$ to capital cost $K$ as follows [36]:

$$\dot{R} = \frac{\dot{L}}{K} \tag{8}$$

The value of $\dot{R}$ generally depends on whether it is based on energy loss rate (in which case it is denoted $\dot{R}_{en}$), or exergy loss rate ($\dot{R}_{ex}$), while in this analysis $\dot{R}_{ex}$ values were used:

$$\dot{R}_{ex} = \frac{\dot{L}_{ex}}{K} \tag{9}$$

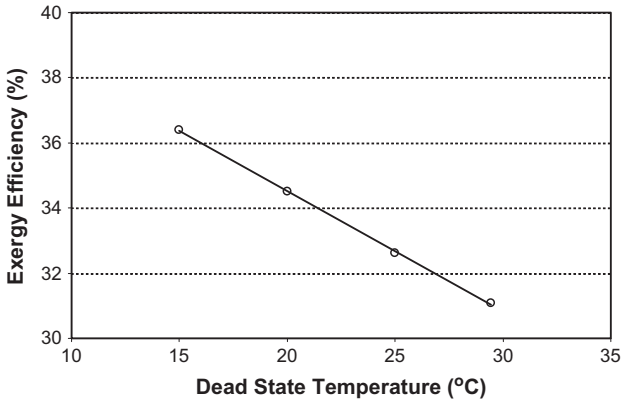



**Fig. 5.** Variation of exergy efficiency values with dead state temperatures for the desiccant cooling system.

## 3. Results and discussion

In the present study, the results obtained from the experiment on 02 September, 2009 at 11:00, 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 [43]. In the calculations, the dead (reference) state values were considered as 15 °C and 101.325 kPa for moist air, water and refrigerant (R134a). The value for the dead state humidity ratio was taken to be daily mean value of ambient air humidity ratio (0.015 kg water/kg dry air). Besides this, a parametric study was conducted at the dead state temperatures ranging from 15 to 29.43 °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.

Mass flow rate for air at states 1–17, water, refrigerant R134a and air at states 26–28 are measured as 1.24, 0.90, 0.147 and 1.67 kg/s, respectively. Measured temperature, pressure and humidity ratio values used for exergy rate calculations are given in Table 2 according to their state numbers. Exergy destruction,

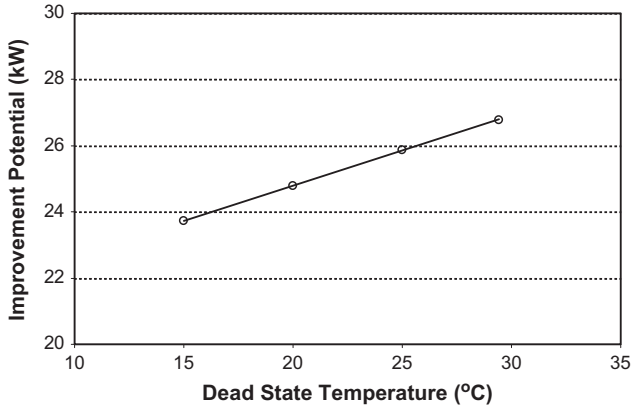



**Fig. 6.** Variation of improvement potential rates with dead state temperatures for the desiccant cooling system.

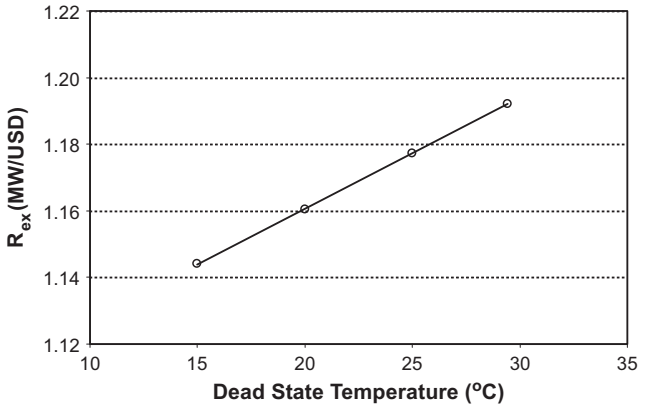



**Fig. 7.** Variation of $\dot{R}_{ex}$ values with dead state temperatures for the desiccant cooling system.

exergy efficiency and improvement potential rate ($I\dot{P}$) data for representative components of the whole system are given in Table 3.

As can be seen from Table 2 and Fig. 3, 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. The exergy efficiency of the system and the compressor, which is maximum, are calculated to be 36.40% and 95.87%, respectively. Van Gool's improvement potential on the rate basis ($I\dot{P}$) 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 22.535 kW.

In this study, relative irreversibility (RI) is also calculated for analysis of the system. It is found that the highest irreversibility occurs in sub-regions XVI, II and IX with the relative irreversibility of 26.29%, 39.84% and 69.33% for the refrigeration unit, heat exchangers and the whole system, respectively.

Fig. 4 illustrates the Grassmann diagram of desiccant cooling system, which gives the quantitative information related to the share of the exergy input to the system.

The main parameters for performing exergoeconomic analysis that were calculated from the experimental data are listed in Table 4. The costs shown in this table are in 2011 US dollars and were obtained based on the 2011 Turkish-$US exchange rate. The exergoeconomic analysis for the system components showed that electric heater unit, expansion valve, pump, fresh air fan and condenser fan were inefficient due to the overall system (OS) results. Particularly, electric heater unit is important as its exergy loss rate ($\dot{R}_{ex}$) value was 29.36 times greater than OS.

According to the authors' previous study [32], the most important system component was the electric heater unit and the results of exergoeconomic analysis also support this argument.

The analyses were performed at different dead state temperatures ranged from 15, 20, 25, 29.43 °C. Figs. 5–7 illustrate variation of exergy efficiency ($\varepsilon$), improvement potential rate ($I\dot{P}$) and $\dot{R}_{ex}$ with different dead state temperatures for desiccant cooling system. First of all, Fig. 7 indicated that the variation of $\dot{R}_{ex}$ was obtained to be linear and increases with the increase of dead state temperature. According to Figs. 5 and 6, it is clear that variation of the parameters with dead state temperature was linear. While the exergetic efficiencies were obtained to vary between 31.08% and 36.40%, improvement potential rates and $\dot{R}_{ex}$ were in the range of 23.72–26.78 kW and 1.14–1.19 MW/USD, respectively (Figs. 6 and 7). It is also obvious from Figs. 5–7 that exergetic efficiency values decreased as the temperature increased to the contrary of the improvement potential rates and $\dot{R}_{ex}$. According to the results, the equations listed below are able to predict the values of exergy efficiency, improvement potential rate and $\dot{R}_{ex}$ for different environment temperatures and obtained correlations were significant with high $R^2$ value of 0.997–1.000:

$$I\dot{P}(T) = 0.213T + 20.533 \tag{10}$$

$$\varepsilon(T) = -0.369T + 41.907 \tag{11}$$

$$\dot{R}_{ex}(T) = 0.003T + 1.094 \tag{12}$$

## 4. Conclusions

In this study, we have applied the EXCEM method to a novel desiccant based air conditioning system and presented the results to assist in the design, improve and optimize the desiccant cooling system. Some concluding remarks may now be drawn from this study as listed below:

- The exergy efficiencies of the system decreased from 36.40% to 31.08% with increasing the reference state temperatures from 15 to 29.43 °C.
- The electric heater unit was found to have the highest $I\dot{P}$ value with 22.535 kW.
- The highest irreversibility occurred in sub-regions XVI, II and IX with the relative irreversibility values of 26.29%, 39.84% and 69.33% for the refrigeration unit, the heat exchangers and the whole system, respectively.
- Improvement potential rates and $\dot{R}_{ex}$ were in the range of 23.72–26.78 kW and 1.14–1.19 MW/USD, based upon the conditions and parameters considered in the present study.
- The authors also expect that the results presented here will be very beneficial to those dealing with combining exergy with economics and focusing on how desiccant systems could be exergoeconomically analyzed and evaluated.

In the analysis we have considered the ambient temperature, the mass flow rate, temperature and relative humidity of the air-conditioned room. It is recommended that some characteristic parameters, such as the desiccant materials, the configuration, the heat recovery and adsorption heat, may be included in a future study to be conducted.

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