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A Liquid Desiccant Enhanced Two Stage Evaporative Cooling System—Development and Performance Evaluation of a Test Rig

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Abstract: Desiccant technology is found to be a good alternative to conventional cooling systems. It can provide better thermal comfort under hot and humid climatic conditions. The major component of a liquid desiccant cooling system is the desiccant dehumidifier which controls the latent cooling load. In this paper, a newly developed liquid desiccant enhanced evaporative cooling system has been tested experimentally. The effects of ambient conditions and other parameters on the performance of the system are investigated. The system performance curves which help to determine the air outlet conditions and coefficient of performance (COP) of the system are drawn for a wide range of ambient air humidity ratios (0.010–0.026 kg/kg), ambient air temperature (25–40 °C), process air flow rate (1.5–8.0 kg/m^2·s), regeneration air flow rate (1.5–4.5 kg/m^2·s), and regeneration temperature (55–85 °C). The results showed that better supply air conditions are achieved for hot and humid climatic conditions with effectiveness of the system largely dependent on process and regeneration air flow rates, regeneration temperature, and humidity ratio of process air. The dehumidification performance is increased by 62% for a change of ambient air humidity ratio from 0.01 to 0.025 kg/kg. The thermal coefficient of performance improved by 50% for the above variation in humidity ratio. This shows that such thermally activated systems are feasible options for hot and humid climatic conditions as indicated by better performance under these conditions.

Keywords: liquid desiccant; evaporative cooler; air conditioning; rotary dehumidifier; sustainable development

1. Introduction

Statistics prove that the performance of an individual human being is more effective in a conditioned space compared to that in an untreated environment, as a result of which the thermal comfort requirement becomes essential because people spend most of the time in confined environments. Human comfort conditions are achieved when temperature and moisture are held within a certain narrow range. The acceptable ranges of temperature and relative humidity for human comfort as given by the ASHRAE standards 55 are 20–26 °C and 30–60%, respectively. This increases the demand of air conditioning, both in the residential and commercial sectors. The use of conventional cooling system consumes large amount of energy to fulfill thermal comfort conditions. Furthermore, this technology is not energy efficient for climatic conditions where latent loads are dominant [1].

Some alternative technology is required in order to overcome the drawbacks and rising demands of conventional cooling systems for residential as well as commercial buildings [2,3]. The regional demand of heating, ventilation, and air conditioning (HVAC) equipment is presented in Figure 1 [4]. It can be observed that the HVAC equipment demand is growing at a rapid rate for all regions. Secondly, the consumption of primary energy resources has increased significantly in recent years because of growing demand. An effective alternative is desiccant cooling technology. Due to the affinity to absorb/adsorb moisture, desiccants like silica gel, SiO_2, and zeolite attract moisture without any change in their chemical and physical composition. The desiccant material is regenerated using hot air so that the cycle can be repeated. After the dehumidification of the process air, the evaporative cooler is used to lower the temperature according to the desired comfort conditions [5]. Desiccant cooling system is an effective and cost-efficient way to provide comfort in hot and humid environments. This technology has many potential advantages over other cooling techniques such as, better supply conditions, energy efficient, effective utilization of low grade energy sources, etc. [6].

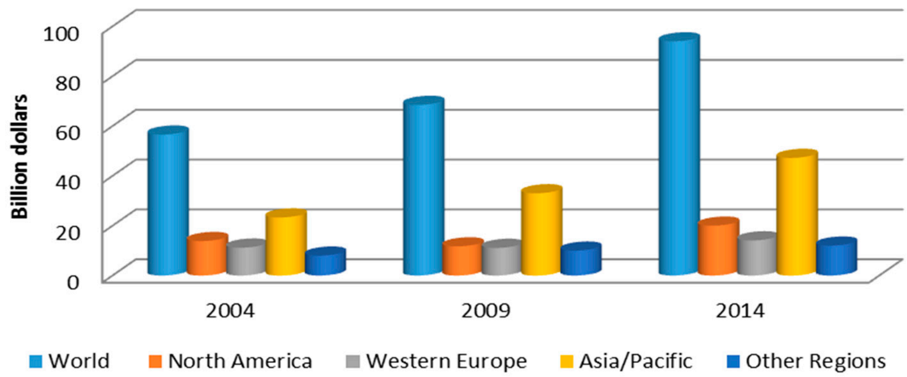


Figure 1. HVAC equipment demand and annual growth [4].

The desiccant cooling technology can be categorized based on the type of desiccant material used. Solid- and liquid-based desiccant cooling are two major types of this thermally driven technology. The solid desiccant cooling technology is more mature as compared to liquid desiccant. Liquid desiccants have advantage over the solid desiccants in that these require lower temperature heat sources (60–85 °C) for regeneration. This makes the usage of renewable energy resources like solar, biomass, etc. more feasible and effective [7]. However, most of the studies carried out on liquid desiccant dehumidification systems are direct contact type in which there is a direct contact between liquid desiccant and the air streams. Direct contact type systems have the drawback that the supply air can carry some liquid desiccant droplets with it and can cause corrosion in the ducting and other equipment. This can increase the maintenance costs and decrease the equipment life. Secondly, desiccant carryover can affect adversely the quality of air and human health. Direct contact type systems also require high fan and pumping power because of the higher pressure drop and continuous supply of desiccant solution.

Different configurations are employed to overcome the problem of desiccant carryover and issues related to packed bed liquid desiccant dehumidifiers [8]. The use of internally cooled/heated [9,10] and membrane type [11] dehumidifiers are few of the solutions to the problem of desiccant carryover [9,10]. The rotary type liquid desiccant cooling system can also be designed to overcome drawbacks of packed bed liquid desiccant cooling systems. These wheels have large surface area and less pressure drops as compared to packed bed dehumidifiers. Some general advantages of using rotary desiccant wheel as a dehumidification unit are:

- These systems are more convenient to install.

- The process of dehumidification and regeneration are synchronized in these systems and operation is continuous.
- For same contact surface area, these systems occupy a smaller space compared to packed bed dehumidifiers.
- Dehumidification/regeneration capacity of the dehumidifier can be efficiently controlled by changing the rotational speed of the desiccant wheel.

In the past, most of the investigations were carried out for the solid desiccant rotary dehumidifiers and no in-depth work is available for energy analysis of rotary liquid desiccant dehumidifiers. The purpose of this research is to develop rotary type dehumidifier using a liquid desiccant material instead of solid desiccant to lower the required input heat for the operation of the system. In this paper, an experimental unit is designed and built to study the performance of the rotary liquid desiccant dehumidifier operating in conjunction with two stage evaporative cooler considering different parameters. The combination of direct and indirect evaporative cooler is used as a cooling unit. The experimental results show that better supply air conditions could be obtained to achieve human comfort in the hot and humid climate with effectiveness of the system largely dependent on air flow rate, regeneration temperature and humidity ratio of the process air. The developed rotary liquid desiccant cooling system is expected to overcome the disadvantages of liquid desiccants.

2. Two Stage Evaporative Cooling

In a desiccant-based cooling system, the latent load is controlled by the desiccant dehumidifier whereas for sensible load evaporative coolers can be utilized. Different configurations of evaporative cooler can be used in conjunction with desiccant dehumidifier depending upon the climatic conditions. The two basic evaporative cooler configurations are the direct and indirect type. Two stage evaporative coolers are an advanced technique mostly used for hot and humid climatic conditions. In hot and humid conditions, it is difficult to cool the air using only an indirect evaporative cooler and the use of larger size direct evaporative coolers will not be an economical option. Two stage indirect-direct evaporative coolers can cool the air to lower temperatures as compared to one stage evaporative coolers [12]. This cooling option is much more energy efficient than cooling with refrigerants. According to ASHRAE, 60–75% savings on electricity can be achieved by replacing conventional vapor compression cooling systems with advanced two stage evaporative coolers.

Many researchers have investigated the use of the two stage evaporative cooler configuration as a standalone unit and in conjunction with desiccant dehumidifiers. Farahani et al. [13] investigated a two stage evaporative cooling system for the climatic conditions of Tehran (Iran). The results demonstrated that the use of two stage evaporative cooling can fulfill the human comfort demands efficiently. El-Dessouky [14] developed and tested a modified two stage evaporative cooler as an alternative to single stage evaporative cooling. The results showed that efficiency of two stage evaporative cooling system is better than a single stage evaporation system. Al-Sulaiman et al. [15] also utilized two stage evaporative cooling system with liquid desiccant dehumidifier. Tashtoush et al. [16] obtained 20% COP with two stage evaporative cooler than that achieved when employing either direct or indirect evaporative cooler systems alone. Different researchers have utilized multistage evaporative cooling systems to achieve desired supply conditions for thermal comfort [17,18]. Rafique et al. [1] mentioned that the effectiveness of direct, indirect and indirect-direct evaporative coolers is 80–90%, 85%, and 110%, respectively. Furthermore, the energy saving potential of two stage evaporative coolers is better compared to single stage evaporative coolers.

3. Materials and Methods

The developed desiccant wheel is a rotor with radius R and width L. A number of narrow slots covered with porous media carrying liquid desiccant solution are distributed uniformly over the cross-section of the rotor as shown in Figure 2. The wheel is divided into two sections for process and regeneration air with both flowing in a counter-current arrangement. The process air is dehumidified after passing through the desiccant wheel. On the regeneration side, air is heated up to the required regeneration temperature and then is passed through the desiccant wheel to desorb moisture from the desiccant material.

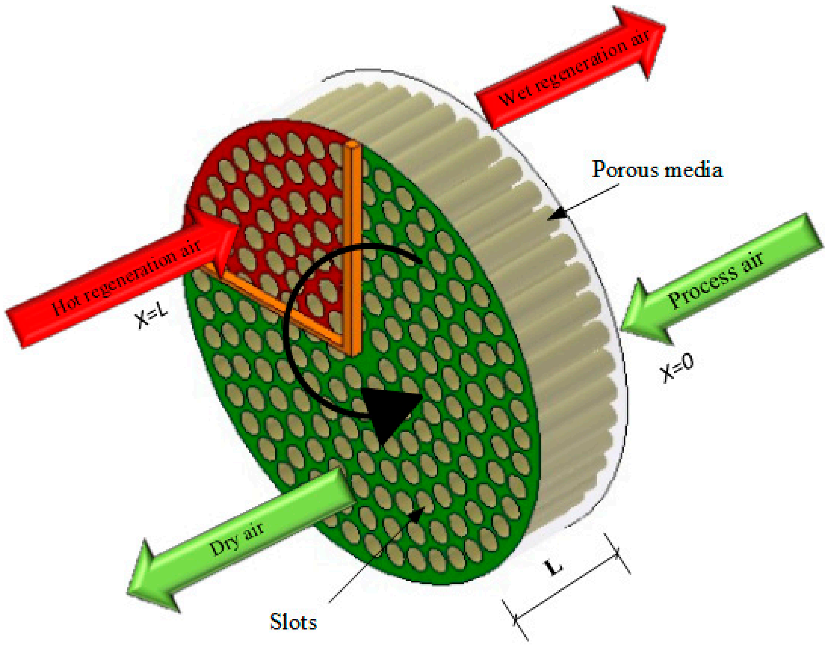


Figure 2. Schematic of rotary liquid desiccant dehumidifier.

Figure 3a illustrates the schematic of the designed and developed desiccant enhanced evaporative cooling system. The system comprises of two major parts: the rotary liquid desiccant dehumidifier and the cooling unit, including direct and indirect evaporative coolers. The dehumidifier is a heat and mass exchanger in which moisture is absorbed from process air by a desiccant material. The dehumidified air is passed through the indirect evaporative cooler and the temperature of the air is lowered using cooling water. At this stage, there is no direct contact between air and cooling water. The air is passed through the tubes with cooling water sprayed over them. For further cooling, air is passed through a direct evaporative cooler in which there is a direct contact between air and cooling water. The process air is then supplied to the conditioned room. In order to remove the moisture from the desiccant dehumidifier for continuous operation of the cycle, the ambient air is heated in an electrical heating system. This hot air from the heater is then passed through the rotary dehumidifier. The rotary dehumidifier rotates at very low speed for proper absorption and desorption of the moisture. The generalized psychrometric representation of complete cycle is illustrated in Figure 3b. The process air is dehumidified and its temperature is increased from state 1 to 2. From state point 2 to 3, only temperature decreases whereas from state 3 to 4, the temperature further decreases while the humidity increases. On the regeneration side, air is heated sensibly from state 1 to 5 and removes the absorbed moisture from the desiccant wheel from state 5 to 6. In this test setup, electric heater is employed to observe the performance of the system but for a real system, a renewable energy such as solar can be employed to achieve required temperature of regeneration (60–85 °C).

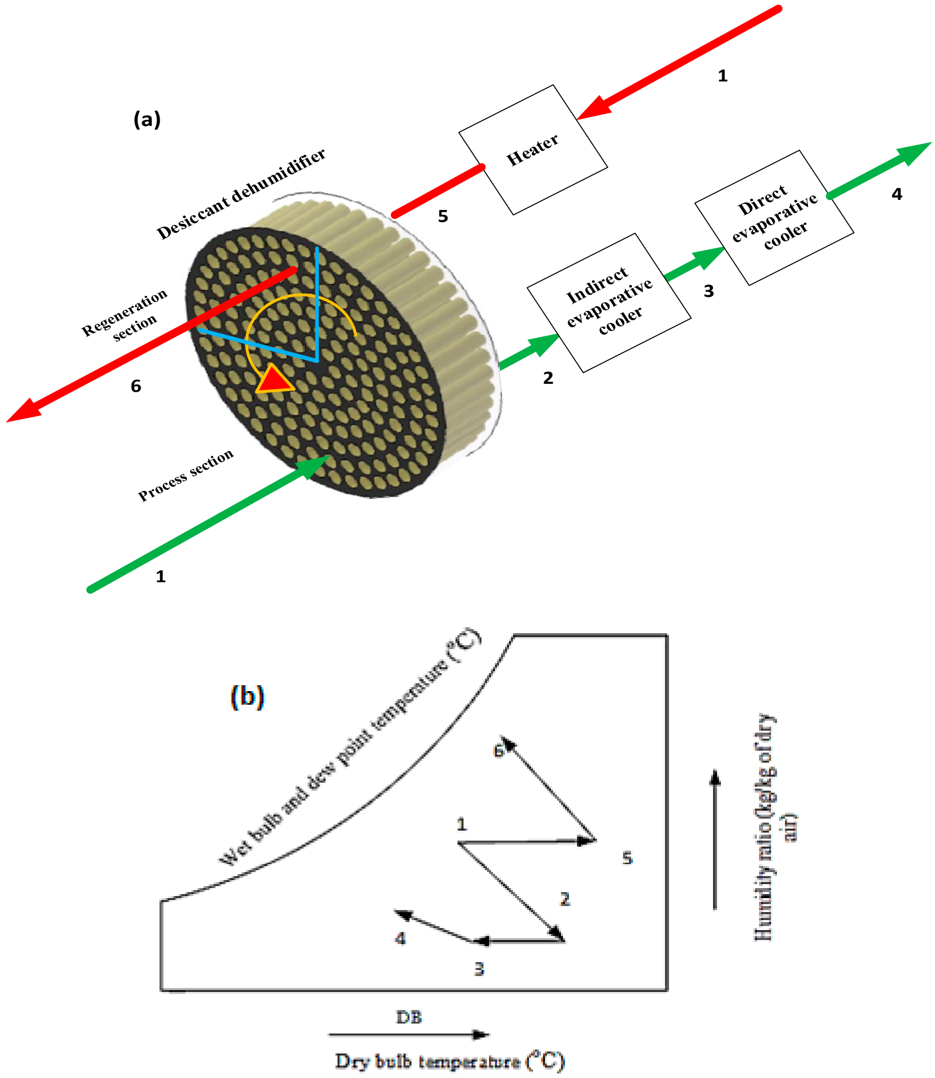


Figure 3. (**a**) Schematic of developed desiccant enhanced cooling cycle; (**b**) Representation of liquid desiccant cooling cycle on psychrometric chart.

4. Test Chamber

A detailed schematic of the experimental setup is shown in Figure 4. The test facility mainly consists of the following components:

- A rotary liquid desiccant dehumidifier (30 cm diameter).
- A direct type evaporative cooler (30 × 60 cm^2).
- An indirect type evaporative cooler (30 × 60 cm^2).
- A water spraying system (0.37 kW pump).
- A variable capacity electric heater (0–4 kW).
- A variable speed electric motor (0–4 rpm).
- Two air blowers.

The details of all other supporting and operation accessories are shown in Figure 5. The cylindrical shape rotating desiccant wheel of 30 cm diameter is fabricated from flexi glass. Flow area is created using pipes of chlorinated polyvinyl chloride of diameter 21 mm with minimum spacing between two pipes. These pipes are uniformly distributed over the cross-section of the wheel. To form the absorbing surface, a wick cloth layer impregnated with calcium chloride solution is placed inside the pipes using springs. The variable speed motor is used to vary the rotational speed of the desiccant wheel. The regeneration heat is provided using a variable capacity electric heater. In the process air stream, after the desiccant dehumidifier direct and indirect type evaporative coolers are used for cooling purposes. The packing material for the direct evaporative cooler is composed of cellulose pad. The size of the pad is 15 cm deep 30 cm wide and 60 cm high. The water is distributed using spray nozzles at the top of the pad. In indirect evaporative cooler, the cooling water is circulated inside the coils and the air cools as it passes across the tubes. The indirect evaporative cooler coils are constructed from copper tubes with bonded aluminum fins. Fins are attached to the outside of tubes to enhance the surface area for heat transfer.

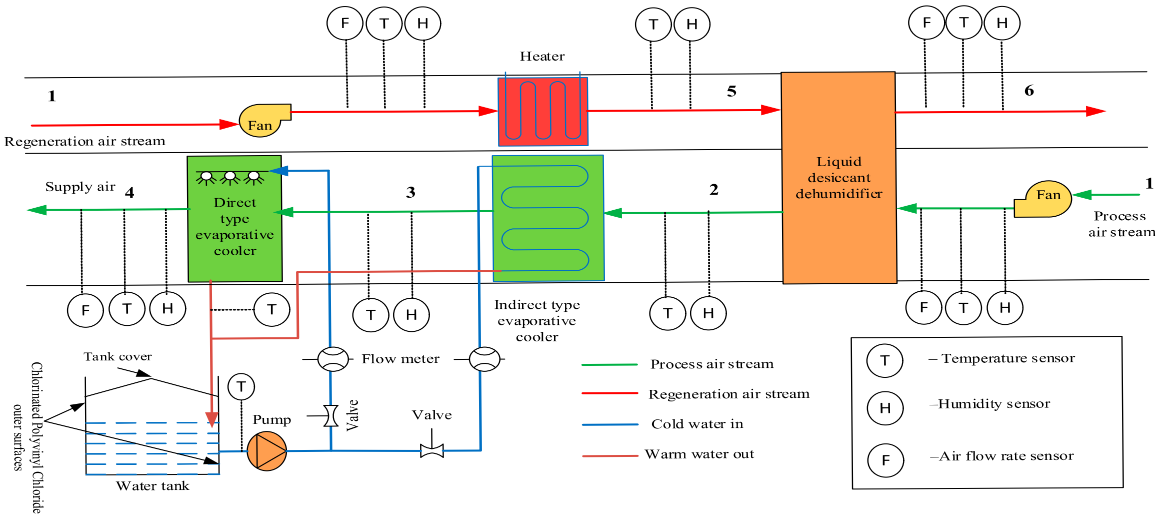
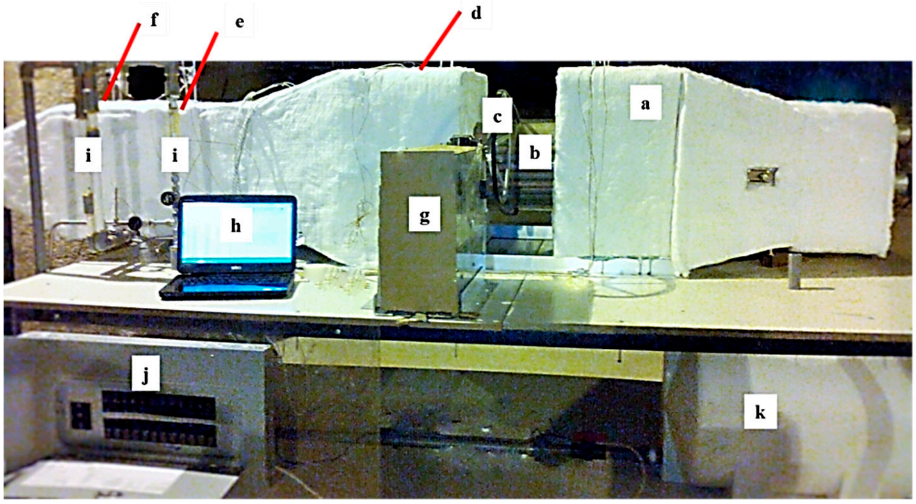


Figure 4. Schematic of the experimental setup.



(a)- Ducting with insulation, (b)- Desiccant dehumidifier (wheel), (c)- Pulley and belt, (d)- Heater (inside), (e)- Indirect evaporative cooler (inside), (f)- Direct evaporative cooler (inside), (g)- Variable speed motor, (h)- Data acquisition system, (i)- Flow meter, (j)- Electrical panel, (k)- Water tank

Figure 5. Photographic view of the experimental setup.

A one-fourth section of the wheel is dedicated for regeneration air stream and 3/4 for the process air stream. The process air is dehumidified after passing through the dehumidification section of the wheel where the heated air (55 to 85 °C) removes the moisture absorbed by the wheel. For better performance of the desiccant cooling system and reliable results, efforts are made to minimize any leakage. The desiccant dehumidifier is located in the middle of two aluminum frames with gaskets on both sides of the desiccant wheel. The dehumidifier rotates with minimum friction because of the smooth surface provided by the glass fiber gaskets. High temperature insulation is provided between the hot and cold side of air to minimize the transfer of heat through the walls.

5. Measuring Instruments

Temperatures of the process and regeneration air streams at various state points are measured using k-type thermocouples, as shown in Figures 4 and 5. Digital hygrometer and anemometer are used to measure the humidity and the velocity of the air, respectively at different points. The details of all the sensors are summarized in Table 1. All the sensors are calibrated under testing conditions.

Table 1. Specifications of sensors.

Parameters	Devices	Model No.	Accuracy
Temperature	K-type thermocouple	KK-K-30	±0.2 °C @23 ± 5 °C
Relative humidity	Hygrometer	RHXL3SD	For ≥ 70% RH ±(3% + 1% RH) For <70% RH ± 3% RH
Flow rate	Anemometer	HHF1001A	1.5% Full scale accuracy
Heater capacity	Wattmeter	GH-019D	±(0.2% reading +0.05% FS)
Density	Hydrometer	-	±1 kg/m^3

6. Test Procedure and Conditions

Table 2 summarizes the ranges of the different operating parameters. The initial temperature and humidity values were set by turning on the test chamber. After that, both process and regeneration air fans and variable speed motor for desiccant dehumidifier were turned on. After certain time, the regeneration temperature was achieved using electric heater. Once the steady state conditions were achieved, multiple measurements were taken with the instruments listed in Table 1. The data acquisition (DAQ) was established with LabVIEW 2013, which has the ability to simultaneously control and record the temperature, humidity, and mass flow rates at each state point.

Table 2. Operating parameters.

Parameter	Value Range	Base Value
Ambient air temperature (°C)	25–40	35
Ambient air humidity ratio (kg_v/kg_a)	0.010–0.025	0.020
Process air mass flux ($kg/s{\cdot}m^2$)	1.5–8.0	3.0
Regeneration air flow rate ($kg/s{\cdot}m^2$)	1.5–4.5	1.0
Regeneration temperature (°C)	55–85	70

7. Uncertainty Analysis

No physical quantity can be measured with perfect certainty. There are always errors in any measurement and inaccuracies can and do happen. Measurements should be made with great care to reduce the possibility of error as much as possible. The general error propagation (u) is given by Equation (1). Here u is a function of x, y, and z. Δu is the uncertainty of u, similar to Δx, Δy, and Δz:

$$\Delta u = \sqrt{\left(\frac{\partial u}{\partial x}\right)^2 \times (\Delta x)^2 + \left(\frac{\partial u}{\partial y}\right)^2 \times (\Delta y)^2 + \left(\frac{\partial u}{\partial z}\right)^2 \times (\Delta z)^2} \tag{1}$$

The combined uncertainty (U) is determined by the calibration of all sensors and is calculated using Equation (2):

$$U = \sqrt{(U_{\text{instrument}})^2 + (U_{\text{caliberation}})^2 + (\text{U}_{\text{random}})^2} \tag{2}$$

where, U_{random}, is calculate with Equation (3):

$$U_{\text{random}} = \frac{S_{\mu,95\%}\vartheta}{\sqrt{N_D}} \tag{3}$$

$$\mu = N - 1 \tag{4}$$

For all the thermocouples used in this experimental setup, an uncertainty of 0.10–0.25 K was observed. The calculated uncertainties for different instrumentations are listed in Table 3.

Table 3. Uncertainties of measuring instruments used in experiment.

Instrument	Parameter	Uncertainty
K-type thermocouple	Temperature	± 0.2 °C
Anemometer	Velocity of air	± 0.1 m/s
Density meter (hydrometer)	Desiccant density	± 1 kg/m^3
Electronic weighing machine	Desiccant mass	± 0.001 kg
Rotameter	Water flow rate	± 0.5 L/min

8. Performance Indicators

A number of parameters are used in order to evaluate the performance of desiccant based cooling system. This section describes the performance indices used in the present system. These parameters are used for performance of a liquid desiccant enhanced cooling system to have lower regeneration temperature, no carryover of desiccant solution, and better supply air conditions. The relationships for dehumidification coefficient of performance (DCOP), cooling capacity, coefficient of performance (COP), electric coefficient of performance (ECOP), thermal coefficient of performance (TCOP) and sensible energy ratio (SER) are presented in Equations (5)–(10), respectively [19–21]:

$$\text{DCOP} = \frac{\dot{m}_p \times h_{fg} \times (\omega_1 - \omega_2)}{\dot{m}_r \times (h_7 - h_6)} \tag{5}$$

$$\text{CC} = \dot{m}_p(h_1 - h_4) \tag{6}$$

$$\text{COP} = \frac{\text{CC}}{E_{\text{thermal}} + \frac{(E_{\text{cool}} + E_{\text{pumping}})}{\beta}} \tag{7}$$

where, the value of equivalent conversion coefficient (β) is taken as 0.3 for the present study [12]:

$$\text{ECOP} = \frac{\text{CC}}{E_{cool} + E_{pumping}} \tag{8}$$

$$\text{TCOP} = \frac{\text{CC}}{E_{\text{thermal}}} \tag{9}$$

The additional cooling required along with the dehumidification system in order to achieve comfort conditions in a conditioned room is defined by sensible energy ratio (SER) [21]:

$$\text{SER} = \frac{\dot{m}_p \times (h_2 - h_1)}{\dot{m}_r \times (h_7 - h_6)} \tag{10}$$

For better performance of the system, the achieved values of DCOP, CC, COP, ECOP, and TCOP should be higher whereas value of SER should be lower. The lower value of SER means less cooling is required after the desiccant dehumidifier.

9. Results and Discussion

The performance of the desiccant cooling system is tested to have lower regeneration temperature and better supply air conditions. Experiments were carried out using the ranges of operating parameters listed in Table 3. Generally, high values of the determination coefficient or R-squared (R^2) were obtained in all the cases with a linear fitted regression line. The high value of R^2 indicates that data is uniformly distributed along the regssion line.

The effect of the process air inlet humidity on system's performance was investigated and the results are shown in Figure 6. The electrical, thermal, and overall performance increased with the increase in inlet air humidity ratio. The mass transfer potential enhanced with increased humidity ratio of ambient air which in turn improved the capacity and performance of the system. In fact, the driving force for mass transfer is increased with the rise of humidity ratio of inlet air. Although, with the increase in ambient air humidity the required input heat for desorption of moisture also increased. However, this increase is not as high as capacity of the system to remove latent load. Thus, the higher the ambient air humidity, the better the performance (COP, TCOP, and ECOP, DCOP) of the system will be.

The COP, TCOP, ECOP, and DCOP represent the overall, thermal, electrical and dehumidification coefficient of performance, respectively. Similar to the moisture removal rate, a higher DCOP was achieved at higher inlet humidity ratio due to an increase in the moisture absorption capacity of desiccant dehumidifier as shown in Figure 6. For instance, the DCOP increased by 62% when the humidity ratio is changed from 0.01 to 0.025 kg/kg. With regard to sensible energy ratio, the temperature at the exit of dehumidifier strongly increased with humidity ratio of inlet air because of larger quantity of water vapor absorbed by the desiccant. Therefore, temperature at state 2 increased due to the increase in the released heat of absorption. Thus, increase in humidity ratio of the inlet air increases the sensible energy ratio of the system at fixed regeneration value and supply air temperature.

Figure 7 illustrates the dependence of different performance parameters on air temperature. As regards to DCOP, an increase in the process air inlet temperature causes a slight reduction of moisture removal capacity but an increase in enthalpy of inlet air increases DCOP. In case of SER, the effect of inlet air temperature is insignificant due to change in temperature of inlet air because of small variations in absorption heat and air temperature at the exit of the dehumidifier.

An increase in the mass flux of the air leads to an improvement in the system performance coefficients COP, ECOP, and TCOP, (Figure 8). This is due to the fact that the cooling capacity of the system increases while the input energy is kept constant. It is to be noted that the mass flux of regeneration air is kept constant and so the required thermal load remains the same. Due to respective high and low mass flow rates, the absorbed moisture may not be efficiently removed from the dehumidifier. Similarly, at low flow rate of process air and high flow rate of regeneration air, the required heat input will increase and may cause a decrease in system performance. The moisture absorbed by the desiccant dehumidifier may increase with the process air mass flux due to enhanced driving force for moisture transfer. However, the total moisture removal capacity decreases due to less residence time at higher flow rates. Thus, DCOP decreases with mass flux of process air. Furthermore, increasing the inlet flow rate of process air leads to an increase in both the supply air humidity ratio and temperature inside the dehumidifier. Thus, keeping the regeneration temperature constant, sensible energy ratio increases with inlet process air mass flux as shown in Figure 8.

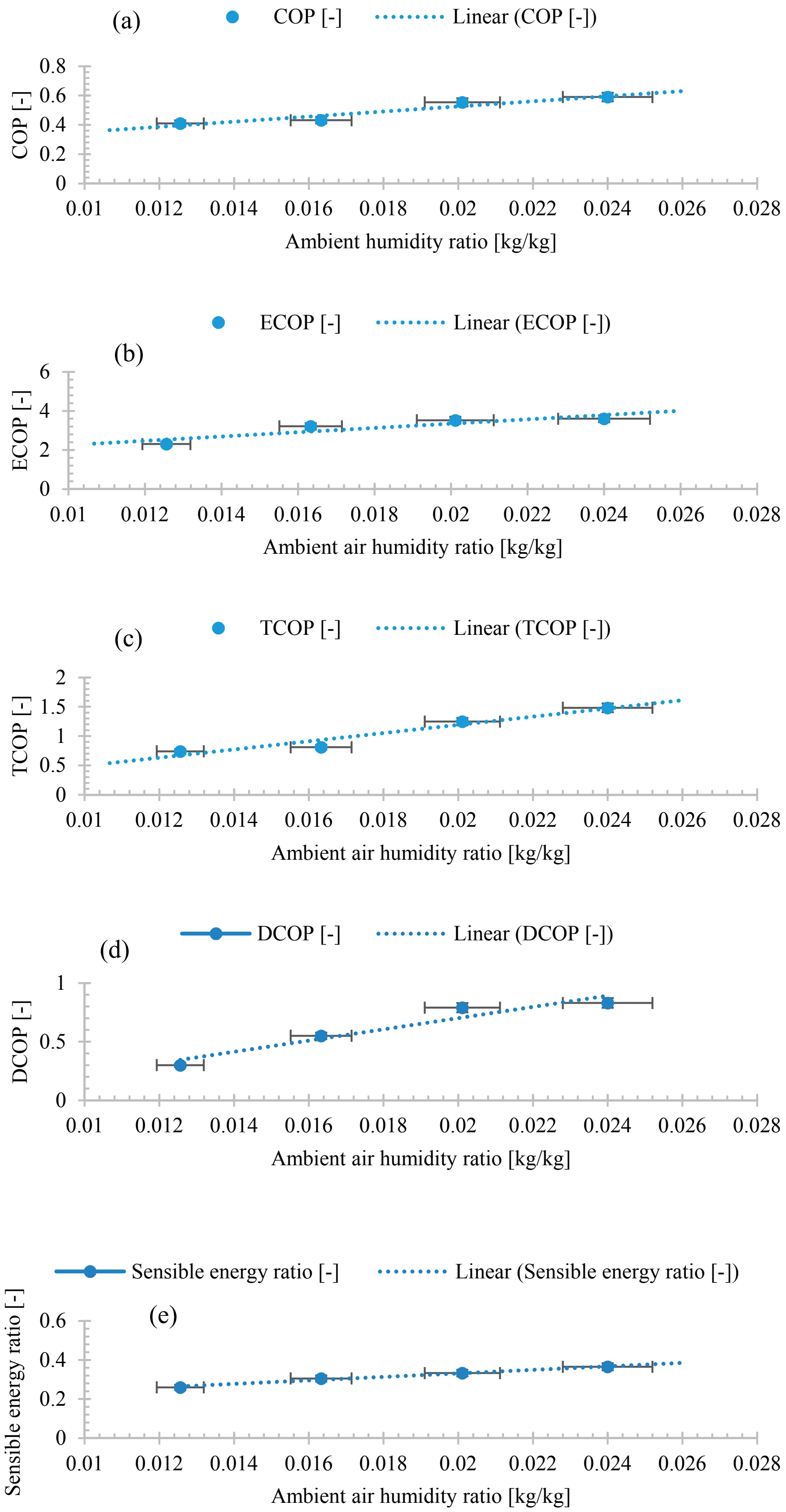


Figure 6. Variations of (**a**) COP; (**b**) ECOP; (**c**) TCOP; (**d**) DCOP; (**e**) SER with ambient air humidity ratio.

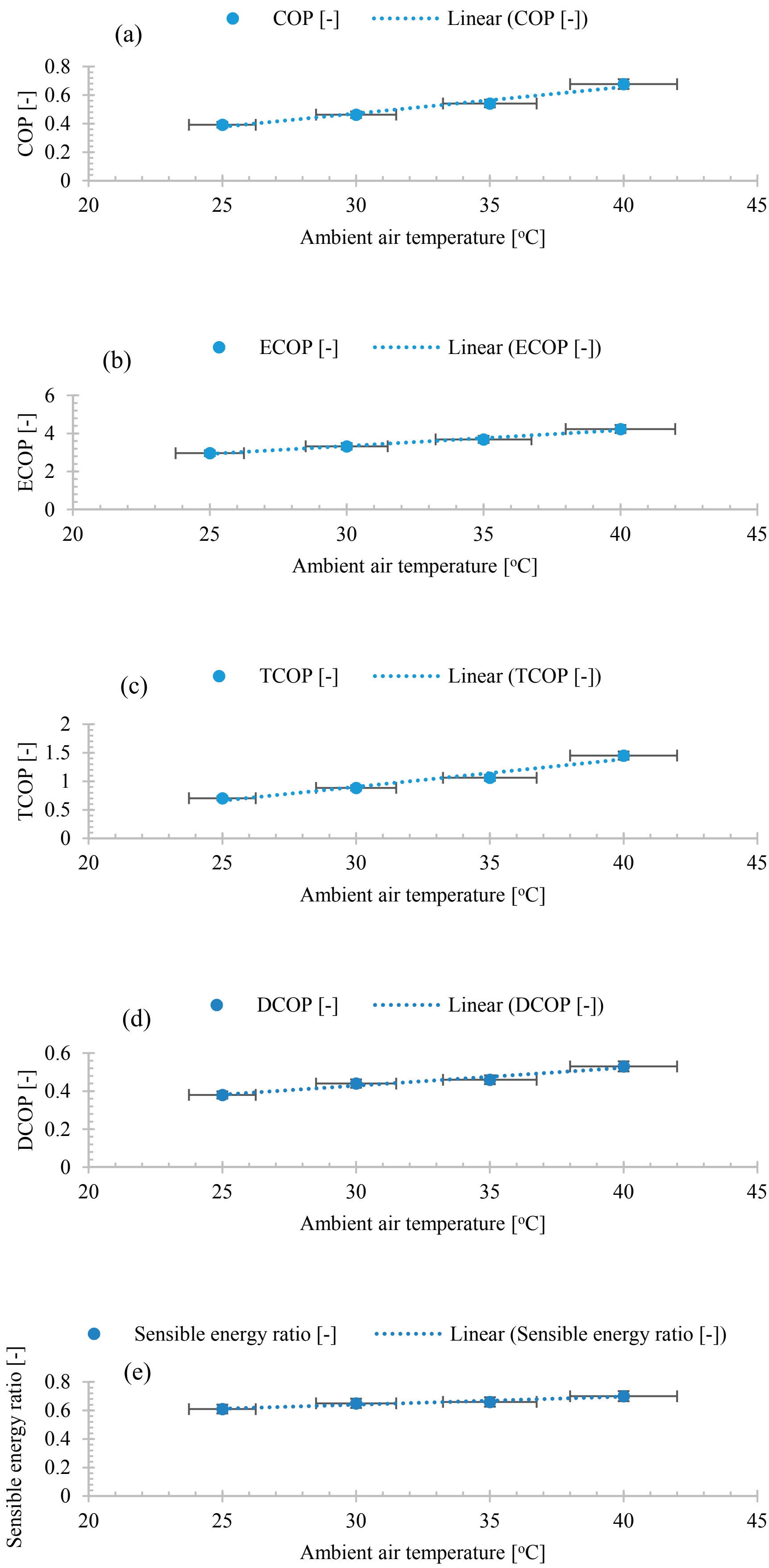


Figure 7. Variations of (**a**) COP; (**b**) ECOP; (**c**) TCOP; (**d**) DCOP; (**e**) SER with ambient temperature.

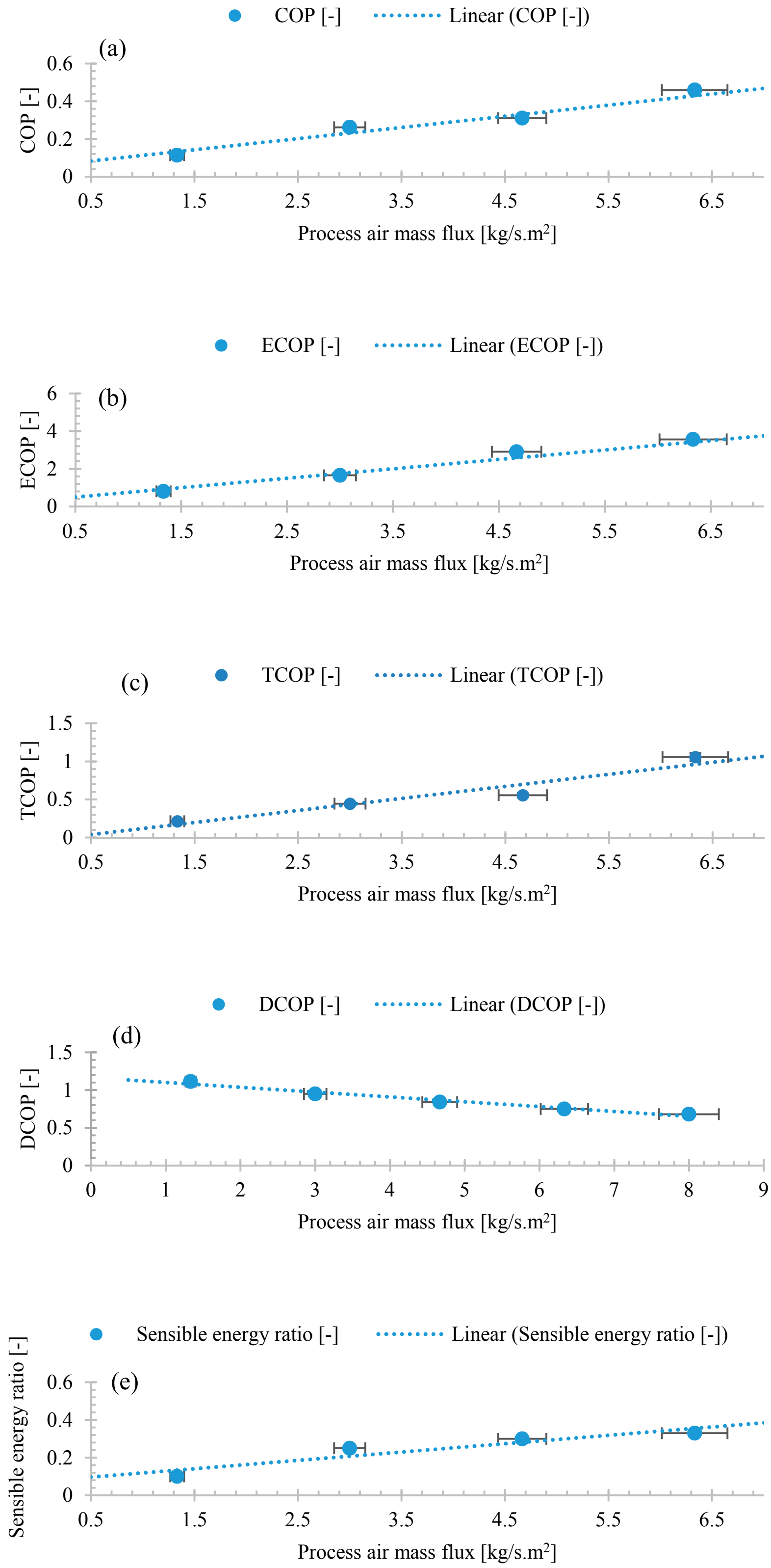


Figure 8. Variations of (**a**) COP; (**b**) ECOP; (**c**) TCOP; (**d**) DCOP; (**e**) SER with process air mass flux.

The effect of regeneration air flow rate on performance of the system is presented in Figure 9. The COP and TCOP decreased by 25 and 60%, respectively with the regeneration air flow rate changed from 1.5 to 4.5 $kg/m^2 \cdot s$. An increase in regeneration flow rate decreases DCOP, as shown in Figure 9.

This is due to a proportional rise in the energy requirement for regeneration. With regards to SER, an increase in regeneration air flow rate resulted in the rise of absorbed moisture and heat of absorption. The regeneration temperature inversely affects the overall COP, TCOP, and DCOP of the system as it can be observed from Figure 10. The variation of regeneration temperature from 55 to 85 °C, the values of COP, TCOP, and DCOP of the system decreased from 0.91 to 0.58, 3.4 to 1, and 0.6 to 0.4, respectively. The sensible energy ratio is decreased with the increase in regeneration temperature (Figure 10). This decrease of SER is due to the increase in the difference between the regeneration and the process air inlet temperatures. As described above, the increase in ratio of flow rates causes a decrease in DCOP for desiccant dehumidifier due to an increase in regeneration heat. Also, the increase in this ratio will cause an increase in temperature at the exit of the dehumidifier due to increase in released heat of absorption. This increase in exit temperature in turn decreases the SER. Thus, with the increase in ratio of air flow rates and regeneration temperature, a reduction in SER occurs.

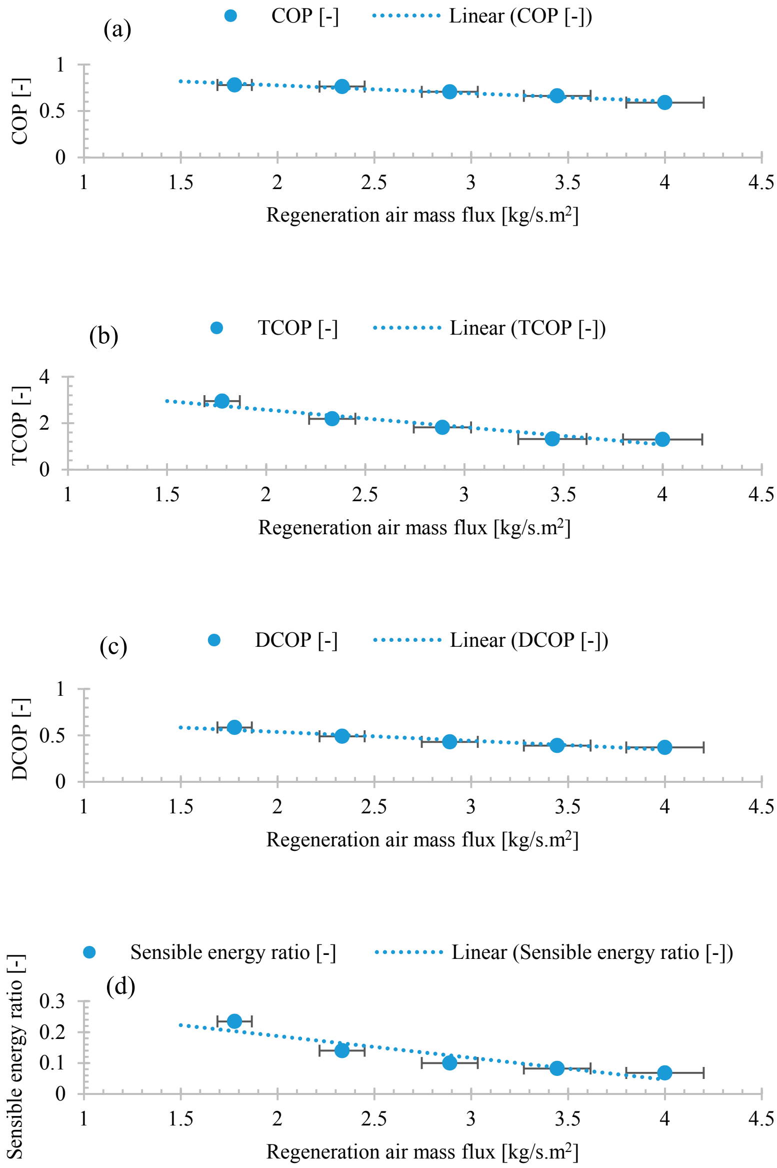


Figure 9. Variations of (**a**) COP; (**b**) TCOP; (**c**) DCOP; (**d**) SER with regeneration air mass flux.

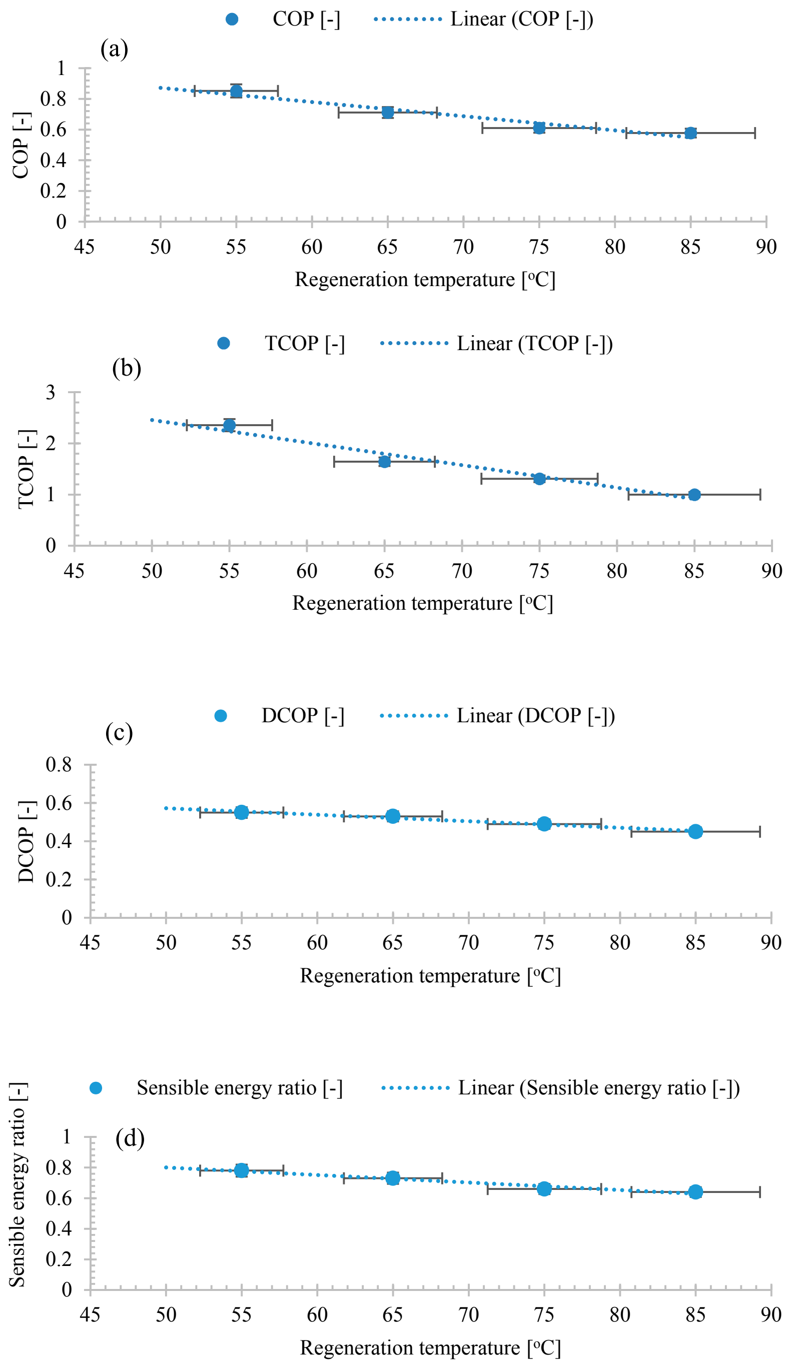


Figure 10. Variations of (**a**) COP; (**b**) TCOP; (**c**) DCOP; (**d**) SER with regeneration temperature.

The developed system showed better performance compared to the results reported in the literature using other configurations of this technology. The results obtained at a regeneration temperature of 55 °C for the present system are; COP = 0.8, TCOP = 2.4 whereas the results obtained by Abdel-Salam et al. [20] for the membrane liquid desiccant dehumidifier under similar operating conditions were: COP = 0.63 and TCOP = 1.48. Furthermore, the value of COP obtained by Bourdoukan et al. [22] for the effect of regeneration temperature (55 °C), ambient temperature (35 °C), and inlet humidity ratio (0.014 kg/kg) were reported as 0.42, 0.29, and 0.35, respectively, whereas, under similar operating conditions, the present system give COP values of 0.8, 0.5, and 0.44 for the effect of regeneration temperature, ambient temperature and inlet humidity ratio, respectively. In comparison to a solid desiccant rotary cooling system, the achieved performance of the proposed system is much better. The dehumidification coefficient of performance achieved by Ge et al. [19] for a solid desiccant cooling system at a regeneration temperature of 60 °C was 0.38 whereas in the present case it was 0.59. The performance of liquid desiccant assisted cooling system is expected to be higher compared to solid desiccant cooling system due to lower regeneration heat requirement. The required regeneration temperature in case of liquid desiccant (calcium chloride) is 60–85 °C [7] whereas for solid desiccant (silica gel) it is 60–120 °C [23]. More fieldwork is required in order to compete with other technologies available in the market. A summary of potential advantages which can be achieved by the implementation of proposed system is provided in Figure 11.

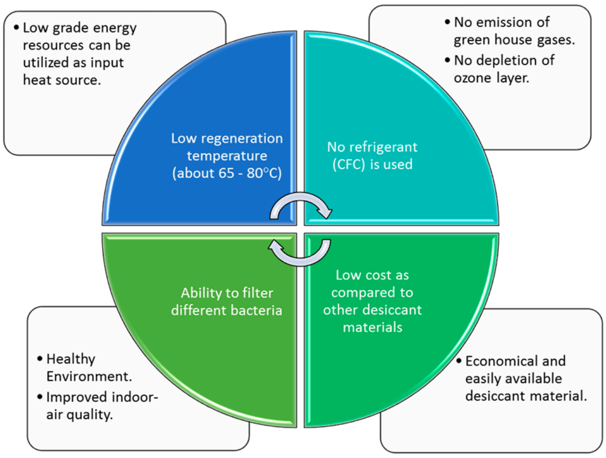


Figure 11. Potential advantages of liquid desiccant based cooling.

10. Conclusions

Solar thermal cooling is not a new concept, nevertheless, it has been gaining relevance to provide efficient cooling at low costs without generating CO_2 emissions. Throughout history, energy has been used in different forms such as mechanical, electrical, chemical, heat, etc. The use of alternative sources of energy can result in a more energy efficient and cost effective systems. Many efforts have been made to make the effective use of renewable energy sources due to escalating oil prices and the cost of other primary energy resources in recent years.

In this paper, the performance of a desiccant enhanced evaporative cooling system is investigated experimentally. The effects of different performance parameters on the system performance have been studied. With the increase in regeneration temperature, DCOP decreased due to increase in input energy at higher regeneration temperatures. It was concluded that the efficient control of input and output parameters can provide effective dehumidification capacity with the tested system. The total moisture removal capacity decreases at high process air flow rates due to the reduced residence time. The electrical, thermal, and overall performance increase with the increase in inlet air humidity ratio. The values of COP, ECOP, and TCOP increased from 0.41 to 0.59, 2.30 to 3.6, and 0.74 to 1.48, respectively, when the ambient air humidity ratio was changed from 0.01 to 0.025 kg/kg. A higher DCOP was achieved at higher inlet humidity ratio due to an increase in the moisture absorption capacity of the desiccant dehumidifier. For an increase in humidity ratio from 0.01 to 0.025 kg/kg the DCOP increased by 62%. The variations of regeneration temperature greatly affect the values of COP and TCOP. The increase in regeneration temperature from 55 to 85 °C lowered the COP and TCOP of the system by 32% and 53%, respectively. The desiccant-based technology is beneficial from both an economic as well as an environmental point of view. The system could be improved by developing composite desiccant materials and further lowering the required regeneration temperature. In this way better performance of the system can be achieved.

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Nomenclature

COP	coefficient of performance
C_p	specific heat (kJ/kg·K)
CC	cooling capacity (kW)
E	input energy (kW)
h	specific enthalpy (kJ/kg)
h_{fg}	latent heat of vaporization (kJ/kg)
$\dot{m}$	air mass flow rate (kg/s)
N_D	number of data points
S	student test at a 95% confident interval
T	temperature (K)
U	uncertainty error
Greek letters	
ω	solute humidity (kg/kg)
ϑ	standard deviation
Subscripts	
1, 2, 3 . . .	state points
a	air
p	process
r	regeneration
μ	degree of freedom

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