Hakan Caliskan[1]
Department of Mechanical Engineering,
Faculty of Engineering,
Usak University,
64200 Usak, Turkey
e-mail: hakan.caliskan@usak.edu.tr

Dae-Young Lee
Center for Urban Energy Research,
Korea Institute of Science and Technology,
Seoul 136-791, South Korea
e-mail: ldy@kist.re.kr

Hiki Hong[1]
Department of Mechanical Engineering,
Kyung Hee University,
Yongin 449-701, South Korea
e-mail: hhong@khu.ac.kr

Assessments of High-Efficient Regenerative Evaporative Cooler Effects on Desiccant Air Cooling Systems

In this paper, the effects of regenerative evaporative coolers on the dry desiccant air cooling system are assessed. Thermodynamic analysis is performed point by point on the unmodified ($\varepsilon = 0.67$) and modified ($\varepsilon = 1$) regenerative evaporative cooler supported systems. It is found that the effectiveness and efficiency of the system were significantly increased by modification. Effectiveness of the system increases from 0.95 to 2.16 for the wet bulb and from 0.63 to 1.43 for dew point effectivenesses, while the exergy efficiency increases from 18.40% to 41.93%. Exergy and energy performances of the system increase 1.28 times and 0.61 times, respectively. Finally, sustainability is increased by 40% with the modification of the regenerative evaporative cooler. Also, changing the regenerative evaporative cooler of the solid desiccant wheel with the effective one can increase the overall system efficiency and performance without changing the sensible heat and desiccant wheels.
[DOI: 10.1115/1.4046523]

Keywords: desiccant air cooling, effectiveness, heat transfer, performance, regenerative evaporative cooler, energy conversion/systems, energy systems analysis, heat energy generation/storage/transfer

1 Introduction

Utilization of energy has been increasing daily due to the rapid increase population and energy consumption. Some of the global carbon emissions occur in the building sector [1,2]. The use of air conditioning systems (e.g., HVAC&R units) is a major reason for energy utilization in constructions. The increase in energy consumption and fossil fuel prices also plays an important role to change the traditional air conditioning systems with effective ones [3,4]. As an important output of HVAC, air conditioning is extensively used in many building areas such as public buildings, shopping malls, and transportation stations/ports. Especially in hot and arid areas, air conditioning is necessary to maintain normal life standards [5–7].

The traditional air conditioning system is the mechanical vapor compression refrigerator in which the refrigerant flows through the evaporator, compressor, condenser, and expansion valve by changing its phase in a loop. This kind of traditional air conditioning systems has some advantages such as low cost, acceptable coefficient of performance (COP) rates (2–4), and life cycle time. On the other hand, there is a high electrical energy necessity for the compressor of these type of systems. Most of the electrical energy in the world is produced by fossil fuels, so traditional air conditioning systems are not environmental friendly and sustainable [5,8,9]. In this regard, the desiccant air conditioning system is considered as an alternative solution in this market by replacing the major sections of the mechanical vapor compression systems [10]. Desiccant air cooling systems are generally known as liquid and dry types. Liquid types have generally conditioning (absorption of incoming air moisture and dehumidification of desiccant) and regeneration (transferring of desiccant moisture to the exhaust air) chambers, and they use the liquid sorbent and cooling material. Solid desiccant dehumidification (also known as desiccant wheel dehumidification) systems use adsorbents, such as silica gel, zeolite, and alumina, to collect moisture by chemically or physically (without phase change). The desiccant is in a rotating wheel turning through the process and regenerating air streams in this process. In the wheel, the first part (desiccant coated) removes the moisture from the air that enters the systems, and the process air becomes drier compared to the initial condition. After the rotation of the wheel, regeneration air in the second section of the wheel takes the desiccant, and the moisture (removed from process air) is expelled by the regenerating air stream. Because of the vapor pressure differences, moisture is transferred. The moisture is trapped in the desiccant (process air stream channel) when the vapor pressure of air is higher than the wheel surface's vapor pressure. If the wheel surface's vapor pressure rises, the desiccant releases moisture (regeneration air stream channel). Differences in the relative humidity in the process and regeneration air streams play an important role in this moisture transfer [11]. Desiccant air cooling systems usually include air dehumidification and sensible cooling processes using evaporative coolers (e.g., regenerative evaporative cooler (REC)). The dehumidification process uses liquid and solid desiccants. Generally, desiccant wheels are chosen due to their simple design, low cost, and effective method [10,12]. For the evaporative cooler section, the regenerative evaporative cooling method is preferred due to its advantage of cooling air under the wet-bulb temperature while the humidity is not rising [13].

There are some studies about the thermodynamic analyses and designs of desiccant air cooling systems and their indirect evaporative cooler in the literature [14–22]. Goldsworthy and White [14] worked on the performance of the solar desiccant indirect evaporative cooler system. The heat and mass transfer analyses were done by focusing on the desiccant wheel's various supply and regeneration air flowrates, and indirect evaporative cooler's various primary and secondary air flowrates. White et al. [15] compared the characteristics of various desiccant wheels that were made from zeolite, superadsorbent polymer, and silica gel for low regeneration

[1]Corresponding authors.

Contributed by the Advanced Energy Systems Division of ASME for publication in the Journal of Energy Resources Technology. Manuscript received August 22, 2019; final manuscript received February 15, 2020; published online February 28, 2020. Assoc. Editor: Luis Serra.

temperatures. Peng et al. [16] simulated the liquid desiccant air cooling system. The thermodynamic performance and the effects of the parameters were determined. Panaras et al. [17] studied the design parameters and their effects on the performance of the solid desiccant air conditioning systems. Enteria et al. [18] applied exergy analysis on the desiccant evaporative air cooling system to determine the performance. The effects of the regeneration temperature on the system parameters were also assessed. Hwang et al. [19] compared the heat pump and sensible wheel-based desiccant cooling systems. It was found that the heat pump based system had higher cooling power and COP rates than the sensible wheel-based system. Chung and Lee [20] studied two different configurations of desiccant cooling systems with the regenerative evaporative cooler, desiccant wheel, sensible heat exchanger, filter, direct evaporative cooler, and fan. The components were the same for both the systems. The only difference was the configuration/location of these components in the system. Numerical analyses were conducted and the performances were compared for these two configurations. Labban et al. [21] compared the conventional vapor compression system, desiccant-based cooling system, and membrane-based cooling system. The outdoor humidity and temperature effects on the performances of the system were also assessed. Caliskan et al. [22] developed a desiccant air cooling system with the desiccant wheel, sensible heat wheel, and evaporative air cooler. The developed system was analyzed by energy, exergy, and sustainability methods and compared with previous studies.

In this study, the REC of the desiccant air cooling system is modified. The effectiveness rate of the modified REC is 1, while the unmodified REC has a 0.67 effectiveness rate. This study is first in the literature to compare the effects of REC on the system parameters. There is no refrigerant in these systems. Hence, the designed systems are environmentally benign. Also, only the fan consumes the major electric power in this design. On the other hand, the system is assessed not only by the first law of thermodynamics but also the second law of thermodynamics.

2 System Description

The schematic of the system is shown in Fig. 1. The overall system's size is approximately $2 \times 1.5 \times 1.5$ m. The designed dry desiccant cooling system comprises sensible heat (SHW), desiccant (DW) wheels, REC, heating coil, fan, and filter units. Alternatively, the regenerative evaporative air cooler is modified by accepting the effectiveness (ε) as the maximum (e.g., Maisotsenko cycle [23–26]). The volume of the desiccant cooling system is generally determined by the sizes of the heat mass exchanger. So, one heating coil is considered in this novel system. On the other hand, the fan consumes major electric power, hence the number of fans is minimized in this design compared to the traditional desiccant air cooling systems.

The regenerative evaporative cooler is an energy-efficient component. It does not include the compressor that consumes energy in vapor compression refrigeration. There is only water as a working fluid in REC, so it is environmentally friendly [27]. The effectiveness of REC can change. In this paper, the effectivenesses of the RECs are 0.67 and 1 for unmodified [22] and modified (Maisotsenko cycle) RECs. The components, except regenerative evaporative coolers, are the same for unmodified and modified systems. There are "process" and "regenerative" channels for "process air" and "regenerative air" streams in the system, respectively. The regenerative evaporative cooler, filter, fan, and one side of the sensible heat and desiccant wheels are present in the process channel, while the regenerative channel includes heating coil, filter, fan, and other sides of the wheels. Air that enters the process channel becomes dry after passing through the desiccant wheel due to water vapor adsorption. After that it is cooled by the sensible heat

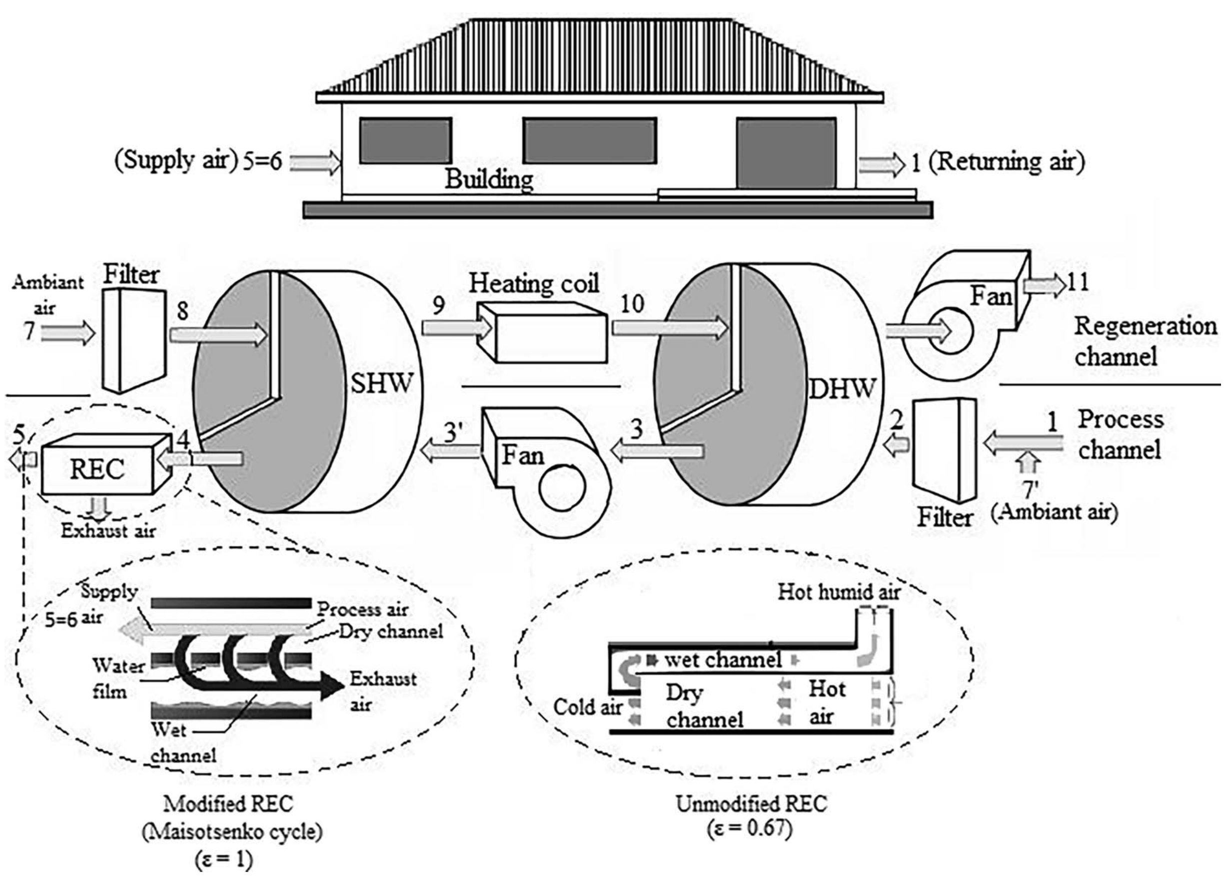

Fig. 1 Schematic of the system

wheel (via heat transfer between process and regeneration channels) and the regenerative evaporative cooler supplies the final cooling quantity required for the air sent to the building (supply air).

An indirect type of evaporative cooling is used in REC in which the air is cooled sensibly and the humidity of the air does not increase. Also, a little amount of cooled air is exhausted. At the same time, the same amount of outdoor air is sent to the entrance part of the process channel of the system. In the sensible heat wheel, process air is pre-cooled and regeneration air is preheated. The preheated (regeneration) air in the regeneration channel is heated and the temperature reaches the regeneration temperature via a heating coil. The wheel uses this regeneration air (hot air in the regeneration channel) for its heat transfer mechanism. Hence, the process air in the process (dehumidification) channel becomes dry and loses its moisture. The desiccant wheel rotates and the moisture is transferred to the regeneration air that is released to the atmosphere. This wheel is made of silica gel material. Its frontal area is 1 m^2, face velocity is 2 m/s, and a fraction of the process and regeneration area is 1/0.7 [19,22]. The psychometric charts of the unmodified and modified systems are shown in Fig. 2. The REC has special channel designs (dry and wet). Process air flows into the dry side of the channel and splits. Some flow is directed to the wet side of the channel with a special design for the pairs (dry/wet), while the other part of the air goes to the dry side of the channel. The extraction ratio of REC is 0.3. In the wet side, the wet surface evaporates and absorbs the heat of the air (dry channel) thus making it cooler. Then, the cooled air is sent, while remaining air from the wet side of the channel is released to the atmosphere. Hence, the air temperature reaches the dew point temperature without an increase in the humidity ratio. For more information on the regenerative evaporative air cooler, see Refs. [22–29].

The measurements were done in the summer season of the Republic of Korea. The environment air conditions are as follows: temperature is 35 °C, pressure is 101.325 kPa, relative humidity is 39.75%, humidity ratio is 13.727 g/kg_{da}, and saturated vapor pressure is 5.63 kPa. There are pressure drops compared to environmental atmospheric pressure such as 0.05 kPa at point 2, 0.25 kPa at point 3, 0.28 kPa at point 3′, 0.05 kPa at point 8, 0.25 kPa at point 9, 0.3 kPa at point 10, and 0.5 kPa at point 11. The air volumetric flow at 1, 5–7, and 8–11 points of the system is 0.823 m^3/s, while it is 1.176 m^3/s at points 2–4, and 0.353 m^3/s at point 7′. The return air's (that leaves the building) temperature is 27 °C for unmodified and modified systems (point 1). In the unmodified system, the temperature is reduced to 19.80 °C by the traditional regenerative evaporative cooler (point 5 = 6). After the modification of the REC, the temperature is reduced to 10.72 °C (point 5 = 6). The electrical energy consumptions of the process channel, regeneration channel, and exhaust fan are 1250 W, 820 W, and 60 W, respectively [22]. Some of the data are taken from Ref. [22] which are of the authors. In the present study, an effective REC is used instead of less-effective REC in Ref. [22] and both REC effects are presented in the current study.

3 Analysis

The energy rate of each point (1–11) in the system ($\dot{E}n_i$) can be calculated by

$$\dot{E}n_i = \dot{m}_i h_i \tag{1}$$

where $\dot{m}_i$ is the air mass flowrate and h_i is the air enthalpy rate. Subscript i means the ith component (points 1–11 in Fig. 1).

The exergy rate of each point (1–11) in the system ($\dot{E}x_i$) is as follows:

$$\dot{E}x_i = \dot{m}_i ex_{tot,i} = \dot{m}_i (ex_{ch,i} + ex_{th,i}) \tag{2}$$

where $ex_{tot,i}$, $ex_{ch,i}$, and $ex_{th,i}$ are the total specific exergy, specific chemical exergy, and specific thermal exergy of air, respectively [22,28,29].

\color{blue}

$$ex_{ch,i} = \dot{m}_i \left[R_a T_0 \left[(1 + ((1.608)\omega_i)) \ln \frac{(1 + ((1.608)\omega_0))}{(1 + ((1.608)\omega_i))} + ((1.608)\omega_i) \ln \frac{((1.608)\omega_i)}{((1.608)\omega_0)} \right] \right] \tag{3}$$

where R_a is the general gas constant, ω is the humidity ratio of air, T is the temperature, and subscript "0" means the environmental condition.

$$ex_{th,i} = \dot{m}_i \left[(c_{p,i} + \omega c_{p,i,v}) \left[T_i - T_0 - T_0 \ln \frac{T_i}{T_0} \right] \right] \tag{4}$$

where $c_{p,i,v}$ and $c_{p,i}$ are specific heats of water vapor and air, respectively.

The cooling capacity of the building ($\dot{Q}_{cool}$) is expressed by

$$\dot{Q}_{cool} = \dot{m}_1 (h_1 - h_6) \tag{5}$$

where the subscripts are connected with the points in Fig. 1.

The cooling capacity exergy rate ($\dot{E}x_{cool}$) is determined from

$$\dot{E}x_{cool} = \dot{Q}_{cool} \left| 1 - \frac{2T_0}{T_1 + T_6} \right| \tag{6}$$

Heat rate with regeneration ($\dot{Q}_{reg}$) is found to be

$$\dot{Q}_{reg} = \dot{m}_9 (h_{10} - h_9) \tag{7}$$

Regeneration exergy rate ($\dot{E}x_{reg}$) is computed as follows:

$$\dot{E}x_{reg} = \dot{Q}_{reg} \left| 1 - \frac{2T_0}{T_9 + T_{10}} \right| \tag{8}$$

The COP rate considering thermally driven units (COP_{th}) is determined from

$$COP_{th} = \frac{\dot{Q}_{cool}}{\dot{Q}_{reg}} = \frac{\dot{m}_1 (h_1 - h_6)}{\dot{m}_9 (h_{10} - h_9)} \tag{9}$$

The exergetic COP of the system considering thermally driven units ($COP_{ex,th}$) is expressed by

$$COP_{ex,th} = \frac{\dot{E}x_{cool}}{\dot{E}x_{reg}} \tag{10}$$

The COP of the system considering electrically driven units (COP_{el}) is calculated as follows:

$$COP_{el} = \frac{\dot{Q}_{cool}}{\dot{W}_{tot}} \tag{11}$$

where W_{tot} is the total electricity consumption.

The exergetic COP of the system considering electrically driven units ($COP_{ex,el}$) is found by

$$COP_{ex,el} = \frac{\dot{E}x_{cool}}{\dot{W}_{tot}} \tag{12}$$

Wet-bulb effectiveness (ε_{wb}) can be computed from [22–24]

$$\varepsilon_{wb} = \frac{T_{1,db} - T_{6,db}}{T_{1,db} - T_{1,wb}} \tag{13}$$

where subscripts db and wb are the dry bulb and wet bulb, respectively ($T_{1,wb} = 19.45$ °C).

Dew point effectiveness (ε_{dp}) is found by [25,26]

$$\varepsilon_{dp} = \frac{T_{1,db} - T_{6,db}}{T_{1,db} - T_{1,dp}} \tag{14}$$

where subscript dp is the dew point (15.58 °C).

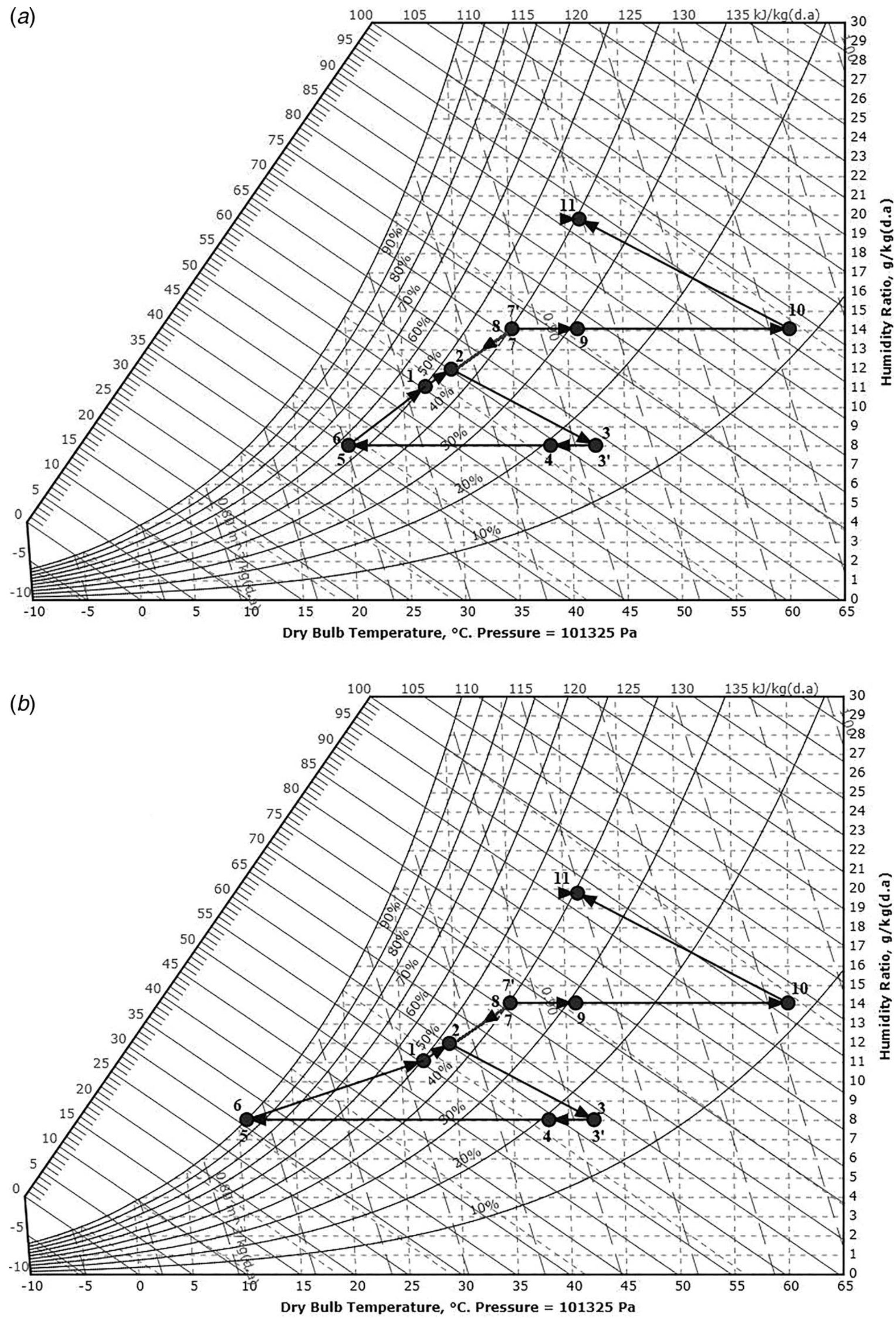

Fig. 2 Psychometric charts of the (a) unmodified and (b) modified systems

Exergy efficiency (Ψ_{sys}) is found by

$$\Psi_{sys} = \frac{\dot{E}x_{cool}}{(\dot{E}x_{10} - \dot{E}x_9) + \dot{W}_{tot}} \tag{15}$$

The dew point effectiveness of the REC ($\varepsilon_{dp,rec}$) is determined by

$$\varepsilon_{dp,rec} = \frac{T_5 - T_4}{T_{4,dp} - T_4} \tag{16}$$

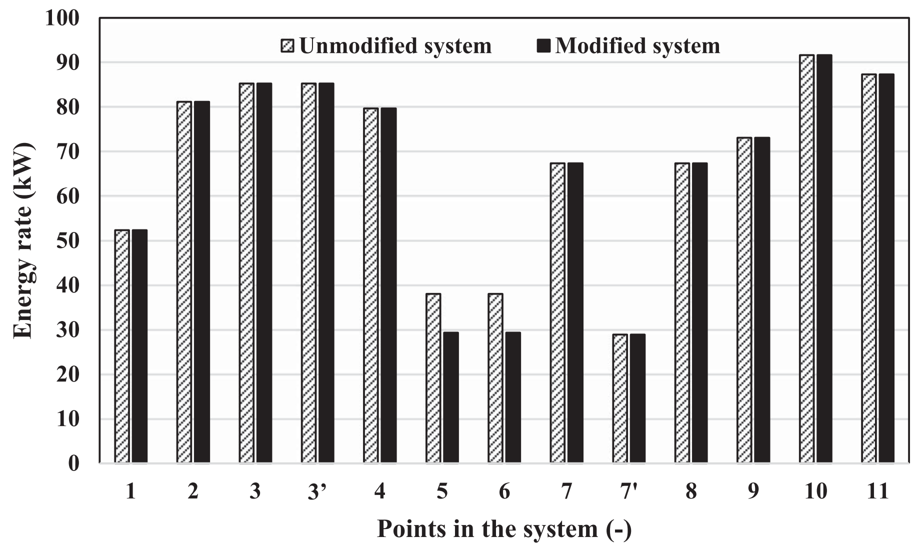

Fig. 3 Energetic results of unmodified and modified system points

The sustainability index (SI) helps to assess/compare the sustainability of the systems. It is calculated by [28]

$$\mathrm{SI} = \frac{1}{1 - \Psi_{sys}} \tag{17}$$

4 Results and Discussion

Energetic results of the unmodified and modified solid desiccant cooling system points are shown in Fig. 3. After the regenerative evaporative cooler (at point 5), the air temperature is decreased to 19.8 °C and the energy rate is 38.02 kW for the unmodified system, while the corresponding temperature is decreased to 10.72 °C and the energy rate is found to be 29.28 kW for the modified system. The other energy rates of the points are the same for both the systems. This is because only the regenerative evaporative cooler, which is the final device before sending the cooled air to the building, is modified.

Exergetic results of the unmodified and modified system points are given in Fig. 4. After the regenerative evaporative cooler, the exergy rate is determined to be 0.56 kW for the unmodified unit, while the corresponding rate is found to be 1.17 kW for the modified unit. The rates at the other points are the same for both the systems. The availability (exergy) of the air, before entering the building, for the modified system is more than the unmodified system.

Energetic and exergetic performances of the systems can be seen in Fig. 5. According to the energy analysis assessment, the electrically driven COP rates are found to be 6.71 and 10.81, while the thermally driven COP rates are calculated to be 0.77 and 1.24 for the unmodified and modified systems. According to exergy analysis assessment, the electrically driven exergetic COP rates are found to be 0.26 and 0.60, while the thermally driven exergetic COP rates are calculated to be 0.63 and 1.44 for the unmodified and modified systems. The modified system has higher energetic and exergetic COP rates considering all possible options as electrically and thermally driven.

The regenerative evaporative cooler is modified and its effectiveness is increased from 0.67 (unmodified) to 1 (modified). The effectiveness and exergy efficiencies of the unmodified and modified solid desiccant cooling systems are presented in Fig. 6. The wet-bulb effectivenesses of the unmodified and modified systems are found to be 0.96 and 2.16, respectively; also the corresponding dew point effectivenesses are computed to be 0.63 and 1.43, respectively. Exergy efficiencies are calculated to be 18.40% and 41.93%. On the other hand, the sustainabilities of the unmodified and modified systems are 1.23 and 1.72, respectively. According to energetic

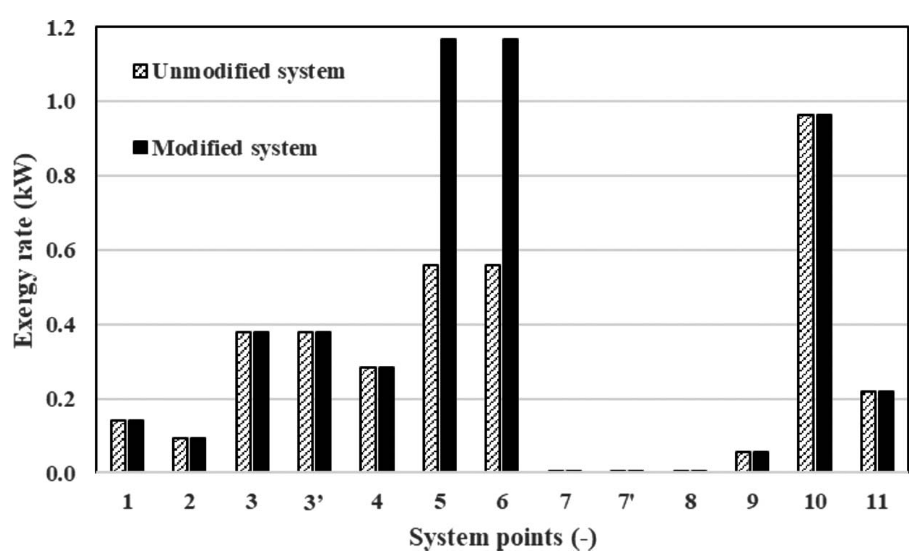

Fig. 4 Exergy analysis results of the unmodified and modified system points

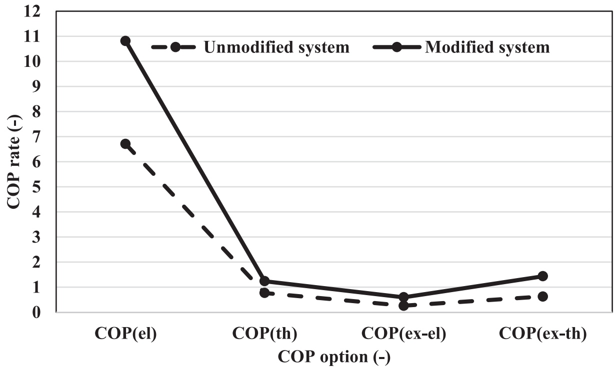

Fig. 5 Energetic and exergetic performances of the systems

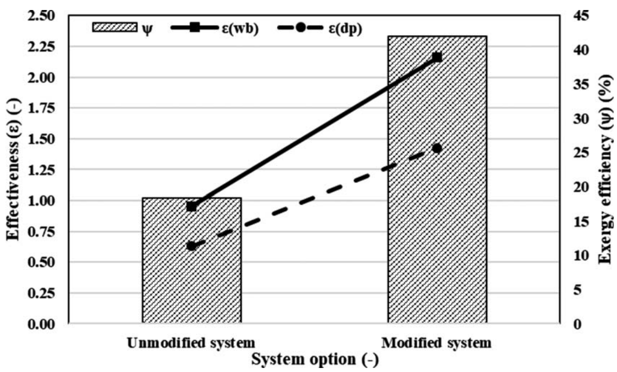

Fig. 6 Effectiveness and exergy efficiencies of the unmodified and modified solid desiccant cooling systems

effectiveness and exergetic efficiency approaches, the modified system is more efficient and sustainable than the unmodified system.

5 Conclusions

The regenerative evaporative cooler is modified and its effects on the dry desiccant air cooling unit are assessed. Thermodynamic analysis is performed point by point on the unmodified and modified systems. The effectiveness of the REC is increased from 0.67 (unmodified) to a maximum (modified). It is found that system effectiveness and efficiency significantly increased. The increase in system effectiveness is significant (from 0.95 to 2.16 for the wet bulb and from 0.63 to 1.43 for dew point effectivenesses), while the increase in the exergy efficiency is high (from 18.40% to 41.93%). Performances (COP rates) of the solid desiccant cooling systems increase drastically by using the modified REC. Sustainability is increased by 40% with the modification of the REC. Changing REC of the solid desiccant wheel with the effective one (e.g., Maisotsenko cycle based REC) can increase the overall system efficiency and performance without changing the sensible heat and desiccant wheels.

References

[1] Zhu, J., and Chen, W., 2014, "Energy and Exergy Performance Analysis of a Marine Rotary Desiccant Air-Conditioning System Based on Orthogonal Experiment," Energy, **77**, pp. 953–962.

[2] Aridhi, E., Albouchi, A., and Mami, A., 2017, "Observability and Controllability Study of a Household Refrigerator Exposed to an Outdoor Cold Airflow Using Bond Graph Approach," Int. J. Air-Conditioning Refrig., **25**(1), p. 1750001.

[3] Pandelidis, D., Anisimov, S., Worek, W. M., and Drąg, P., 2016, "Comparison of Desiccant Air Conditioning Systems With Different Indirect Evaporative Air Coolers," Energy Convers. Manage., **117**, pp. 375–392.

[4] Jradi, M., and Riffat, S. B., 2014, "Experimental and Numerical Investigation of a Dew-Point Cooling System for Thermal Comfort in Buildings," Appl. Energy, **132**, pp. 524–535.

[5] Duan, Z., Zhan, C., Zhang, X., Mustafa, M., Zhao, X., Alimohammadisagvand, B., and Hasan, A., 2012, "Indirect Evaporative Cooling: Past, Present and Future Potentials," Renew. Sustain. Energy Rev., **16**(9), pp. 6823–6850.

[6] Enteria, N., Yoshino, H., Mochida, A., Takaki, R., Satake, A., Baba, S., and Tanaka, Y., 2015, "Performance Test of Desiccant Heating, Ventilating and Air-Conditioning System by Using Multiple Tracer Gas Dilution Method," Int. J. Air-Conditioning Refrig., **23**(04), p. 1550027.

[7] Enteria, N., Yoshino, H., and Sataki, A., 2016, "Exergoenvironmental Evaluation of the Desiccant Air-Conditioning System Subjected to Different Regeneration Temperatures," Int. J. Air-Conditioning Refrig., **24**(04), p. 1650023.

[8] Zhang, L. Z., 2006, "Energy Performance of Independent Air Dehumidification Systems With Energy Recovery Measures," Energy, **31**(8–9), pp. 1228–1242.

[9] Ahmadu, T. O., Folayan, C. O., and Anafi, F. O., 2016, "Modeling, Simulation and Optimization of a Solar Absorption Air Conditioning System for an Office Block in Zaria, Nigeria," Int. J. Air-Conditioning Refrig., **24**(02), p. 1650012.

[10] Gao, W., Worek, W., Konduru, V., and Adensin, K., 2015, "Numerical Study on Performance of a Desiccant Cooling System With Indirect Evaporative Cooler," Energy Buildings, **86**, pp. 16–24.

[11] Polygon, 2017, https://www.polygongroup.com/en-US/blog/liquid-desiccant-vs-desiccant-wheel-dehumidification–whats-the-difference/

[12] Zhu, J., and Chen, W., 2015, "A Novel Multivariate Linear Prediction Model for the Marine Rotary Desiccant Air-Conditioning by Adding a Dynamic Correction Factor," Appl. Therm. Eng., **78**, pp. 101–109.

[13] Lee, J., Choi, B. S., and Lee, D. Y., 2013, "Comparison of Configurations for a Compact Regenerative Evaporative Cooler," Int. J. Heat Mass Transf., **65**, pp. 192–198.

[14] Goldsworthy, M., and White, S., 2011, "Optimisation of a Desiccant Cooling System Design With Indirect Evaporative Cooler," Int. J. Refrig., **34**(1), pp. 148–158.

[15] White, S. D., Goldsworthy, M., Reece, R., Spillmann, T., Gorur, A., and Lee, D.-Y., 2011, "Characterization of Desiccant Wheels With Alternative Materials at Low Regeneration Temperatures," Int. J. Refrig., **34**(8), pp. 1786–1791.

[16] Peng, D., Zhou, J., and Luo, D., 2017, "Exergy Analysis of a Liquid Desiccant Evaporative Cooling System," Int. J. Refrig., **82**, pp. 495–508.

[17] Panaras, G., Mathioulakis, E., and Belessiotis, V., 2011, "Solid Desiccant Air-Conditioning Systems—Design Parameters," Energy, **36**(5), pp. 2399–2406.

[18] Enteria, N., Yoshino, H., Takaki, R., Mochida, A., Satake, A., and Yoshie, R., 2013, "Effect of Regeneration Temperatures in the Exergetic Performances of the Developed Desiccant Evaporative Air-Conditioning System," Int. J. Refrig., **36**(8), pp. 2323–2342.

[19] Hwang, W. B., Choi, S., and Lee, D. Y., 2017, "In-Depth Analysis of the Performance of Hybrid Desiccant Cooling System Incorporated With an Electric Heat Pump," Energy, **118**, pp. 324–332.

[20] Chung, J. D., and Lee, D.-Y., 2011, "Contributions of System Components and Operating Conditions to the Performance of Desiccant Cooling Systems," Int. J. Refrig., **34**(4), pp. 922–927.

[21] Labban, O., Chen, T., Ghoniem, A. F., Lienhard V, J. H., and Norford, L. K., 2017, "Next-Generation HVAC: Prospects for and Limitations of Desiccant and Membrane-Based Dehumidification and Cooling," Appl. Energy, **200**, pp. 330–346.

[22] Caliskan, H., Lee, D. Y., and Hong, H., 2019, "Enhanced Thermodynamic Assessments of the Novel Desiccant Air Cooling System for Sustainable Energy Future," J. Cleaner Prod., **211**, pp. 213–221.

[23] Caliskan, H., Dincer, I., and Hepbasli, A., 2011, "Exergetic and Sustainability Performance Comparison of Novel and Conventional Air Cooling Systems for Building Applications," Energy Buildings, **43**(6), pp. 1461–1472.

[24] Caliskan, H., Hepbasli, A., Dincer, I., and Maisotsenko, V., 2011, "Thermodynamic Performance Assessment of a Novel Air Cooling Cycle: Maisotsenko Cycle," Int. J. Refrig., **34**(4), pp. 980–990.

[25] Caliskan, H., Dincer, I., and Hepbasli, A., 2012, "A Comparative Study on Energetic, Exergetic and Environmental Performance Assessments of Novel M-Cycle Based Air Coolers for Buildings," Energy Convers. Manage., **56**, pp. 69–79.

[26] Caliskan, H., Dincer, I., and Hepbasli, A., 2012, "Exergoeconomic, Enviroeconomic and Sustainability Analyses of a Novel Air Cooler," Energy Buildings, **55**, pp. 747–756.

[27] Lee, J., and Lee, D., 2013, "Experimental Study of a Counter Flow Regenerative Evaporative Cooler With Finned Channels," Int. J. Heat Mass Transf., **65**, pp. 173–179.

[28] Dincer, I., and Rosen, M. A., 2007, *Exergy: Energy Environment and Sustainable Development*, Elsevier, New York.

[29] Shukuya, M., and Hammache, A., 2002, *Introduction to the Concept of Exergy—for a Better Understanding of Low Temperature Heating and High Temperature Cooling Systems*, Julkaisija Utgivare, Espoo. VTT Research Notes 2158, LOWEX.