Cairo University

Journal of Advanced Research

ORIGINAL ARTICLE

Experimental performance study of a proposed desiccant based air conditioning system

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A B S T R A C T

An experimental investigation on the performance of a proposed hybrid desiccant based air conditioning system referred as HDBAC is introduced in this paper. HDBAC is mainly consisted of a liquid desiccant dehumidification unit integrated with a vapor compression system (VCS). The VCS unit has a cooling capacity of 5.27 kW and uses 134a as refrigerant. Calcium chloride ($CaCl_2$) solution is used as the working desiccant material. HDBAC system is used to serve low sensible heat factor applications. The effect of different parameters such as, process air flow rate, desiccant solution flow rate, evaporator box and condenser box solution temperatures, strong solution concentration and regeneration temperature on the performance of the system is studied. The performance of the system is evaluated using some parameters such as: the coefficient of performance (COP_a), specific moisture removal and energy saving percentage. A remarkable increase of about 54% in the coefficient of performance of the proposed system over VCS with reheat is achieved. A maximum overall energy saving of about 46% is observed which emphasizes the use of the proposed system as an energy efficient air conditioning system.

Introduction

Increasing of occupant comfort demands are leading to rising requirement for air conditioning, but deteriorating global energy and environment crisis are starving for energy saving and environmental protection. The need to come up with the new energy saving as well as environmentally friend air conditioning systems has been more urgent than ever before. The liquid desiccant dehumidification systems integrated with VCS driven by low-grade heat sources can satisfactorily meet those

needs; meanwhile, they provide an ideal area for the application of waste heat discharged from local factories, and the employment of brine solutions as absorbent brings less damage to environment. The earliest liquid desiccant system was suggested and experimentally tested by Lof [1] using triethylene glycol as the desiccant. Many researchers [2–5] have all described different air handling systems using liquid desiccants.

Adnan et al. [6] introduced an energy efficient system using liquid desiccant which is proposed to overcome the latent part of the cooling load in an air conditioning system. It can be concluded that the proposed system can be used effectively to reduce electric energy consumption in air conditioning to about 0.3 of the energy consumed by a conventional air conditioning system. Mohan et al. [7] studied the performance of absorption and regeneration columns for a liquid desiccant-vapor compression hybrid system. They reported that higher the specific humidity and lower the temperature of the inlet air, higher will be the dehumidification in the absorber. Similarly,

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Nomenclature

C_p	specific heat at constant pressure, kJ/kg K		Reg	regeneration
h	enthalpy, kJ/kg		s	desiccant solution
$\dot{m}$	mass flow rate, kg/s		vap	vapor
$\dot{Q}$	heat transfer rate, kW			
T	temperature, °C		*Abbreviations*	
V	volume flow rate, l/min		AH	auxiliary heater
X	desiccant solution concentration, kg_d/kg_s		COP	coefficient of performance
y	air humidity ratio, kg_v/kg_{da}		$CaCl_2$	calcium chloride
			HDBAC	hybrid desiccant based air conditioning system
Subscripts			SMR	specific moisture removal, kg_v/kg_s
a	air		VCS	vapor compression system
da	dry air			

the regeneration can be increased by increasing the temperature and decreasing the specific humidity of the inlet air to the regenerator. Jia et al. [8] introduced a hybrid desiccant-assisted air conditioner and split cooling coil system, which combines the merits of moisture removal by desiccant and cooling coil for sensible heat removal, which is a potential alternative to conventional vapor compression cooling systems. It is found that, compared with the conventional VCS; the hybrid desiccant cooling system economizes 37.5% electric energy consumption.

Ge et al. [9] introduced a solar driven two-stage rotary desiccant cooling system and a vapor compression system are simulated to provide cooling for one floor in a commercial office building in two cities with different climates: Berlin and Shanghai. Results illustrated that the required regeneration temperatures are 55 °C in Berlin and 85 °C in Shanghai. As compared to the vapor compression system, the desiccant cooling system has better supply air quality and consumes less electricity. Jongsoo et al. [10] provided a detailed evaluation of the performance of a four-partition desiccant wheel to make a low-temperature driving heat source possible and achieve considerable energy saving by the simulation and experiment. They mentioned that hybrid air-conditioning system improves COP by approximately 94% as compared to the conventional vapor compression-type refrigerator. Niu et al. [11] introduced a performance analysis of liquid desiccant based air-conditioning system under variable fresh air ratios. They reported that, compared to a conventional air-conditioning system with primary return air, the liquid desiccant based system consumes notably less power. The maximum power saving ratio is 58.9% when the fresh air ratio is 20%, and the minimum is 4.6% when the fresh air ratio is 100%. Researches on hybrid cooling system are also reported [12–16]. Jiazhen et al. [17] introduced and tested a desiccant wheel (DW)-assisted separate sensible and latent cooling (SSLC) air-conditioning systems by using CO_2 and R-410a as refrigerant. They found that at a regeneration temperature of 50 °C, the coefficient of performance (COP) of the vapor compression cycles improved by 7% from the respective baseline systems for both refrigerants. A two desiccant-coated heat exchangers (DCHEs), which are actually fin-tube heat exchanging devices coated with silica gel and polymer materials respectively, are investigated experimentally by Ge et al. [18]. An experimental setup was designed and built to test the performance of this unit. They found that

this desiccant-coated fin-tube heat exchanger well overcomes the side effect of adsorption heat which occurs in desiccant dehumidification process, and achieves good dehumidification performance under given conditions. The silica gel coated heat exchanger behaves better than the polymer one. The influences of regeneration temperature, inlet air temperature and humidity on the system performance in terms of average moisture removal rate and thermal coefficient of performance were also analyzed. The performance of DCHE system, using conventional silica gel as desiccant material and a novel solar driven desiccant coated heat exchanger cooling (SDCC) system is also proposed by Ge et al. [19,20].

In this paper, experimental tests are carried out to investigate the performance of the proposed HDBAC system. The effects of the relevant operating parameters on the performance of the whole system are studied and analyzed. The HDBAC system is designed to meet the needs of cooling, dehumidification and reducing energy consumption in hot humid areas; places with high latent load portions; such as supermarkets, theaters or auditoriums.

Experimental

The schematic diagram of the proposed HDBAC system is shown in Fig. 1a. HDBAC system is consisted mainly of a liquid desiccant dehumidification unit integrated with a vapor compression system (VCS). The experimental test-rig of this system is shown in Fig. 1b. Calcium chloride is used as a working desiccant material in this investigation. The VCS unit has a cooling capacity of 5.27 kW and uses 134a as refrigerant.

From the schematic diagram shown in Fig. 1a, the proposed HDBAC comprises on different four energy cycles. These cycles are: desiccant solution cycle, process air cycle, VCS cycle and cooling water cycle. The evaporator box (A) includes the evaporator (cooling coil) of the VCS unit. The strong desiccant solution at state (6) is cooled by the cooling coil to the desired conditions. The evaporator and condenser boxes are made of a 0.5 mm stainless steel sheet with dimensions of 60 cm × 60 cm × 25 cm.

For air cycle, the process air at state (1) is injected into the evaporator box. The process air is then cooled and dehumidified to the desired conditions at state (2) to be supplied to the conditioned space. The ambient air conditions are fluctuates from 41 °C, 48% RH and 42 °C, 46% RH.
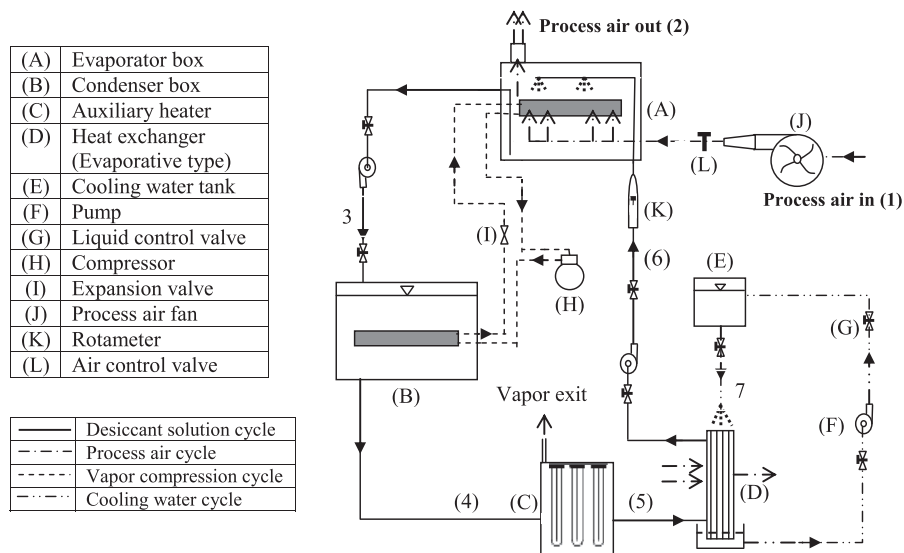
(A)	Evaporator box
(B)	Condenser box
(C)	Auxiliary heater
(D)	Heat exchanger (Evaporative type)
(E)	Cooling water tank
(F)	Pump
(G)	Liquid control valve
(H)	Compressor
(I)	Expansion valve
(J)	Process air fan
(K)	Rotameter
(L)	Air control valve

———	Desiccant solution cycle
—·—·—	Process air cycle
- - - -	Vapor compression cycle
—··—··—	Cooling water cycle

Fig. 1a Schematic diagram of HDBAC system.
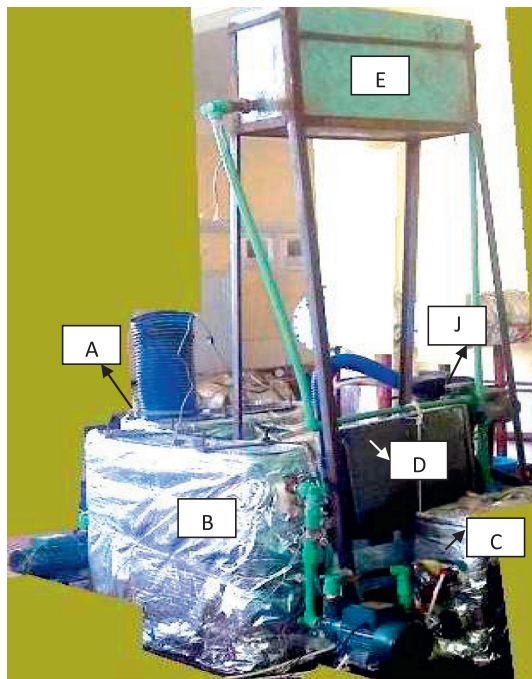
Fig. 1b The experimental test-rig of HDBAC system.

Fig. 1c Psychometric process of the proposed HDBAC system.

For desiccant solution cycle, the strong desiccant solution at state (6) is directly sprayed on the VCS evaporator inside the evaporator box. While the dilute desiccant solution at state (3) is pumped to the condenser box (B) which contains the condenser of the VCS unit. The dilute desiccant solution is preheated to state (4) by the condenser heat. The preheating process is intended to save some of the energy required for desiccant solution regeneration process. An auxiliary heater (C) is used to completely regenerate the desiccant solution to the required operation concentration.
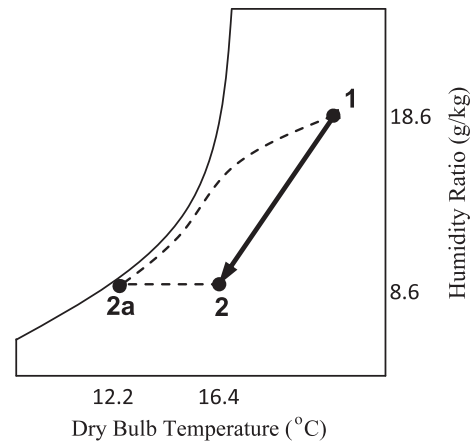
For cooling water cycle, an evaporative type heat exchanger (D) with an effectiveness of 0.85 is used for pre-cooling the strong desiccant solution from state (5) to state (6) before it has been delivered to the evaporator box. The cooling water required for this process is received from a cooling water tank (E) at state (7). The cooling water temperature is kept nearly constant during experiment at 24 °C.

Some components of the experimental test rig are perfectly insulated. These components are such as, the evaporator box, condenser box and auxiliary heater. The process air and desiccant solution flow rates are controlled by using control valves.

The psychometric chart of the process air of the proposed system is shown in Fig. 1c. The solid line; process 1–2; denotes the HDBAC system process. The dashed line 1–2a–2 represents the comparable conventional system (process 1–2a is cooling with dehumidification over the direct expansion evaporator of the VCS unit and process 2a–2 is reheating to the desired conditions of the supply air). This conventional system is called VCS with reheat.

Measurements and instrumentation

Suitable measuring devices for data recording of the experimental runs are used. Air and solution temperatures are measured using type K thermocouples and a digital temperature reader with accuracy of 0.11 °C. Solution flow rates are measured using glass rotameters with 2% full scale accuracy. The density of the desiccant solution is measured using an accurate digital scale with accuracy of 0.01 g. These densities at its known temperatures are used to determine the concentrations of the desiccant solution from $CaCl_2$ properties table. Air velocity and humidity are measured using a multi-function hot wire measuring device with accuracy of 0.015 m/s for the velocity and of 3% for the relative humidity. The power consumption is measured using a watt meter with accuracy of 0.035 kW. The uncertainty of air and solution mass flow rate is 5.8% and 4.5%; respectively. The uncertainty of air enthalpy, heat transfer rate and COP is 2.7%, 6.1% and 8.2%; respectively. The uncertainty of specific moisture recovery is 8.5%.

Experimental tests are carried out to evaluate the performance of the proposed HDBAC system at different conditions. The following variables are required to be measured, temperature and humidity of the process air at the inlet and exit of the evaporator box, solution regeneration temperature, solution concentrations and temperatures, air velocity for process air and solution flow rates.

Performance analysis

In the present work, some important parameters are used for evaluating the performance of the proposed HDBAC system as follows.

Coefficient of performance (COP)

The proposed system coefficient of performance COP_a is calculated from:

$$COP_a = \frac{\dot{m}_a(h_1 - h_2)}{\dot{W}_C + \dot{Q}_{AH}} \tag{1}$$

where $\dot{m}_a$ is the mass flow rate of air, h is the enthalpy of air, $\dot{W}_C$ is the compressor power consumption and $\dot{Q}_{AH}$ is the auxiliary regeneration heat rate which may be calculated from:

$$\dot{Q}_{AH} = \dot{m}_S(h_{S5} - h_{S4}) \tag{2}$$

where $\dot{m}_S$ is the mass flow rate of desiccant solution and the enthalpy of $CaCl_2$ solution may be obtained from [21] as follows:

$$h_S = C_{PS}T_S \tag{3}$$

where C_{ps} is the specific heat of $CaCl_2$–H_2O solution at constant pressure in (J/kg °C) and it can be calculated in terms of its concentration X_s (kg$_d$/kg$_s$) and temperature T_s (°C) from:

$$C_{PS} = 4027 + 1.859T_s - 5354X_s + 3240X_s^2 \tag{4}$$

The VCS with reheat coefficient of performance COP_b is calculated as follows:

$$COP_b = \frac{\dot{m}_a(h_1 - h_{2a})}{\dot{W}_C + \dot{Q}_{Re\ heat}} \tag{5}$$

$$\dot{Q}_{Re\ heat} = \dot{m}_a(h_2 - h_{2a}) \tag{6}$$

Specific moisture removal (SMR)

The specific moisture removal is defined as the amount of moisture removed from process air per each kilogram of desiccant solution. It can be calculated from:

$$SMR = \frac{\dot{m}_a(y_{a1} - y_{a2})}{\dot{m}_S} \tag{7}$$

where y_{a1} and y_{a2} are the humidity ratio of process air at inlet and exit of evaporator box; respectively.

Results and discussion

Experimental tests have been carried out at different parameters to evaluate the performance of the presented HDBAC system. These parameters are such as, desiccant solution flow rate, air flow rate, evaporator box and condenser box solution temperatures, strong solution concentration and regeneration temperature.

Effect of evaporator box solution temperature

Figs. 2a and 2b show the effect of desiccant solution temperature inside the evaporator box ($T_{S,ev}$) on the proposed system coefficient of performance (COP_a) and specific moisture removal (SMR); respectively. Fig. 2a shows that the COP_a increases with the increase of both $T_{S,ev}$ and desiccant solution volume flow rate (V_S). When the desiccant solution temperature inside the evaporator box is increased from 10 °C to 22 °C at constant V_s of 4 l/min, the COP_a of the HDBAC system will achieve an increase of 40.5%. On the other hand from Fig. 2b, by increasing $T_{S,ev}$ from 10 °C to 22 °C at constant V_s of 4 l/min, the SMR is decreased by about 36.2%. The analysis of the previous situation may be viewed as follows, when the desiccant solution temperature inside the evaporator box increases; the ability of desiccant solution to absorb moisture
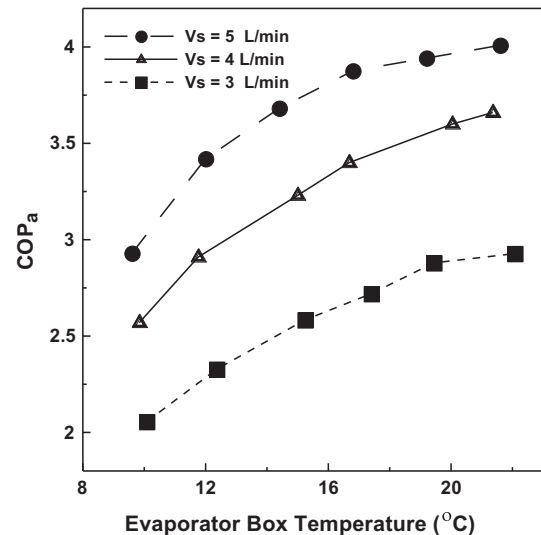
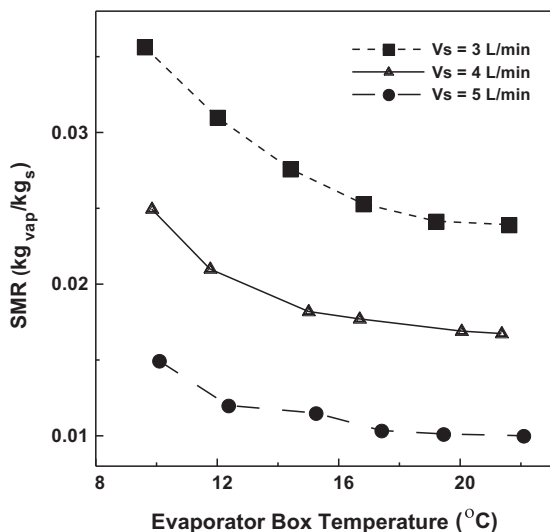**Fig. 2a** Effect of evaporator box temp. on COP_a.

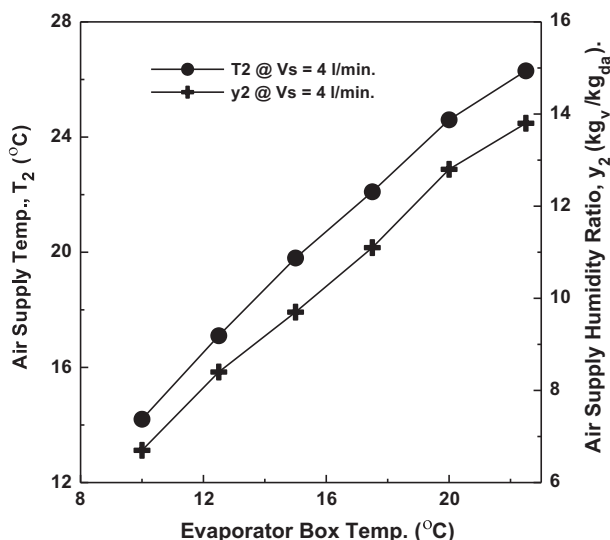**Fig. 2b** Effect of evaporator box temp. on SMR.

Fig. 2d Effect of evaporator box temp. on T_2 and y_2.

from the process air is reduced. This is referred to the decrease of the vapor pressure difference between process air and desiccant solution resulting in lowering SMR. Also, as $T_{S,ev}$ increases, the desiccant concentration at the exit of the evaporator box is increased leading to low regeneration heat and higher COP_a. Also, as $T_{S,ev}$ increases, the condenser box temperature is increased resulting in low additional regeneration heat in the auxiliary heater. From Fig. 2a, at V_s of 3 l/min the cooling capacity and additional regeneration heat are 9.1 kW and 1.8 kW at $T_{S,ev}$ of 14 °C while these values at $T_{S,ev}$ of 20 °C are 7.9 kW and 1.1 kW; respectively. This will lead to a COP_a of 2.36 at $T_{S,ev}$ of 14 °C while a COP of 2.82 at $T_{S,ev}$ of 20 °C. The comparison between the coefficient of performance of the presented system and that of the VCS with reheat at different $T_{S,ev}$ is shown in Fig. 2c. The COP_a of the proposed system is found to be 54% greater than that of VCS with reheat. The higher latent load gain by the HDBAC system with less power consumption explains the increase of

COP_a compared to COP_b of VCS with reheat. The effect of $T_{S,ev}$ on the supply air temperature and humidity ratio is shown in Fig. 2d.

Effect of regeneration temperature

The effect of regeneration temperature (T_{reg}) on the system COP_a, strong solution concentration (x_6) and SMR is shown in Figs. 3a–3c; respectively. From Fig. 3a, the COP_a increases with the increase of T_{reg} until it reaches nearly to 70 °C, then COP_a starts to decrease. This may be explained as follows, increasing T_{reg} will directly increase the strong solution concentration at state (6) and hence increasing the SMR as shown in Figs. 3b and 3c; respectively. As x_6 increases, the ability of the desiccant solution to absorb moisture increases, leading to high latent load removing capacity by the HDBAC system. As a result, the COP_a increases. For further increase in regeneration temperature, the regeneration heat required at the same
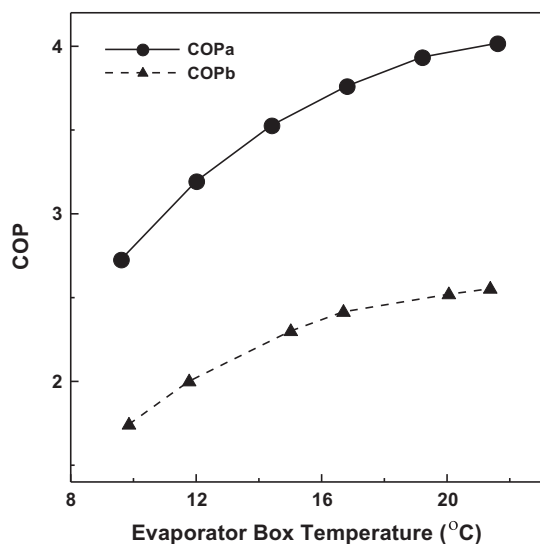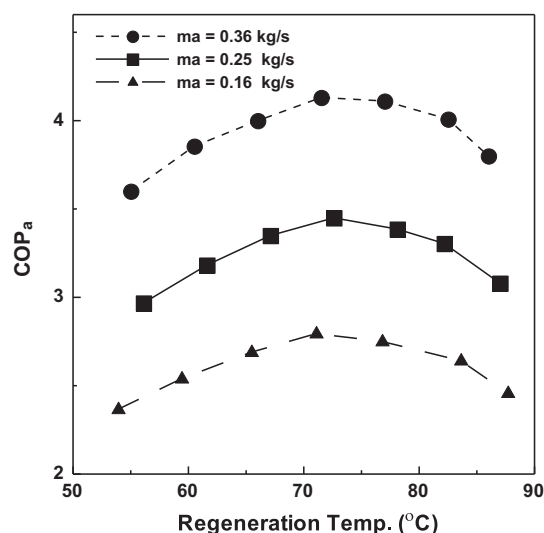

Fig. 2c Effect of evaporator box temp. on COP.

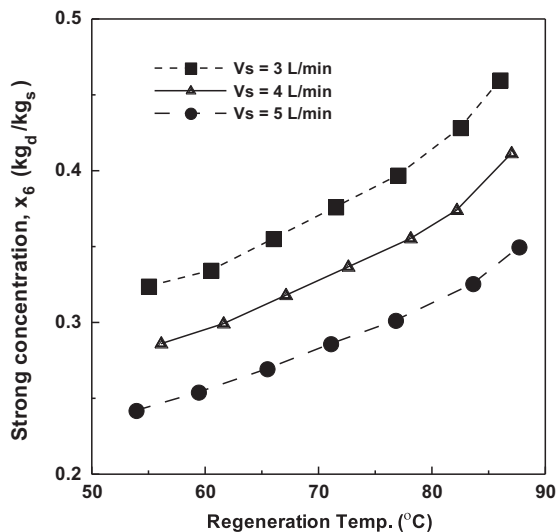**Fig. 3a** Effect of regeneration temp. on COP_a.

Fig. 3b   Effect of regeneration temp. on x_6.
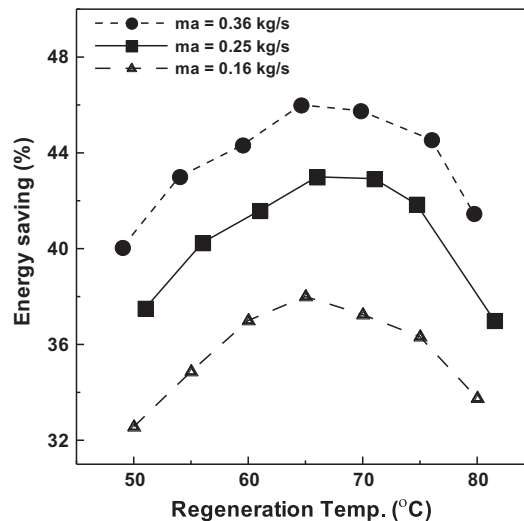

Fig. 3d   Effect of regeneration temp. on energy saving%.
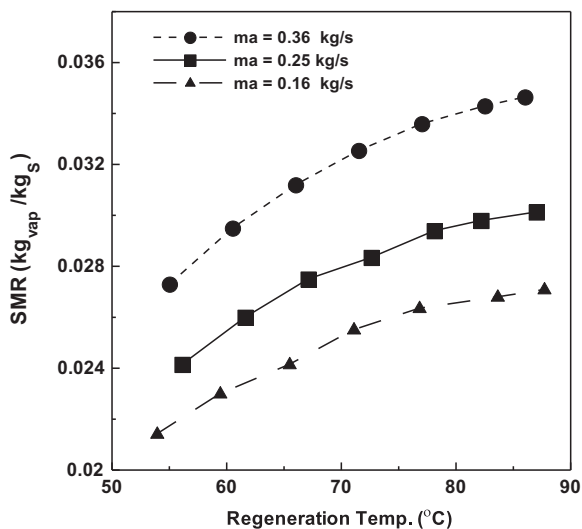

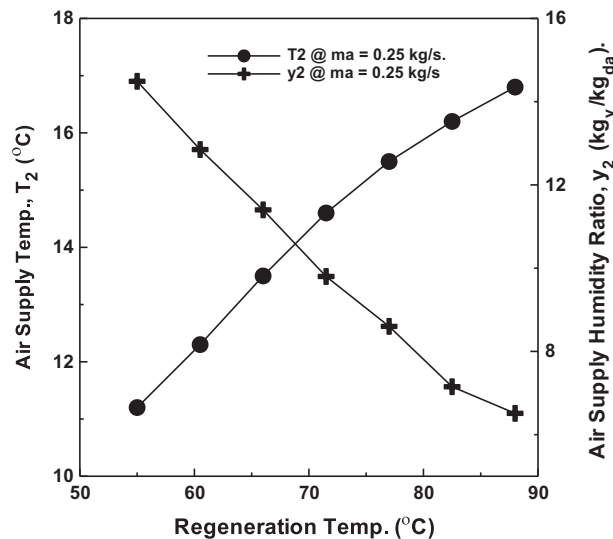**Fig. 3c**   Effect of regeneration temp. on SMR.

Fig. 3e   Effect of regeneration temp. on T_2 and y_2.

desiccant solution flow rate will increase. The increase of the regeneration heat will represent a penalty on the COP_a and resulting in its decrease. At air mass flow rate of 0.36 kg/s and a desiccant solution volume flow rate of 3.0 l/min, increasing the regeneration temperature from 70 °C to 88 °C (which represents an increase of about 24.3% of the regeneration heat), will decrease the COP_a by a percentage of 12.6%. At the same conditions, both SMR and x_6 will increase by about 6.25% and 22.3%; respectively. Also, from Fig. 3a the COP_a is directly increased with the air mass flow rate due to the increase in the total cooling capacity of the process air. On the other hand as shown from Fig. 3b, by increasing the desiccant solution volume flow rate, the desiccant solution concentration x_6 is decreased at the same regeneration temperature. This may be explained as follows, increasing the desiccant solution volume flow rate will decrease the contact time between the desiccant solution and the auxiliary heater. As a result, the

evaporation rate from the auxiliary heater is decreased resulting in low desiccant concentration.

The percentage of energy savings of the proposed system with the regeneration temperature at different air mass flow rate is shown in Fig. 3d. Increasing the regeneration temperature will increase the percentage of the energy saving till T_{reg} reaches nearly to 70 °C. The maximum percentage of energy saving is achieved at T_{reg} near to 70 °C. When the air mass flow rate increases, the percentage of energy saving is also increased. An overall energy saving in the range of 33–46% is observed during experiments. The effect of T_{reg} on the supply air temperature and humidity ratio is shown in Fig. 3e.

Effect of condenser box solution temperature

The effect of the desiccant solution temperature inside the condenser box ($T_{S,cond}$) on the system performance measures
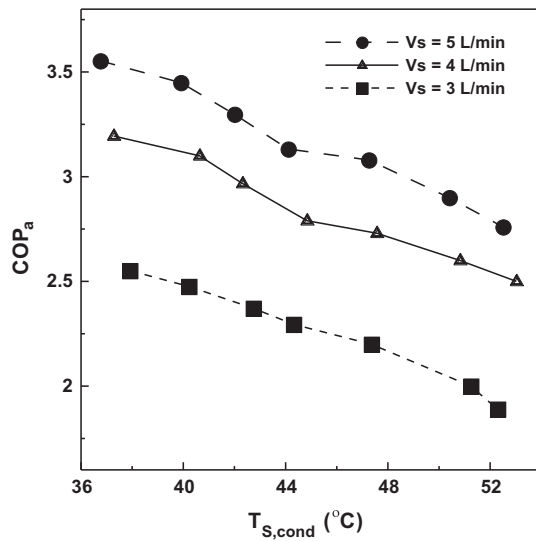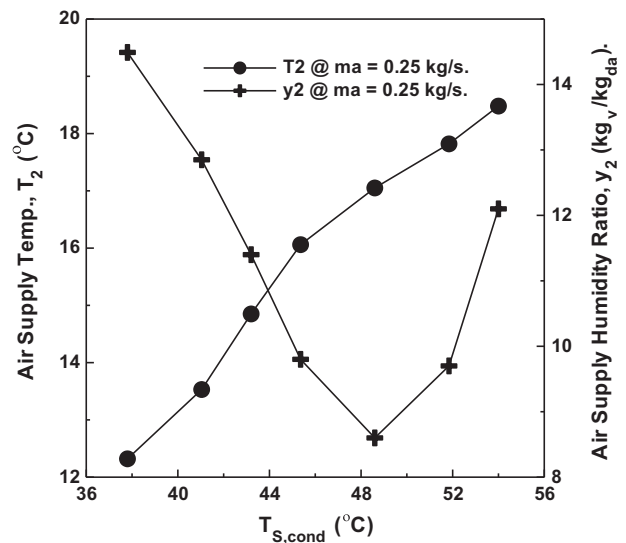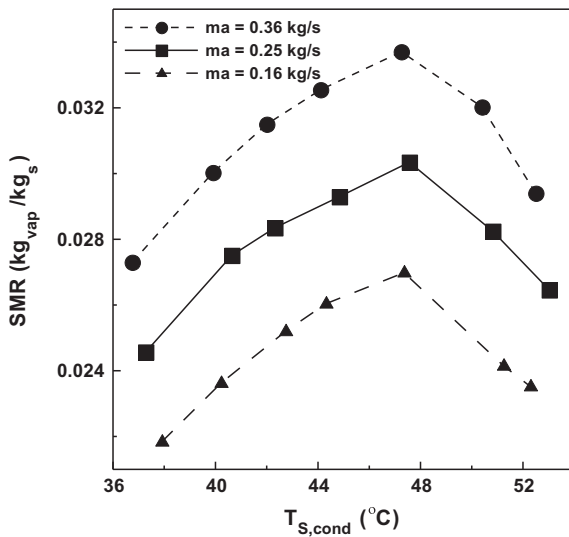
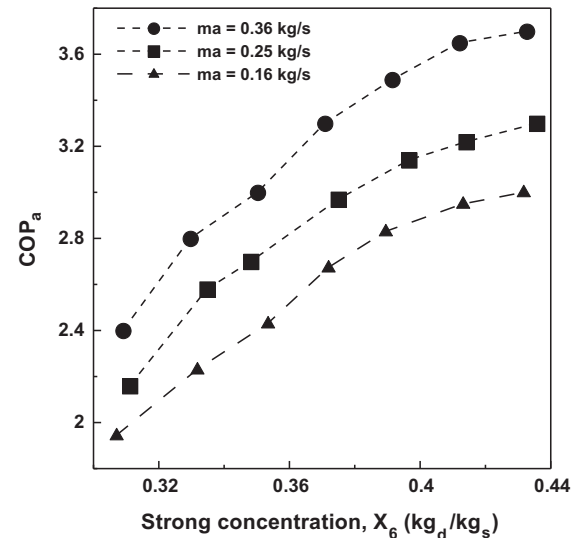**Fig. 4a** Effect of condenser box temp. on COP_a.

Fig. 4c Effect of condenser box temp. on T_2 and y_2.

Fig. 4b Effect of condenser box temp. on SMR.

Fig. 5a Effect of strong concentration, x_6 on COP_a.

COP_a and SMR is shown in Figs. 4a and 4b; respectively. Increasing $T_{S,cond}$ will directly decrease the COP_a. This may be viewed as; the increase of the desiccant solution temperature will increase the condenser temperature leading to high compressor power which represents a penalty on COP_a. On the other hand the SMR increases with the increase of the $T_{S,cond}$ till it reaches nearly to 47 °C, then it starts to decrease. At air mass flow rate of 0.16 kg/s and by increasing the $T_{S,cond}$ from 47 °C to 52 °C, the SMR is decreased by a percentage of 14.1. This may be partially referred to that, the increase of the condenser temperature will increase the temperature of cooling coil and then reduces the ability of desiccant solution to absorb moisture from the process air. The effect of $T_{S,cond}$ on the supply air temperature and humidity ratio is shown in Fig. 4c.

Effect of strong solution concentration

Figs. 5a and 5b show the effect of strong solution concentration x_6 on the system COP_a and SMR; respectively. As x_6 in-

creases, both COP_a and SMR are increased. Increasing x_6 will increase the affinity of desiccant solution to absorb moisture, leading to an observed increase in both COP_a and SMR. This will be explained as follows, as the moisture absorbed from process air is increased, the cooling load that has been removed from air is increased. This increase results in higher COP_a and SMR. An increase of x_6 from 0.32 to 0.43 at an air mass flow rate of 0.36 kg/s will increase the COP_a and SMR by about 36.28% and 31.2%; respectively. The effect of x_6 on the supply air temperature and humidity ratio is shown in Fig. 5c.

Conclusions

A hybrid desiccant based air conditioning system of a small capacity is designed and experimentally tested. At specific design and operating conditions and from the analysis of the
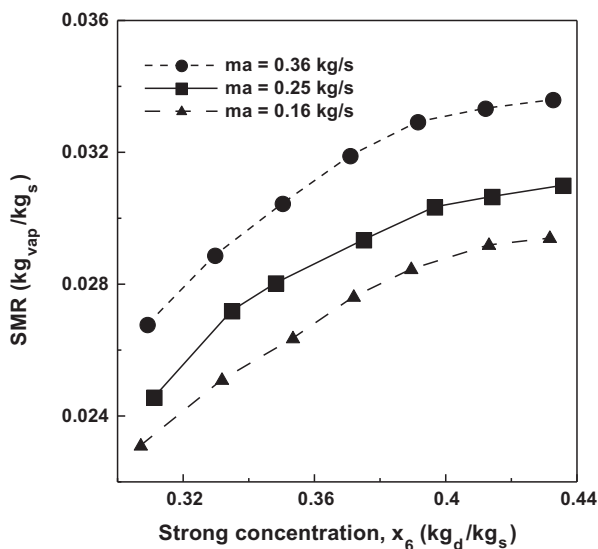
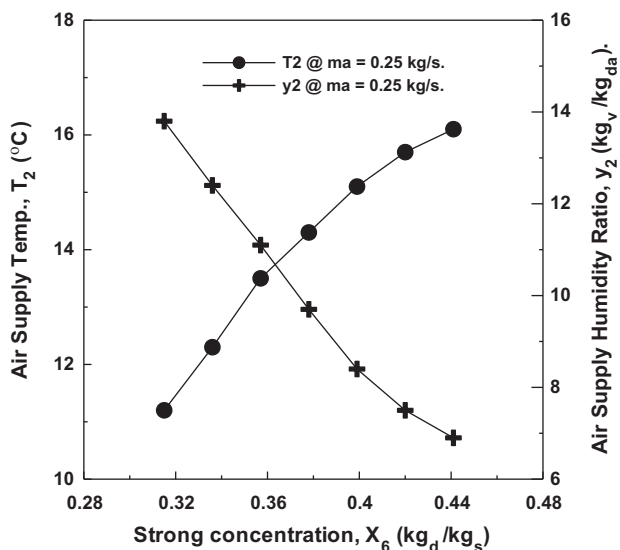**Fig. 5b**   Effect of strong concentration, x_6 on SMR.

Fig. 5c   Effect of strong concentration, x_6 on T_2, y_2.

experimental results, some important conclusions can be summarized as follows:

- The coefficient of performance of the proposed system is found to be 54% greater than that of VCS with reheat at typical operating conditions.
- The HDBAC system integrated with a 5.27 kW conventional VCS can replace a VCS with reheat with a cooling capacity of 9.13 kW.
- The coefficient of performance and the specific moisture removal of the proposed system are both increased with increasing both air and desiccant solution flow rates.
- An increase of strong solution concentration will increase both COP_a and SMR.
- The COP_a increases and SMR decreases by increasing the temperature of the desiccant solution inside the evaporator.

- The COP_a is decreased and SMR is increased when the regeneration temperature is increased.
- The HDBAC system has been achieved a percentage of an energy savings in the range of 33–46%.

Conflict of interest

The authors have declared no conflict of interest.

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