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Experimental investigation of a novel configuration of desiccant based evaporative air conditioning system

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ABSTRACT

A novel configuration of desiccant based evaporative cooling system for air conditioning application is developed and tested. At the beginning of the design stage of the system, an analysis is carried out in order to maximize the performance of the system. It is found based on configuration that outdoor air must be used for regeneration to increase performance of the system and so three air channels are used. Experiments are carried out to investigate the total performance of the system and performance of the components used during summer season in a hot and humid climate. Effectiveness values for both heat exchangers and evaporative coolers are calculated through this work. In addition to the cooling capacity, coefficient of performance (COP) and energy consumption of the system are also evaluated. Results show that the effectiveness for the heat exchangers and evaporative coolers are very high under different outdoor conditions. It is also shown from the results that indoor air conditions are in the range of thermal comfort zone defined by ASHRAE and expanded comfort zone for evaporative air conditioning applications.

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1. Introduction

Desiccant dehumidification assisted air conditioning systems are increasingly applied in commercial and institutional buildings, such as supermarkets, schools, ice arenas, cold warehouses, hotels, theaters, and hospitals [1]. They can be operated in recirculation mode or in ventilation mode. In recirculation cycle, the process air is the return air from the space being conditioned and the regeneration air is the outdoor air. In the ventilation mode, the process air is the outdoor air and the regeneration air can either be the outdoor air or the conditioned space exhaust air [2]. A desiccant based air conditioning system is a hybrid system of desiccant dehumidification, evaporative cooling and regeneration process to cool and dehumidify the space air and maintain it at a required temperature and relative humidity with adequate outdoor ventilation air. It improves at the same time the efficiency of energy use [3]. A number of experimental investigations are reported in the literature regarding of the hybrid desiccant air conditioning system [4,5]. In many studies, solid desiccant based evaporative cooling cycles have been reported. One of the earliest cycles and probably the most commonly reported was proposed by Penington [6].

One of the most commonly cited advantage of desiccant based air conditioning system is the operation by thermal energy sources such as solar energy. Bourdoukan et al. [7] studied experimentally a desiccant air handling unit powered by vacuum tube solar collectors. In the study, the components under various operating conditions were analyzed and overall performance of the installation over a day for a moderately humid climate with regeneration by solar energy was evaluated.

Khalid and Dhaidan [8] evaluated the performance of solar assisted heating and desiccant cooling system for a domestic two story residence located in Baghdad. Desiccant air conditioning systems are not only energy efficient and environment friendly, but also cost-competitive, especially for hot dry and hot humid areas. A conventional air conditioning system often needs dehumidification and re-heating, which require more energy and higher initial investment cost.

Evaporative cooling has been used for thousands of years in various forms for comfort cooling and is still in common use around the world because of its simplicity, low cost and effectiveness. In evaporative cooling systems, air is drawn through evaporative cooler and its sensible heat energy evaporates water; the heat and mass transfer between the air and water decreases the air dry bulb temperature and increases the humidity at a constant wet-bulb temperature. Evaporative cooling technologies were either utilized as direct, indirect or direct/indirect. Khalid and Mehdi [9] conducted the application of the indirect evaporative

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Nomenclature

C	heat capacity (kW/K)
c_p	specific heat (kJ/kg K)
COP_T	total coefficient of performance
COP_W	total work coefficient of performance
COP_H	thermal coefficient of performance
$\dot{E}_{tot}$	total energy consumption (kW)
$\dot{m}$	mass flow rate (kg/s)
$\dot{Q}_{cc}$	cooling capacity (kW)
$\dot{Q}_{cl}$	cooling load (kW)
$\dot{Q}_{reg}$	regeneration heat input to the desiccant wheel (kW)
$\dot{Q}$	amount of actual heat transfer (kW)
$\dot{Q}_{max}$	maximum possible heat transfer (kW)
T	temperature (°C)
W	absolute humidity (kg/kg)
ΔW	absolute humidity difference (kg/kg)

Greek letters

η	effectiveness (%)

Subscripts

1, 2,…,17	state numbers
c	cold
EC	direct evaporative cooler
calib	calibration
f	fresh air
daq	data acquisition
h	hot air
HE	heat exchanger
i	inlet air
min	minimum
o	outlet
oth	other
r	regeneration air
wb	wet bulb

cooling in fulfillment of the variable cooling load of a typical Iraqi dwelling and they show that indirect evaporative cooling provides comfortable indoor condition. Heidarinejad et al. [10] experimentally investigated cooling performance of two stage indirect/direct evaporative cooling system in the various simulated climatic conditions. Hindoliya and Mullick [11] analyzed hourly ambient temperature and humidity of 60 major cities of India to assess the utilization potential of direct evaporative cooling for comfort conditioning.

The basic idea of desiccant air conditioning is to integrate the technologies of desiccant dehumidification and evaporative cooling together. Bourdoukan et al. [12] investigated the effect of outside conditions and effectiveness of the system components on the performance of two configurations of desiccant systems through simulation. In their study, a model of a desiccant air handling unit was presented and validated experimentally.

La et al. [13] analyzed and compared a desiccant cooling system using regenerative evaporative cooling and a one-rotor two-stage desiccant cooling system. They found that the system with regenerative evaporative cooling can handle air to much lower temperature while maintaining good thermal performance. Panaras et al. [14] presented a theoretical model for the operation of a desiccant air conditioning system, developed on the basis of existing approaches for the modeling of the main subsystems of such a system. Kanoğlu et al. [15] reported an application of an open desiccant cooling process with ventilation and recirculation modes of the system operation. Kodama et al. [16] performed measurements on a solid desiccant cooling unit operated in the ventilation mode and established the entropy balance of the unit utilizing experimental data. They showed that the sum of all the considered entropy productions completely explain the difference between the Carnot COP and the actual COP of the unit.

Hürdoğan et al. [17] investigated experimentally a novel desiccant based air conditioning system to improve the indoor air quality and reduce energy consumption. In the system studied, the moisture of the fresh air was reduced passing it through a solid desiccant wheel and then its temperature decreased by the "dry coil" of a vapor compression cycle. They showed that a heat exchanger for pre-heating the regeneration air with exhaust air was feasible to install. It was also reported that although COP changes between 0.4 and 4 according to the electric heaters switching on or off, the daily mean value was 1.35.

Desiccant air conditioning technologies have many advantages that include the following: (i) very small electrical energy is consumed and the sources for the regenerating thermal energy can be diverse (i.e. solar energy, waste heat, natural gas); (ii) a desiccant system is likely to eliminate or reduce the use of ozone depleting CFCs (depending on whether desiccant cooling is used in conjunction with evaporative coolers or vapor compression systems, respectively); (iii) control of humidity can be achieved better than those cases employing vapor compression systems since sensible and latent cooling occur separately. Also, desiccant systems have the capability of removing airborne pollutants [18].

Based on literature survey, it is seen that waste air from conditioned space is generally used for regeneration in desiccant based evaporative air conditioning systems. The desiccant based air conditioning design needs new configuration for optimized operation of the system. In this study, at the beginning of the design stage of the system, an analysis is carried out in order to maximize the performance of the system. It is found that outdoor air must be used for regeneration air to increase the regeneration heat and the capacity of desiccant wheel due to lower inlet absolute humidity and higher ambient air temperature. In this study, a new configuration of desiccant based evaporative air conditioning system is constructed and tested in Çukurova University, Adana, Turkey. The system has a novel design in terms of both air channels and heat exchangers used. Regeneration air is taken from outdoor and a rotary regenerative type heat exchanger, which is not common to this type of systems, is used for pre-heating the regeneration air with exhaust air. Even the places of the ventilators are carefully selected. Detailed description of the system is explained in the followings. In this study, total performance of the system designed and performance of the components used are investigated during summer season in a hot and humid climate.

2. Experimental setup

2.1. Description of the system

A desiccant based evaporative air conditioning system, which includes dehumidification, evaporative cooling, heat and cool recovery, are used for air conditioning of a space. Fig. 1 shows a schematic view of the desiccant based evaporative air conditioning

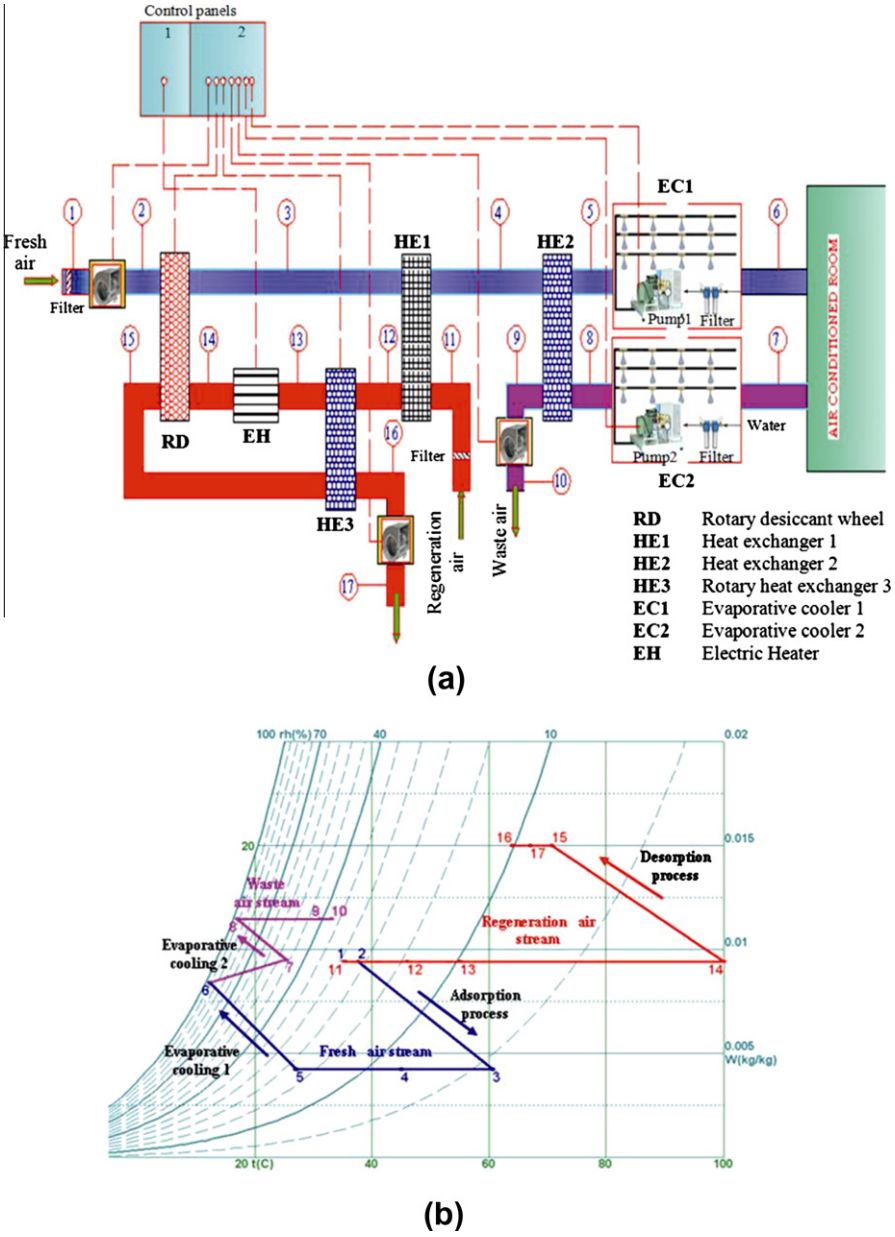


Fig. 1. (a) Schematic view of the desiccant based evaporative air conditioning system and (b) approximately representation all processes on psychrometric chart.

system and all processes on a psychrometric chart. Experimental setup consists mainly of one rotary desiccant dehumidifier, three heat exchangers, two direct evaporative coolers, electric heater unit, fans (used for fresh, regeneration and waste air), pumps (used for evaporative coolers), filters, control units and channels. The desiccant based evaporative air conditioning system presented in Fig. 1 is different from previous designs such as of Pons and Kodama [19], Camargo et al. [20], Enteria et al. [21] and La et al. [22].

The system, as can be seen from Fig. 1, has three main air channels. The first channel (fresh air channel) is used to take fresh air from outdoor and blow it into the conditioned room. In this study, the supply air is 100% fresh air, which improves the indoor air quality. The waste air is discharged from the room by a fan through the second channel (waste air channel). The advantage of this channel is that waste air which has lower temperature is cooled again by an evaporative cooler and thus, the fresh air is pre-cooled by it in heat exchanger without increasing fresh air moisture content.

Third channel (regeneration air channel) is used for regeneration air stream. The advantage of this channel is that the absolute humidity of the ambient air is lower than that of outlet waste air and ambient air temperature is generally higher than that of outlet waste air. Thus, the using of ambient air is generally more beneficial than that of the waste air. Besides, after rotary desiccant wheel, temperature of exhaust regeneration air can be higher than that air at of inlet of electric heater unit. Therefore, we utilize exhaust regeneration air to reheat regeneration air by using a rotary heat exchanger. The air flow rates are controlled by a control unit. Filters are placed inlets of the fresh and regeneration air channels.

The rotary desiccant wheel (RD) is followed by heat exchanger-1 (HE1) and heat exchanger-2 (HE2) to pre-cool the fresh air in the fresh air channel. After HE2, evaporative cooler-1 (EC1) is utilized to cool the fresh air to the blowing temperature.

The air sucked from the outdoor into the fresh air channel by a fan (process 1–2) enters to RD in which moisture is absorbed by the desiccant materials of the wheel (process 2–3). The dry fresh

air is pre-cooled first by the regeneration air in HE1 (process 3–4) and then enters into HE2 (process 4–5) where it is further pre-cooled by the evaporatively cooled waste air. Fresh air finally enters into EC1 (process 5–6) in which it is cooled down to blowing conditions and supplied to the conditioned room (state 6).

Waste air, which is sucked from conditioned room by a fan, first passes through evaporative cooler-2 (EC2) (process 7–8) and enters HE2 (process 8–9). As the waste air passes through HE2, its temperature increases and consequently temperature of the fresh air decreases. Waste air is discharged to atmosphere after HE2.

For regeneration, air is sucked from outdoor by a fan and sent into HE1. Due to heat transfer from the fresh air in HE1, temperature of the regeneration air increases (process 11–12) and that of the fresh air decreases (process 3–4). Regeneration air is sent into heat exchanger-3 (HE3) for further pre-heating (process 12–13). Electric heaters (EH) are used for heating regeneration air before entering into the rotary dehumidifier (state 14).

The regeneration air is pre-heated first by the recuperative HE1 and then by the regenerative HE3. The heat required for further heating of the regeneration air to the desired regeneration temperature is supplied by the electric heater unit. Subsequently regeneration air passes through the desiccant wheel (process 14–15) where its moisture content increases and then it is sent into HE3 (process 15–16). Finally, regeneration air is discharged to atmosphere (state 17). The photographic view of the system is shown in Fig. 2.

2.2. Description of system components

Silica gel water adsorption refrigeration is one kind of energy saving and environmental friendly refrigeration methods. Therefore, it has attracted more attentions from researchers in the world. Many theoretical [23] and technical improvements [24] about silica gel water adsorption refrigeration usually used in open cycle air conditioning system have been performed. Desiccant wheel is the one of most important component of the desiccant cooling system. Physical properties of the desiccant wheel used in this study are shown in Table 1. Shell and desiccant wheel have connected by rubber pad to minimize the leakage. A driving subsystem is used to rotate the desiccant wheel. It consists of an electromotor, a wheel disk and a belt. The wheel disk and belt are used to transfer power from electromotor to desiccant wheel.

Direct type evaporative air cooling units are used in the experimental setup. In direct evaporative cooling units, very small water droplets obtained by using high pressure pumps are sprayed to the air in opposite direction. A pump that has ceramic pistons is used to pressurize the water into working pressure of the nozzles. The properties of spray nozzles and pumps can be seen in Table 2.



Fig. 2. Photographic view of the experimental system.

Table 1
The physical properties of desiccant wheel.

Rotor diameter	965 mm
Rotor depth	200 mm
Rotor air flow	50% process air–50% regeneration air
Adsorbent	Silica gel
Rotary speed	12 rph
Pressure drop	290 Pa

Table 2
The properties of spray nozzles and pumps.

Hole diameter	0.2 mm
Spraying angle	70°
Droplet diameter	8 μm
Working pressure of pumps	30–100 bar
Flow rate of pumps	1 l min^{-1}

Table 3
The properties of heat exchangers.

HE1 and HE2	
Dimensions	600 × 600 × 600 mm
Plate material	Aluminum
Plate spacing	7 mm
Pressure drop	250 Pa
HE3	
Rotor material	Aluminum
Rotor diameter	965 mm
Rotor depth	200 mm
Rotary speed	12 rpm
Pressure drop	260 Pa

Separators are placed at the exit of the evaporative cooler to prevent the transport of the water droplets.

The regeneration air is finally heated by electric heaters that are used for two reasons: to automatically regulate the regeneration air temperature with a great accuracy and easy to use. Two type heat exchangers are used in this system. Cross-flow plate-fin-type heat exchangers (HE1 and HE2) and rotary regenerator heat exchanger (HE3) are used. The properties of these heat exchangers are given in Table 3.

3. Measurements and controlling

Experiments are carried out by measuring instantly temperature, pressure, relative humidity, electric current and electrical potential difference on the system. Flow rates of the fresh, waste and regeneration air streams at states 3, 7 and 15 are measured using averaging (blade type) pitot tubes and differential pressure transmitters with an accuracy of ±1%. Dry bulb temperature of the air at states 1–17 are measured using K-type thermocouples with an accuracy of 0.1 °C. Measurement of temperature in the channels is carried out at five points at a cross section (at each state). Temperature measurements made at 85 points to increasing the accuracy of the measurements.

Relative humidity of the air at states 1, 3, 4, 5, 6, 7, 8, 11 and 15 are measured using relative humidity transmitters with an accuracy of ±2%. Electric current and electrical potential differences are measured to determine the power consumption of each electrical component used. Monitoring modules are used to measure the electric current and electrical potential difference with an accuracy of ±0.5% and ±1.5%, respectively. Electrical signal outputs of the thermocouples, transmitters and modules are monitored and recorded with a computer controlled data acquisition system.

Two control panels shown in Fig. 3 are designed to control and operate of the system. The system have also two type automatic

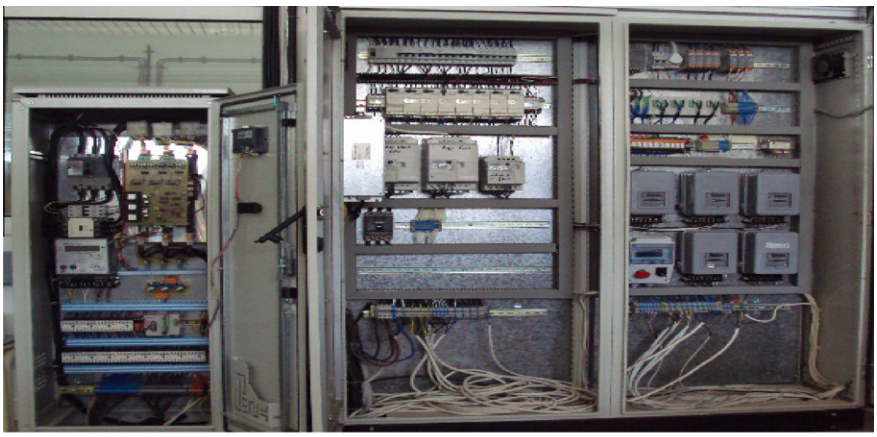


Fig. 3. Control panels used for the operation of the electric heaters (a) and all other components in the system (b).

controller; one of them is programmable logic controller (PLC) and another is proportional integral derivative controller (PID). PLC is used to control the operation of the system. Its functions include: (1) controlling the system operation, (2) controlling the fans operation by frequency inverters according to set values, (3) protecting of the desiccant wheel. If either the regeneration and fresh air fan or the desiccant wheel does not work, the control system stops heating process automatically.

Electric heater unit used to heat the regeneration air is controlled by PID (proportional (P), integral (I), and derivative (D)) automatic controller. This automatic controller is used for regeneration temperature stability in experimental studies. The regeneration air is heated to the desired temperature and enters into the rotary dehumidifier at constant temperature. When the regeneration air reaches the regeneration temperature, the voltage control unit continuously provides desired regeneration temperature with ±0.1 °C accuracy by regulating the voltage automatically.

4. Performance definitions

The following calculations are performed to determine total system performance and performances of the components used:

The effectiveness of the heat exchangers (η_{HE}) is determined by:

$$\eta_{HE} = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{1}$$

where $\dot{Q}$ is amount of actual heat transfer and $\dot{Q}_{max}$ is amount of the maximum possible heat transfer that are calculated respectively by:

$$\dot{Q} = \dot{m} c_p (T_i - T_o) \tag{2}$$

$$\dot{Q}_{max} = C_{min}(T_{hi} - T_{ci}) \tag{3}$$

C_{min} is the minimum of the heat capacity rate of cold (C_c) and hot (C_h) air streams which can be calculated by:

$$C_c = \dot{m}_c c_{pc} \tag{4}$$

$$C_h = \dot{m}_h c_{ph} \tag{5}$$

The effectiveness of the evaporative coolers (η_{EC}) is determined by the following equation:

$$\eta_{EC} = \frac{T_i - T_o}{T_i - T_{wb,i}} \tag{6}$$

The cooling capacity of the system is expressed by [17,25]:

$$\dot{Q}_{cc} = \dot{m}_f (h_1 - h_6) \tag{7}$$

The cooling load of the room is determined by:

$$\dot{Q}_{cl} = \dot{m}_f (h_7 - h_6) \tag{8}$$

and regeneration heat is calculated by:

$$\dot{Q}_{reg} = \dot{m}_r (h_{14} - h_{13}) \tag{9}$$

Total energy consumption of the system is expressed by:

$$\dot{E}_{tot} = \dot{Q}_{reg} + \dot{W}_{fan} + \dot{W}_{oth} \tag{10}$$

$\dot{W}_{oth}$ shows energy consumption of other equipments that are pumps, desiccant wheel motor and rotary regenerator motor. The COP of the system is defined as the ratio of the cooling capacity to the total energy input to the system [17,25]:

$$COP_T = \frac{\dot{Q}_{cc}}{\dot{E}_{tot}} \tag{11}$$

To analyses deeply we can also define COP_W and COP_H for total work and heat supplied to the system as follows.

$$COP_W = \frac{\dot{Q}_{cc}}{W} \tag{12}$$

$$COP_H = \frac{\dot{Q}_{cc}}{\dot{Q}_{reg}} \tag{13}$$

From the last three equations we obtain:

$$\frac{1}{COP_T} = \frac{1}{COP_W} + \frac{1}{COP_H} \tag{14}$$

The absolute humidity differences in dehumidification process (ΔW_1) and humidification processes (ΔW_2 and ΔW_3) in the evaporative coolers are expressed by the following equations:

$$\Delta W_1 = W_1 - W_3 \tag{15}$$

$$\Delta W_2 = W_6 - W_5 \tag{16}$$

$$\Delta W_3 = W_8 - W_7 \tag{17}$$

where the subscripts correspond to the state points in the desiccant cooling system.

5. Uncertainty analysis

Uncertainty analysis is the procedure employed to assess the uncertainty in a calculated result from measured variables with known values of uncertainties. For the calculation of uncertainty, the root of the sum squares is used in this study [26] and can be expressed as:

$$w_R = \left[\left(\frac{\partial R}{\partial x_1}w_1\right)^2 + \left(\frac{\partial R}{\partial x_2}w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n}w_n\right)^2\right]^{1/2} \quad (18)$$

where the result R is a given function of the independent variables $x_1, x_2, \ldots, x_n$ and $w_1, w_2, \ldots, w_n$ are the uncertainties in the independent variables. This calculation method concerning uncertainty analysis is also used in many studies [27,28].

The total uncertainty in the measurement of the relative humidity can be calculated as follows [29]:

$$w_{rh} = (w_{sensor}^2 + w_{calib}^2 + w_{daq}^2)^{1/2} \quad (19)$$

$$w_{rh} = (1,5^2 + 2^2 + 0,1^2)^{1/2} = 2.51\% \quad (20)$$

where w_{sensor} is the uncertainty in the sensor reading, w_{calib} is the uncertainty in the calibration process, w_{daq} is the uncertainty associated with the data acquisition system. These uncertainties include sensor, data acquisition and calibration uncertainty are provided by the manufacturer. Eq. (18) is a general equation that is used to calculate the uncertainty in each result. The temperature, relative humidity, flow rate, electric current and electrical potential differences are measured with appropriate instruments explained previously. The total uncertainties of the measurements are estimated to be ±0.3 °C for the air temperatures, ±2.51% for the relative humidities, ±1.59% for power inputs to the system. The total uncertainty associated with mass flow rates, cooling capacity and COP are found to be ±2.89, ±6.90% and ±7.08%, respectively.

6. Results and discussions

Performance of a desiccant based evaporative cooling system is studied to investigate the applicability of the system designed in the city of Adana where high humidity and hot outdoor air prevail in summer months as can be seen in Table 4. This table shows monthly average ambient conditions between the years 1986–2006 [30].

In this study; fresh, waste and regeneration air streams have equal volume flow rates of 3000 m^3/h. Regeneration air at the inlet of the dehumidifier is adjusted to a fixed temperature of 110 °C during the experiments. As a typical example to the experiments carried out, the measurements taken between the hours 8:30 to 18:30 in 27 August 2010 are given in the paper. The data obtained from the study are shown in Figs. 4–14 for the parameters described above.

Variation of the dry bulb temperature measured with time at states 1–6 in the fresh air channel is given in Fig. 4. Temperature of the fresh air increases (T_1 to T_2) approximately 1.5 °C due to heat gain from the fan. It is seen that poorly placed circulating fans may increase indoor temperature and reduce comfort. Therefore in this study, the fans are placed upstream of the desiccant wheel to increase the effectiveness of the system and comfort. Fresh air flows through the RD and its temperature (T_3) increases up to 60 °C. RD removes moisture from the fresh air, which releases heat and increases temperature of the fresh air. Then the fresh air passes through HE1 and HE2 in which pre-cooling processes occur by regeneration air taken from outdoor and by waste air sucked from the conditioned room. Its temperature decreases to approximately 42 °C in HE1 (T_4) and to 27 °C in HE2 (T_5). Finally the fresh air enters into EC1, where final cooling process occurs, before supplied into the conditioned room approximately at 16 °C (T_6). The average

Table 4
Monthly average ambient conditions for Adana during cooling season.

Month	Temperature (°C)	Relative humidity (%)	Absolute humidity (kg/kg)
May	25.36	53.14	0.0107
June	28.92	56.02	0.0139
July	31.31	59.83	0.0172
August	31.78	59.10	0.0174
September	30.02	50.95	0.0135

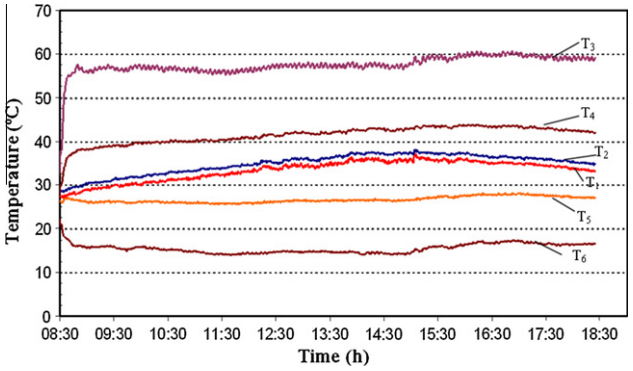


Fig. 4. Variation of dry bulb temperature with time at different states during the day for the fresh air stream.

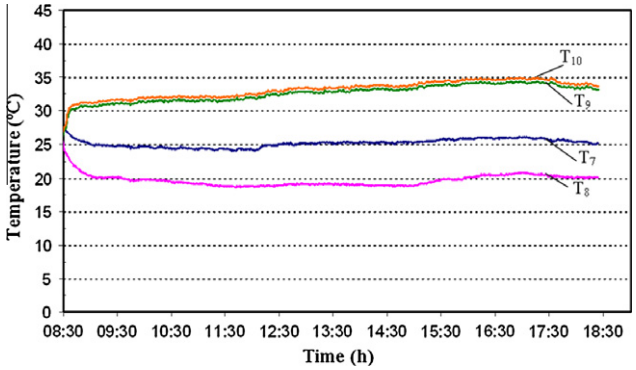


Fig. 5. Variation of dry bulb temperature with time at different states during the day for the waste air stream.

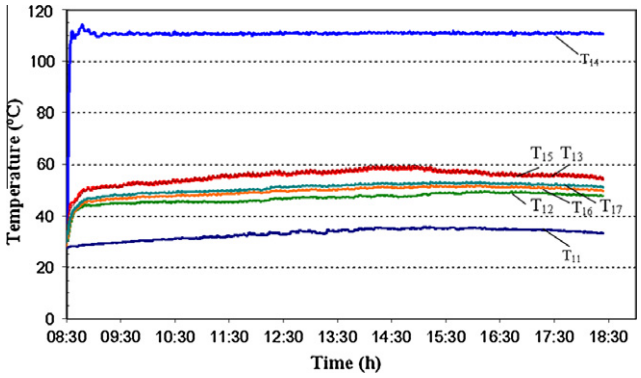


Fig. 6. Variation of dry bulb temperature with time at different states during the day for the regeneration air stream.

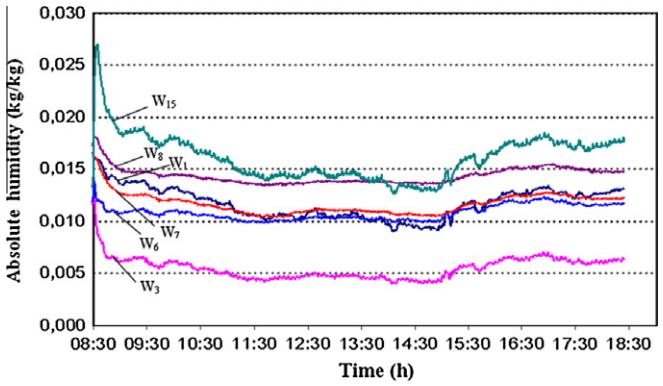


Fig. 7. Variation of absolute humidity with time at different states during the day.

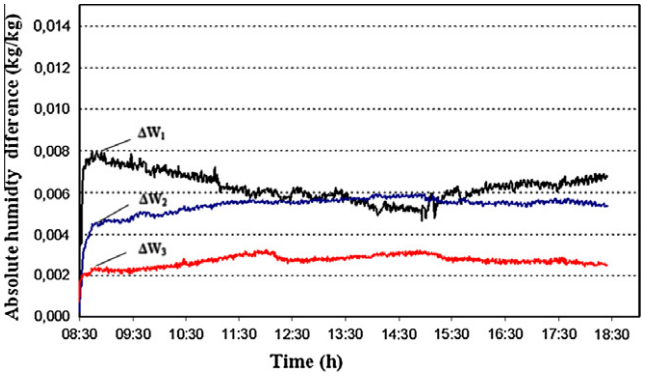


Fig. 8. Variation of absolute humidity difference with time during the day.

temperature difference between entering and leaving of EC1 reaches to approximately 11 °C. It can be seen from the figure that the air conditioning system designed is able to keep the temperature of the conditioned room around 25 °C (T_7) during the day.

The average temperature difference between entering and leaving air in EC2 is approximately 5 °C. The waste air sucked from the conditioned room at 25 °C (T_7) is cooled down to approximately 20 °C in EC2 (T_8) before entering into HE2 in order to increase the cool recovery (Fig. 5). Temperature of the waste air leaving HE2 (T_9) increases to 33 °C and discharged to outdoor at 33.5 °C (T_{10}).

Fig. 6 illustrates variation of dry bulb temperatures with time at different states during the day for the regeneration air stream. The regeneration air is heated till the desired temperature before entering into the RD. For pre-heating of the regeneration air, outdoor air at approximately 35 °C (T_{11}) first enters into HE1 and its temperature increases to 48 °C (T_{12}). The regeneration air is then directed to HE3 in which it is heated to approximately 60 °C (T_{13}) before entering to EH.

At the beginning of the experiment, regeneration temperature (temperature of the air at the inlet of the dehumidifier, T_{14}) is set to 110 °C. As can be seen from Fig. 6, T_{14} increases to the set value in a very short time and stays constant at this level during the day. The regeneration air leaves RD and then HE3 at approximately 55 °C (T_{15}) and 50.5 °C (T_{16}), respectively. Finally, it is discharged to outdoor at 52 °C (T_{17}).

Fig. 7 shows variation of absolute humidity at different states with time during the day. Fresh air and regeneration air taken from outdoor have same absolute humidity at the inlet of the system. Absolute humidity of the outdoor air (W_1) varies between 0.009 and 0.014 (kg/kg). It decreases to 0.004 (kg/kg) (W_3) after dehumidification process (state 3) and the system has minimum absolute humidity at this state. Absolute humidity of regeneration air stream at the exit of RD (W_{15}) increases up to approximately 0.018 (kg/kg) and the system has maximum absolute humidity at this state.

As seen from this figure, the absolute humidity of supply air (W_6) increases up to 0.012 kg/kg. This is due to increasing of process air absolute humidity by EC1. In the conditioned room, absolute humidity (W_7) decreases to 0.010 (kg/kg) during the experiment and its daily average value is approximately 0.012 (kg/kg). Waste air absolute humidity at state 8 (W_8) increases till 0.015 (kg/kg) due to cool recovery process in EC2.

The absolute humidity differences in dehumidification and humidification processes are depicted in Fig. 8. As seen from the

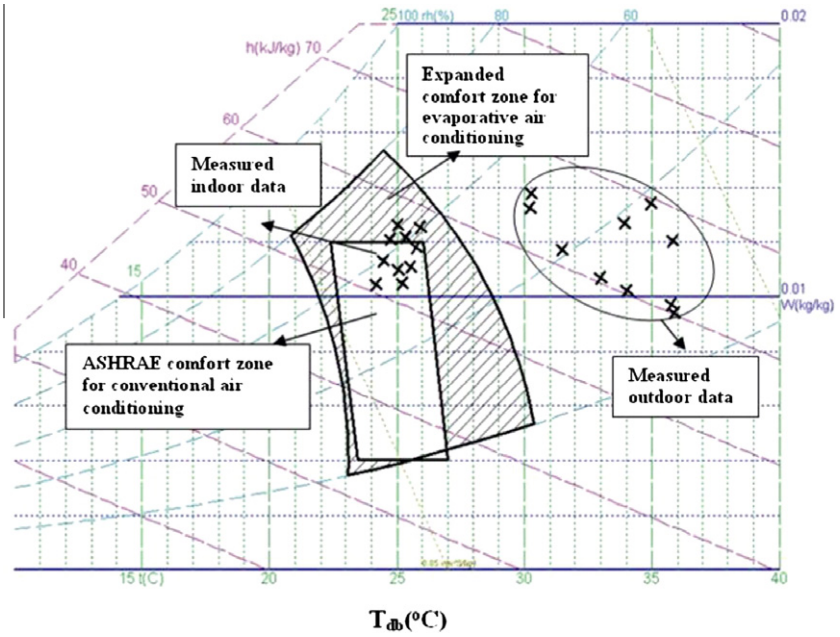


Fig. 9. Thermal comfort zones and measured hourly average data on psychrometric chart.

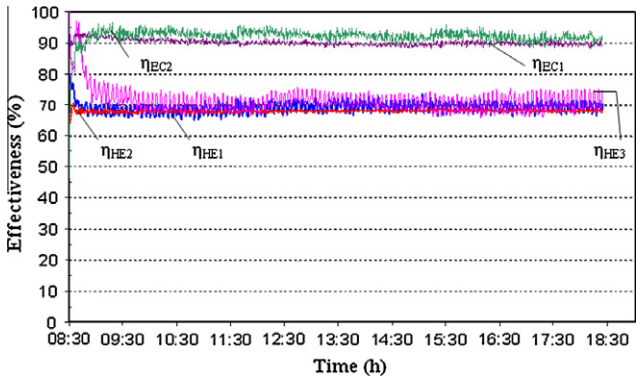


Fig. 10. Variation of effectiveness with time during the day for heat exchangers and evaporative coolers.

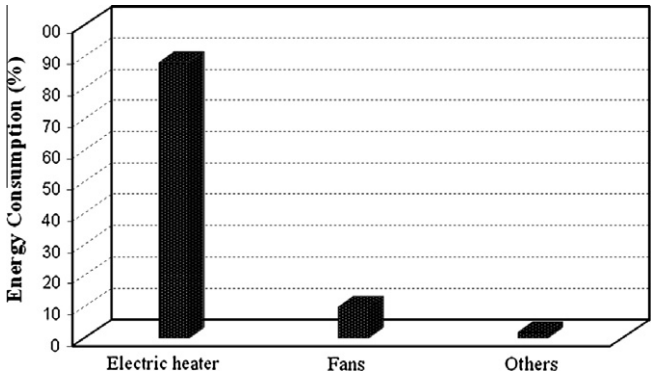


Fig. 11. Daily total energy consumption of electric heater, fans and others.

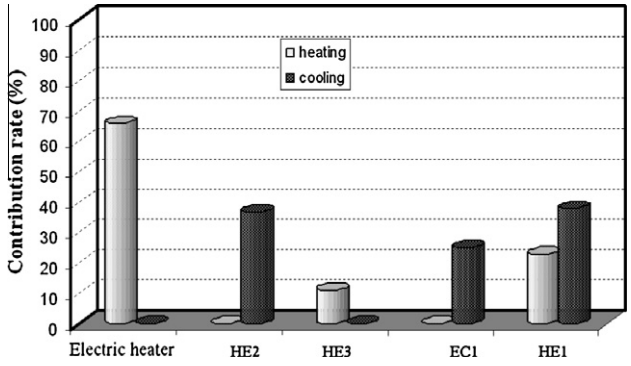


Fig. 12. Daily total contribution of heat exchangers, evaporative cooler 1 and electric heater to cooling and heating processes of the system.

figure, daily average absolute humidity difference in dehumidification process ΔW_1 and humidification processes ΔW_2 and ΔW_3 are found to be 0.006 (kg/kg), 0.005 (kg/kg) and 0.003 (kg/kg), respectively. It is important to notice that moisture removal is higher than that of moisture adding to the fresh air stream. Even if EC1 and EC2 have same capacity and construction, moisture adding to the fresh air in EC1 is higher than that of in EC2. The reason of this is that absolute humidity of the air at the inlet of EC1 is lower than that of the air at the inlet of EC2.

According to ASHRAE standards, maximum indoor absolute humidity must be 0.012 (kg/kg) and temperature of the indoor air must be in the range of 23.5–27 °C in summer season for conventional air conditioning [31,32]. ASHRAE has also developed an expanded comfort zone for evaporative air conditioning applications based on psychrometric charts. It is bounded between 20% and 80% relative humidity curves and between 1.5 m/s to 3.05 m/s air velocity values [33,34]. For evaporative air conditioning, it is more reliable to consider a comfort zone bounded by relative humidity and extended to take into account the cooling effect of increased airflow [35].

Through literature survey, it has been observed that researchers usually focus on supply temperature of desiccant cooling system. Bourdoukan et al. [36] investigated the impact of outside conditions and the efficiency of the system components on the performance of conventional and recirculation configurations of desiccant cooling systems through simulation. They found that for conventional configuration the supply temperature are greater than 25 °C and for recirculation condition the supply temperature is below 20 °C. Camargo et al. [37] analyzed the influence of the outdoor air condition on the air conditioning system performance for several cities. They show that the minimum supply dry bulb temperature is obtained for Brasilia (18.2 °C) and the maximum one for Manaus (22.7 °C).

Fig. 9 depicts the thermal comfort zones defined by ASHRAE for conventional and evaporative air conditioning applications on a psychrometric chart. Hourly average values of the measured data obtained from the experiments for the conditioned room are also presented in this figure. It is observed that indoor air temperature and absolute humidity varies between 24.3–25.9 °C and 0.0106–0.0125 kg/kg, respectively. As can be seen from the figure, some data points sit in the ASHRAE comfort zone for conventional air conditioning and all the data points are in the expanded comfort zone. From this figure, the enhancement of the comfort of outdoor air using the novel configuration of desiccant cooling system can be clearly seen.

The effectiveness for the heat exchangers and evaporative coolers used are given in Fig. 10. Daily average effectiveness of HE1, HE2 and HE3 are calculated as 69%, 68% and 71%, respectively. Daily average effectiveness of EC1 and EC2 are 90% and 93%, respectively.

Fig. 11 shows daily total energy consumption of the electric heater, fans and other equipments. It is shown that energy consumption rates of the electric heater, the fans and other equipments are 86%, 12% and 2%, respectively. As can be seen from the figure, the electric heaters are responsible from a very great portion of the energy consumption. Therefore, a cheap energy source for regeneration is very important to achieve high COP values in these type systems.

Fig. 12 illustrates daily total contribution of the heat exchangers, the evaporative cooler and the electric heater to cooling and heating energy demands. HE1, HE3 and the electric heater are used for heating the regeneration air. HE1 and HE3 together meet 34% of the total heating energy. HE1, HE2 and EC1 are used for cooling process. It is seen that for the fresh air cooling process, the contribution of HE1 and HE2 are higher than EC1, which is the main cooling unit, and these two heat exchangers meet 75% of the total cooling energy. The remaining 25% is produced by EC1.

Cooling capacity and cooling load of the system are shown in Fig. 13. It is observed that the cooling capacity varies between approximately 18–21 kW during the day and the daily average value is 19.67 kW. It is also seen from the figure that the cooling load of the system is in the range of 9–15 kW.

To evaluate the cooling system efficiency, three forms of coefficient of performance (COP) are considered to separately account for total work, thermal energy and total efficiency.

The coefficient of performance is commonly used to compare system performance. Some investigators [15,16] define the COP of desiccant cooling systems as the ratio of conditioned space cooling load to the thermal energy required to regenerate the des-

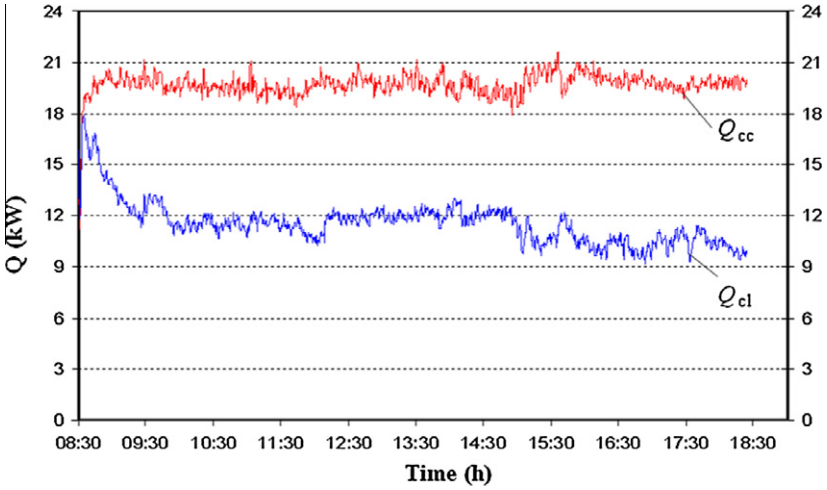


Fig. 13. Variation of cooling capacity and cooling load with time during the day.

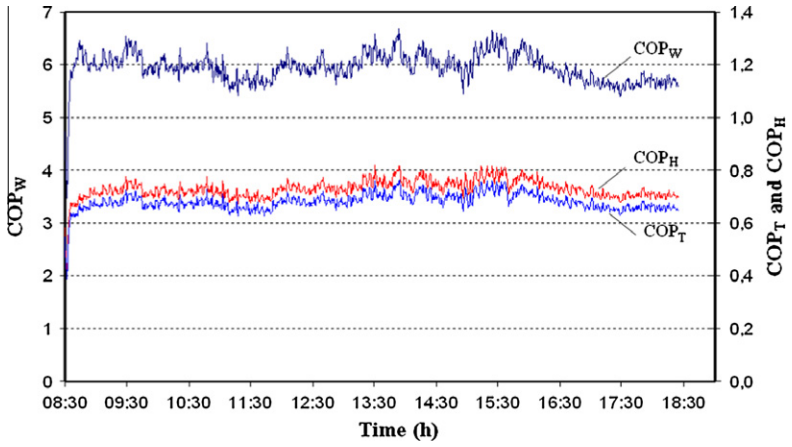


Fig. 14. Variation of COP_T, COP_W and COP_H with time during the day.

iccant. Others [17,25] define the COP as the ratio of the heat removed from the process air stream divided by the thermal energy required to regenerate the desiccant. In this study, the latter definition is used. Several research papers have been dedicated to explore issues about COP of desiccant cooling systems such as [19,20,38,7,39]. It is observed that COP of these former researches varies between 0.35 and 0.7. In Fig. 14, COP_W, COP_H and COP_T are depicted for the whole day. The COP_T of the system varies between 0.64 and 0.76. Meanwhile, the COP_H and COP_W can reach 0.8 and 6.5, respectively. Low COP_T is mostly due to regeneration temperature, the insufficient performance of the desiccant wheel, desiccant material and wheel rotational speed [40]. Although electric heater is used to regeneration heat, the experimental results obtained and discussed in this section illustrate that COP of this system approximately reaches to 0.76.

7. Conclusion

A desiccant based evaporative cooling system is developed and tested experimentally in this study. In the system studied, the moisture of the fresh air is reduced passing it through a solid desiccant wheel and then its temperature is decreased by the direct evaporative cooler. The following conclusions may be drawn from the analysis of the results and discussions:

The temperature of the fresh air received from the outdoor at 35 °C is cooled down to approximately 14 °C supply temperature. Results showed that indoor air conditions are partly in the range of thermal comfort zone defined by ASHRAE for conventional air conditioning and completely in the expanded zone for evaporative air conditioning applications. It is concluded that the system designed can provide comfort conditions in hot and humid climates.

In this study, the impact of the efficiency of key components (desiccant wheel, heat exchangers, and evaporative coolers) on the system has been investigated. Heat exchangers used for heat recovery meet 34% and 75% of total heating and cooling energy, respectively. The evaporative cooler in the fresh air stream (EC1) used as main cooling unit meets 25% of total cooling energy. Average cooling capacity of the system is found to be 19.67 kW and the COP varies between 0.64 and 0.76. The COP_W of the system is found greater than 5. Furthermore, the thermal COP (COP_H) is higher than the total COP (COP_T) and its value reaches to 0.8.

References

[1] Dabrowski A. Adsorption and its applications in industry and environmental protection studies in surface science and catalysis. Elsevier Sci Appl Ind 1998;120:879–916.
[2] Daou K, Wang RZ, Xia ZZ. Desiccant cooling air conditioning: a review. Renew Sustain Energy Rev 2006;10:55–77.
[3] Wang SK. Handbook of air conditioning and refrigeration. McGraw-Hill; 2001.
[4] Fatouh M, Ibrahim TA, Mostafa A. Experimental investigation on a solid desiccant system integrated with a R407C compression air conditioner. Energy Convers Manage 2009;50:2670–9.

[5] Ghali K. Energy savings potential of a hybrid desiccant dehumidification air conditioning system in Beirut. Energy Convers Manage 2008;49:3387–90.
[6] Penington NA. Humidity changer for air conditioning. US patent 2 700 537; 1955.
[7] Bourdoukan P, Wurtz E, Joubert P. Experimental investigation of a solar desiccant cooling installation. Solar Energy 2009;83:2059–73.
[8] Khalid AJ, Dhaidan NS. Application of solar assisted heating and desiccant cooling systems for a domestic building. Energy Convers Manage 2001;42: 995–1022.
[9] Khalid AJ, Mehdi SM. Application of indirect evaporative cooling to variable domestic cooling load. Energy Convers Manage 2000;41:1931–51.
[10] Heidarinejad G, Bozorgmehr M, Delfani S, Esmaeelian J. Experimental investigation of two-stage indirect/direct evaporative cooling system in various climatic conditions. Build Environ 2009;44:2073–9.
[11] Hindoliya DA, Mullick SC. Assessment of utilization potential of direct evaporative cooling for India. Int J Ambient Energy 2006;27:21–8.
[12] Bourdoukan P, Wurtz E, Jouber P. Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components. Energy 2010;35:1057–67.
[13] La D, Dai YJ, Li Y, Ge TS, Wang RZ. Use of regenerative evaporative cooling to improve the performance of a novel one-rotor two-stage solar desiccant dehumidification unit. Appl Therm Eng 2011;xxx:1–7.
[14] Panaras G, Mathioulakis E, Belessiotis V, Kyriakis N. Theoretical and experimental investigation of the performance of a desiccant air-conditioning system. Renew Energy 2010;35:1368–75.
[15] Kanoğlu M, Bolattürk A, Altuntop N. Effect of ambient conditions on the first and second law performance of an open desiccant cooling process. Renew Energy 2007;32:931–46.
[16] Kodama A, Jin W, Goto M, Hirose T, Pons M. Entropic analysis of adsorption open cycles for air conditioning. Part 2: interpretation of experimental data. Int J Energy Res 2000;24:263–78.
[17] Hürdoğan E, Büyükalaca O, Yılmaz T, Hepbaşlı A. Experimental investigation of a novel desiccant cooling system. Energy Build 2010;42:2049–60.
[18] Davanagere BS, Sherif SA, Goswami DY. A feasibility study of a solar desiccant air-conditioning system. Part 1: psychrometrics and analysis of the conditioned zone. Int J Energy Res 1999;23:7–21.
[19] Pons M, Kodama A. Entropic analysis of adsorption open cycles for air conditioning. Part 1: first and second law analyses. Int J Energy Res 2000;24: 251–62.
[20] Camargo JR, Ebinuma CD, Silveira JL. Thermoeconomic analysis of an evaporative desiccant air conditioning system. Appl Therm Eng 2003;23: 1537–49.
[21] Enteria N, Yoshino H, Mochida A, Takaki R, Satake A, Yoshie R, et al. Construction and initial operation of the combined solar thermal and electric desiccant cooling system. Solar Energy 2009;83:1300–11.
[22] La D, Dai Y, Li Y, Ge T, Wang R. Study on a novel thermally driven air conditioning system with desiccant dehumidification and regenerative evaporative cooling. Build Environ 2010;45:2473–84.
[23] Wang D, Zhang J, Xia Y, Han Y, Wang S. Investigation of adsorption performance deterioration in silica gel–water adsorption refrigeration. Energy Convers Manage 2012;58:157–62.
[24] Ge TS, Dai YJ, Wang RZ. Performance study of silica gel coated fin-tube heat exchanger cooling system based on a developed mathematical model. Energy Convers Manage 2011;52:2329–38.
[25] Eicker U, Schneider D, Schumacher J, Ge T, Dai Y. Operational experiences with solar air collector driven desiccant cooling systems. Appl Energy 2010;87: 3735–47.
[26] Holman JP. Experimental methods for engineers. 5th ed. McGraw Hill; 1989.
[27] Esen H, Inalli M, Esen M. Technoeconomic appraisal of a ground source heat pump system for a heating season in eastern Turkey. Energy Convers Manage 2006;47:1281–97.
[28] Sharqawy MH, Mokheimer EM, Habib MA, Badr HM, Said SA, Al-Shayea NA. Energy, exergy and uncertainty analyses of the thermal response test for a ground heat exchanger. Int J Energy Res 2009;33:582–92.
[29] Hepbasli A, Akdemir O. Energy and exergy analysis of a ground source (geothermal) heat pump system. Energy Convers Manage 2004;45:737–53.
[30] State Meteorological Service of Turkey. Meteorological data, Ankara; 1986–2006.
[31] ASHRAE. Handbook of fundamentals, Atlanta; 2005.
[32] ASHRAE, ASHRAE Standard 55. Thermal environmental conditions for human occupancy; 1992.
[33] Crow LW. Weather data related to evaporative cooling. ASHRAE report no 95; 1971.
[34] Watt JR. Evaporative air conditioning handbook. 3rd ed.; 1997.
[35] Bom GJ, Foster R, Dijkstra E, Tummers M. Evaporative air-conditioning applications for environmentally friendly cooling, World Bank technical paper no 421; 1999.
[36] Bourdoukan P, Wurtz E, Joubert P. Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components. Energy 2010;35:1057–67.
[37] Camargo JR, Godoy E, Ebinuma D. An evaporative and desiccant cooling system for air conditioning in humid climates. J Braz Soc Mech Sci Eng 2005; XXVII(3):247.
[38] Jain S, Dhar PL. Evaluation of solid-desiccant-based evaporative cooling cycles for typical hot and humid climates. Int J Refrig 1995;18:287–96.
[39] Medeiros JM, Silva MG, Santos JC, Gurgel JM. In: 20th International congress of mechanical engineering, Gramado, Brazil; 2009.
[40] Kanoğlu M, Çarpınlıoğlu MO, Yıldırım M. Energy and exergy analyses of an experimental open cycle desiccant cooling system. Appl Therm Eng 2004; 24:919–32.