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AN EXPERIMENTAL STUDY ON MULTI-PURPOSE DESICCANT INTEGRATED VAPOR-COMPRESSION AIR-CONDITIONING SYSTEM

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In this paper, a multi-purpose hybrid desiccant integrated vapor compression air conditioning system of a small capacity is experimentally investigated. The system, referred as HDAC, is designed to meet the cooling load of spaces having large latent heat portions and at the same time to extract water from atmospheric air. The system is mainly consisted of a liquid desiccant dehumidification unit integrated with a vapor compression system (VCS). The dehumidification unit uses lithium chloride (LiCl) solution as the working material. The effect of different parameters such as, desiccant solution flow rate, process air flow rate, evaporator and condenser temperatures, strong solution concentration and regeneration temperature on the performance of the system is studied. This system has a water recovery rate of 6.7 l/h.TR of pure water at typical north Egyptian climate. The HDAC system has a COP as high as 3.8 (an improvement of about 68% over the conventional VCS. The system offers a total cooling capacity of about 1.75 TR using a 0.75 TR VCS unit. Finally, the proposed system is found to have a payback time of about 10 months without any considerable extra capital cost compared to the known split air conditioning system. The results emphasize the potential benefits of the HDAC system.

KEYWORDS: *hybrid system, dehumidification, water recovery, vapor compression system, liquid desiccant, life cycle analysis.*

1. INTRODUCTION

As the energy shortage emerges as an issue of growing concern in the world, coupled with the threat to environment posed by the conventional refrigerants, 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 partially 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. A review of liquid-desiccant systems was done by [1]. Desiccant system using triethylene glycol integrated with VCS has long been adopted for both industrial and agricultural purposes, such as humidity control in textile mill and post harvest low-temperature crop-drying in stores, and is now taking a more prominent role in the air conditioning field. Its economical advantages and effective humidity control at low and moderate temperature really dwarfs the conventional method of humidity control, (generally, lowering the air temperature to below the dewpoint temperature). Adnan *et al*. [2] 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*. [3] 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 inlet air, higher will be the dehumidification in the absorber. Similarly, the regeneration can be increased by increasing the temperature and decreasing the specific humidity of inlet air to the regenerator. Jia *et al*. [4] introduced a hybrid desiccant-assisted air conditioner (HDAC) 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 with reheat, the HDAC system economizes 52.5% electric energy consumption.

NOMENCLATURE

C_P	Specific Heat at const. pressure, kJ/kg.K
m	Mass Flow Rate, kg/s
Q	Heat Transfer Rate, kW
T	Air Temperature, °C
V	Volume Flow Rate, l/min
w	Air Humidity Ratio, kg_{vapor}/kg_{air}
x	Desiccant Solution Concentration, kg_d/kg_{sol}

Subscripts

a	Air
cc	Cooling Capacity
d	Desiccant
pre	Preheating
ss	Strong Solution
sol	Desiccant Solution
reg	Regeneration
ws	Weak Solution

Abbreviations

AH	Auxiliary Heating coil
COP	Coefficient Of Performance
IAQ	Indoor Air Quality
LiCl	Lithium Chloride
LCS	Life Cycle Savings
SMR	Specific Moisture Recovery, kg_{vap}/kg_{sol}
VCS	Vapor Compression System
WRR	Water Recovery Rate, l/h
PWF	Present Worth Factor
PP	Payback Period

Ahmed *et al*. [5] simulated a hybrid open-cycle absorption and liquid desiccant system using LiBr for the process of absorption and dehumidification. The simulation model of the hybrid cycle is formulated with a partly closed-open solar regenerator for regenerating the weak desiccant and a packed tower dehumidifier for the dehumidification of ambient air. The COP obtained is about 50% higher than that of a conventional absorption machine. Investigation by Yadav [6] showed that the hybrid vapor compression system was more promising under high latent load or higher ambient humidity conditions, and significant energy saving can be achieved over the conventional VCS. Yadav and Kaushik [7] have studied a hybrid solid desiccant system. It was found that the system resulted in 25% energy saving over a VCS. Dai *et* *al.* [8] introduced a hybrid air conditioning system, which consists of a desiccant dehumidifier, evaporative cooling and vapor compression air conditioning. They found that the cooling production increased by 20-30% compared to the VCS alone, it is also found that the electric power consumption and size of vapor compression cycle can be further reduced. Burns *et al.* [9] studied three hybrid system configurations for supermarket applications (high latent load) and a comparison of their performance with conventional air-conditioning system was made. The cycles termed as ventilation-condenser cycle, recirculation-condenser cycle and ventilation-heat exchanger cycle. They reported that these cycles would give energy savings, in comparison to the conventional air-conditioning systems, ranging from 56.5 to 66%. Close *et al.* [10] uses an energy weight as one unit of electrical energy weighted three that of thermal energy. Singh *et al.* [11] have analyzed the afore-mentioned three hybrid cycles, introduced in [9], for Indian climatic conditions. Modeling of the dehumidifier operating at a fixed regeneration temperature of 135˚C, and regeneration to process air area ratio of 0.33 is done using the performance data from a manufacturer. It is reported that energy savings ranging from 30% to 50% can be achieved at lower SHF.

In the present work, the HDAC system is designed to meet the needs of cooling, dehumidification, reducing energy consumption and extracting pure water from atmospheric air. This system is very suitable to be used in humid climates, places with high latent load components, such as supermarkets, theaters or auditoriums, and also suitable for places with scarcity of potable water, such as coastal areas, islands and workplaces inside oceans (e.g. oil fields). The proposed HDAC system is studied and the effects of the relevant operating parameters on the performance of the whole system are studied and analyzed from the thermal as well as economical point of view.

2. THE HDAC SYSTEM

A schematic diagram of the multi-purpose integrated HDAC system is shown in Fig. 1. The proposed system is mainly consisted of a liquid desiccant dehumidification unit integrated with a vapor compression system. The dehumidification unit uses lithium chloride (LiCl) solution as the desiccant material. The psychometric process of the process air of the proposed system is shown in Fig. 2. This process is denoted by solid line; process 1-2. The dashed line 1-3-2 represents the equivalent conventional system (process 1-3 is cooling and dehumidification over the DX cooling coil and process 3-2 is reheating).

In Fig. 1, the desiccant strong solution of concentration x_8 is pumped and uniformly sprayed over the evaporator surface area. The process air stream at ambient state (1) is passed through the evaporator in cross flow scheme. The process air is then cooled and dehumidified to point (2). The desiccant weak solution is then collected and drawn to the weak solution tank (A). The weak solution (4) is then pumped to the heat exchanger (E) which uses the exhaust heat rejected from the condenser of the VCS to preheat the weak solution. An additional heat is required to completely regenerate the weak solution in the regenerator (B) to reach state (8). The water vapor exit from the regenerator at state (6) is collected and condensed to point (7). The condensate (pure water out) may be used for potable application.

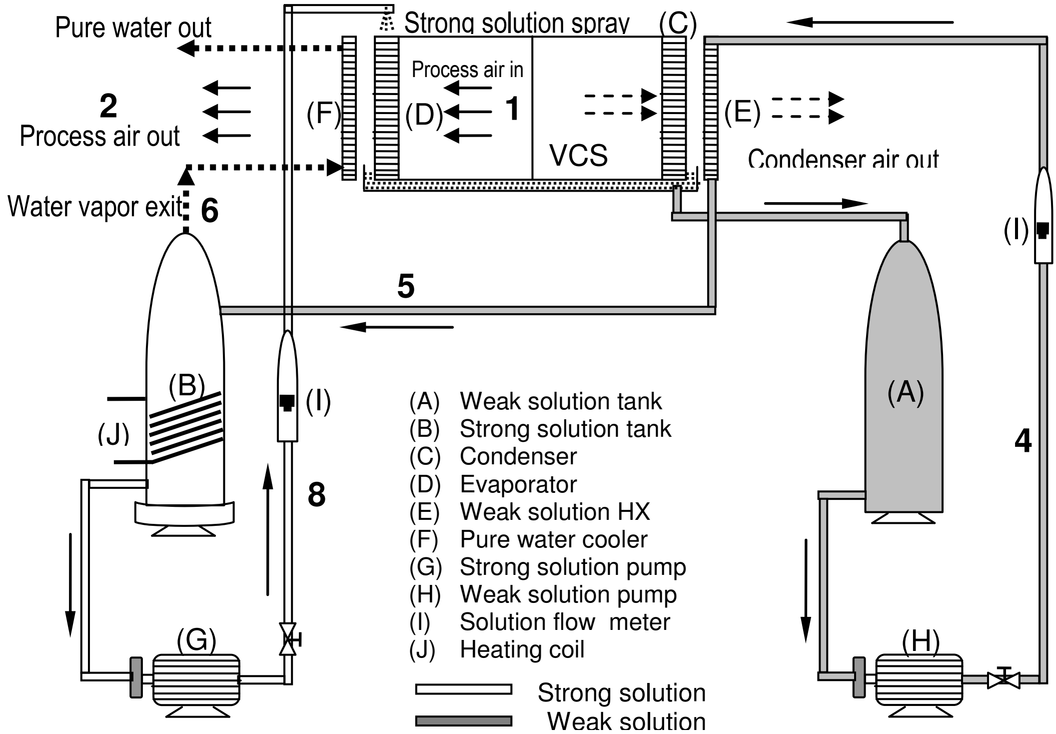

Fig. 1 Schematic diagram of the multi-purpose integrated HDAC system.

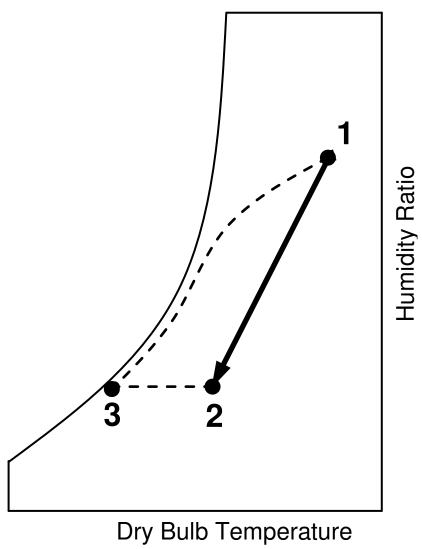

Fig. 2 Typical psychometric process of the proposed HDAC system.

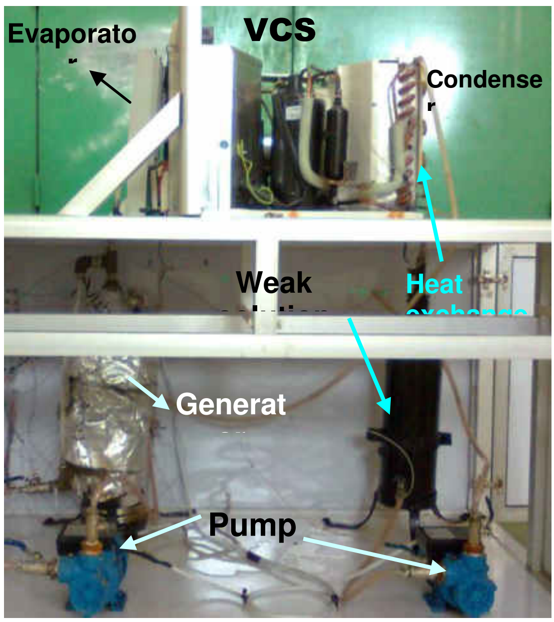

Fig. 3 Photograph of the HDAC system.

3. EXPERIMENTAL SETUP

The experimental set-up of the proposed HDAC system is shown in Fig. 3. The heart of this system is the desiccant solution unit. A vapor compression unit is integrated with the desiccant solution unit. The VCS unit is a window DX type. The unit is 0.75 TR rating with R-134a as refrigerant. The desiccant solution unit uses lithium chloride solution as the working desiccant material. It consists of a weak and a strong solution tanks. A heating coil is wrapped around the strong solution tank (regenerator) and then insulated. Two pumps are used and some control valves to keep the strong and weak solutions flow rates equal.

Air and solution temperatures are measured using type K thermometers and a digital readout device with 0.1°C accuracy. Solution flow rates are measured using rotameters with 2% full scale accuracy. Solution density is measured using a very accurate digital scale (0.01 g accuracy) and is used to determine its concentration from LiCl properties table (at known temperature). Air velocity and humidity are measured using a multi-function hot wire measuring device with accuracy 0.015 m/s for velocity and 3% for relative humidity. The power consumption is measured using a watt meter with 0.005 kW accuracy.

Experimental tests are carried out to evaluate the performance of the proposed multi-purpose HDAC system at different conditions. The following variables are measured: temperature and humidity of the process air at the inlet and exit of the evaporator section, ambient conditions, solution regeneration temperature, solution concentrations (x_4 and x_8) and temperatures, temperature of air leaving condenser as well as evaporator and air velocity for both process and condenser air sides.

4. PERFORMANCE ANALYSIS

The following parameters are used to describe the performance of the proposed system. The proposed system coefficient of performance COP_1 is calculated from

$$COP_1 = m_a(h_1 - h_2)/W \tag{1}$$

where W is the total power consumption.

$$W = W_c + W_{AH} \tag{2}$$

where m_a is the mass flow rate of air, h is the enthalpy of air, W_c is the compressor power consumption, W_{AH} is the auxiliary heat transfer rate for regeneration. Here, one kW of mechanical work energy is equal to three kW of thermal energy. This is because, in general, thermal energy is cheap and easy to get compared to mechanical work. Also this will visualize the potential benefits of the HDAC system especially in laboratory studies (using any available heat source). The VCS system with reheat has the following coefficient of performance

$$COP_2 = m_a\left(h_1 - h_3\right)/\left(W_c + Q_{reheat}\right) \tag{3}$$

$$Q_{reheat} = m_a\left(h_2 - h_3\right) \tag{4}$$

The specific moisture recovery rate SMR of the HDAC system is calculated from

$$SMR = m_a(\Delta W)/ m_d \tag{5}$$

where ΔW is the specific humidity drop of the process air and m_d is the mass flow rate of the desiccant solution. For the proposed HDAC system, the energy saving when using the condenser heat may be calculated from

$$Q_{preheat} = m_d C_{pd} (T_5 - T_4) \tag{6}$$

where C_{pd} is the specific heat of the desiccant solution. The auxiliary heat transfer rate for the generator may be obtained using a hot water coil (using waste heat recovery or solar heater) or any available source and accordingly the regeneration heat can be determined. Note that the absorber heat is removed by the evaporator.

5. RESULTS AND DISCUSSION

An experimental investigation to evaluate the effect of operating parameters on the performance of the proposed HDAC system has been carried out. The results of this experimental study are discussed below. In general, there are some comments that can be drawn from this study. The absorber is an integrated part with the evaporator and, therefore, its heat is removed by the cooling coil. Also, condensing the water vapor is done by the evaporator. These two heat rates may slightly affect the evaporator cooling capacity. But in general, the study showed a considerable improvement in the COP of the HDAC system compared to that of the VCS with reheat, as discussed later. On the other hand, the condenser heat is supplied at relatively lower temperature than that required for the regeneration process. The condenser heat transfer rate is always not enough for regeneration and therefore an auxiliary heat transfer rate is needed.

5.1 effect of regeneration temperature

Figure 4 represents sample temperature records of T_{ev}, T_{amb}, T_{cond} and T_{reg}. They averaged nearly 14, 20, 45 and 65ºC; respectively. Figs. 5, 6 and 7 show the effect of regeneration temperature T_{reg} on the system COP_1, SMR and x_8; respectively. Fig. 5 shows that the COP_1 increases with T_{reg} until it reaches to nearly 65ºC, then it starts to decrease. This may be viewed as follows: the increases in T_{reg} will directly increase the strong solution concentration and hence increasing the SMR as shown in Figs. 6 and 7. As x_8 of desiccant solution increases, the capacity of the solution to absorb moisture from air increases increasing the latent load capacity of the system. And as a result, the COP_1 increases, but for further increase in regeneration temperature, the regeneration heat required at the same desiccant solution flow rate will increase. This will represent a penalty on COP_1. By increasing the regeneration temperature from 65ºC to 80ºC (resulting in 23.1% increase in regeneration heat), the COP_1 decreased by about 17.8% for an air mass flow rate of 0.321 kg/s and a desiccant solution volume flow rate of 2.0 L/min. At the same conditions the SMR and x_8 will increase by about 7.15 % and 28.6%; respectively. Also, from Fig. 5 the COP_1 directly increases with air mass flow rate due to the increase in the total cooling capacity. On the other hand from Fig. 7, by increasing the desiccant solution mass flow rate, the desiccant concentration x_8 is decreased at the same regeneration temperature.

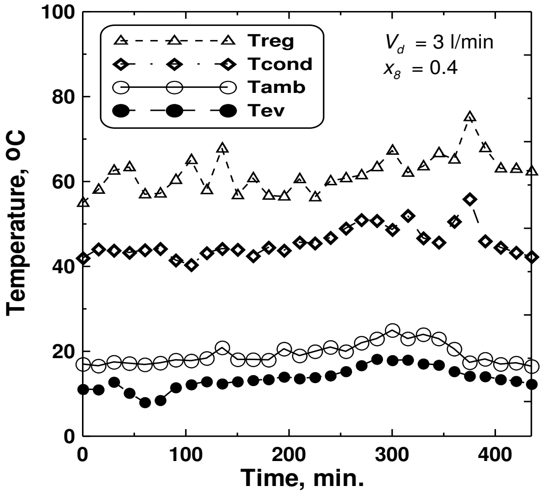

Fig. 4 HDAC temperature records.

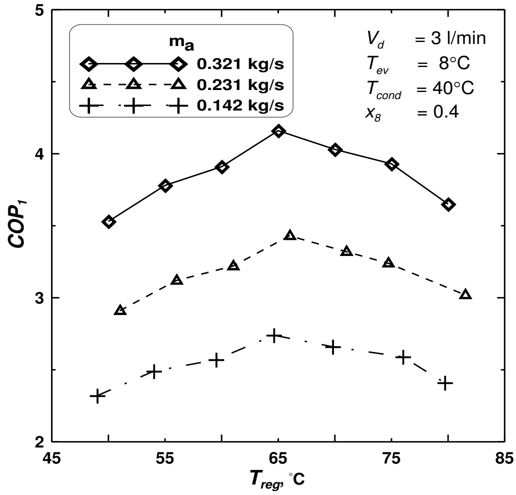

Fig. 5 Effect of Treg on COP1.

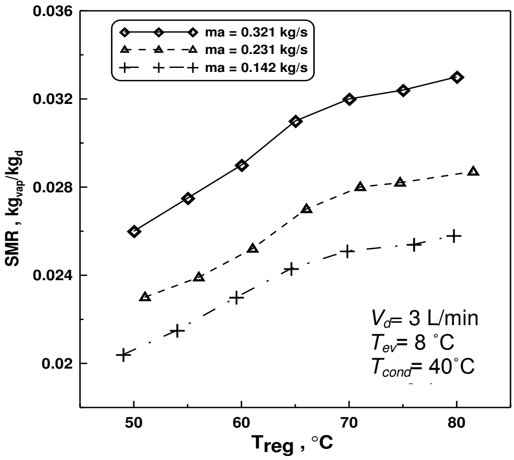

Fig. 6 Effect of T_{reg} on *SMR*.

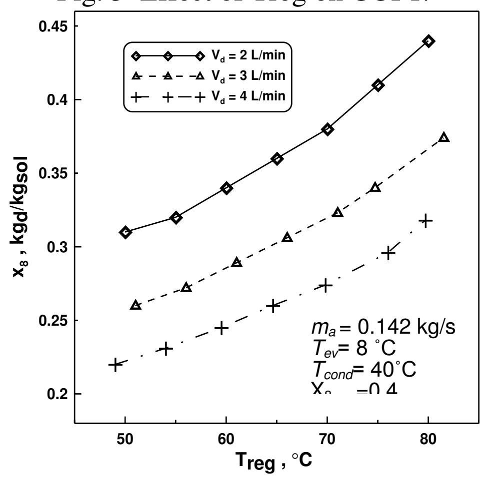

Fig. 7 Effect of T_{reg} on strong sol. Concentration.

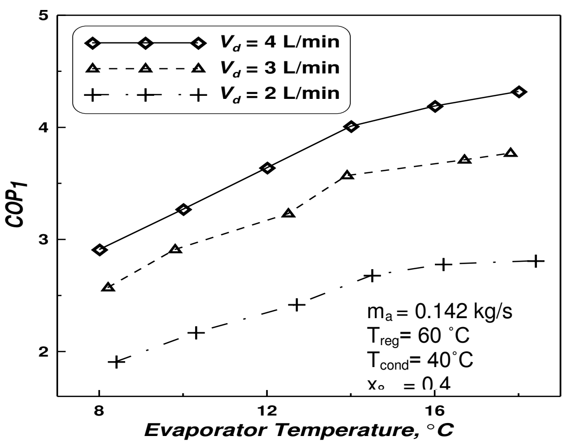

Fig. 8 Effect of evaporator temp. on COP_1.

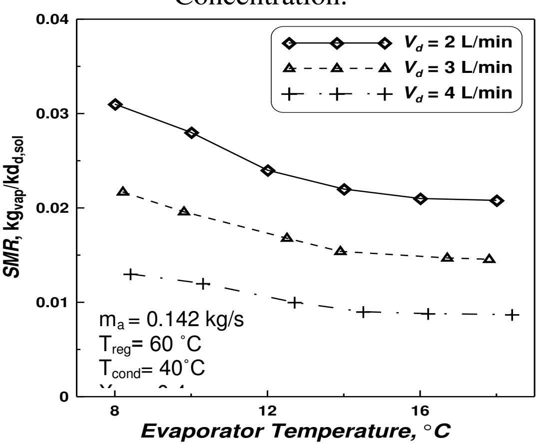

Fig. 9 Effect of evaporator temp. on *SMR*.

5.2 Effect of Evaporator Temperature

Figures 8, 9 and 10 show the effect of evaporator temperature T_{ev} on the HDAC system COP_1, SMR and x_4; respectively. Fig. 8 shows that the COP_1 increases with T_{ev} and with the desiccant solution volume flow rate V_d. By increasing T_{ev} from 8ºC to 18ºC at V_d = 4 l/min, COP_1 will increases by about 41.2%. On the other hand from Fig. 9, by increasing T_{ev} from 8ºC to 18ºC at V_d = 4 l/min, SMR is decreased by about 33.3%. As the evaporator temperature increases, the temperature of the strong solution sprayed over the evaporator will also increase decreasing the solution-air temperature difference. This will reduce the affinity of desiccant solution to recover moisture from process air and hence lowering the SMR. Fig. 10 shows the effect of T_{ev} on weak solution concentration x_4. As T_{ev} increases, the concentration x_4 is directly increased. By increasing T_{ev} from 8ºC to 18ºC at m_a = 0.321 kg/s, x_4 is increased by about 19.8%. This may emphasis the inability of the desiccant solution to catch moisture at higher solution temperature. The COP_1 of the proposed system is found to be 68% greater than that of VCS with reheat as shown in Fig. 11. This may emphasis the potential benefits of using desiccant cooling technology.

Figure 12 shows the energy savings the proposed system vs. T_{reg}. Maximum energy saving is achieved at T_{reg} of about 65ºC and increases as the air flow rate increase. An overall energy saving in the range of 30 – 43% is observed.

5.3 Effect of Condenser Temperature

Figures 13 and 14 show the effect of T_{cond} on the system performance measures COP_1 and SMR; respectively. By increasing T_{cond} , COP_1 will directly decrease. This is because the increase of condenser temperature will increase the compressor power, which represents a penalty on COP_1. On the other hand, SMR increases with T_{cond} until it reaches nearly 45ºC, it starts to decrease. By increasing T_{cond} from 45ºC to 50ºC, the SMR is decreased by about 13.6% at an air mass flow rate of 0.142 kg/s. This is partially referred to that increasing T_{cond} will increase T_{ev} and then reduces the ability of desiccant solution to absorb moisture from the process air.

5.4 Effect of Strong Solution Concentration

Figures 15 and 16 show the effect of strong solution concentration x_8 on the system COP_1 and SMR; respectively. As x_8 increases, the COP_1 and SMR are increased. By increasing x_8, the affinity of desiccant solution to absorb moisture increases, leading to an observed increase in both COP_1 and SMR. An increase of x_8 from 0.3 to 0.42 at an air mass flow rate of 0.321 kg/s will increase the COP_1 and SMR by about 37.85 and 36%; respectively.

5.5 Effect of Desiccant Solution Flow Rate

Figures 17 and 18 show the effect of desiccant solution flow rate V_d on the system SMR and water recovery rate WRR at different air mass flow rates and regeneration temperatures; respectively. From Fig. 17, the SMR reaches to 0.33 g_{vap}/kg_{sol} at V_d of 4 l/min. On the other hand the WRR is nearly about 6.7 l/h.TR at the same conditions.

The curve fitting data for *WRR* at different T_{reg} and V_d is shown below. Note that the ranges of variables for Eqs. (7) and (8) are those found in Fig. 18.

$$WRR = A_0 + A_1 V_d + A_2 V_d^2 \tag{7}$$

$$\begin{aligned} A_0 &= \quad 0.04805142857\ T_{reg} - 4.522047619 \\ A_1 &= -\ 0.03248571429\ T_{reg} + 2.896990476 \\ A_2 &= -\ 0.001085714285\ T_{reg} + 0.3548571428 \end{aligned} \tag{8}$$

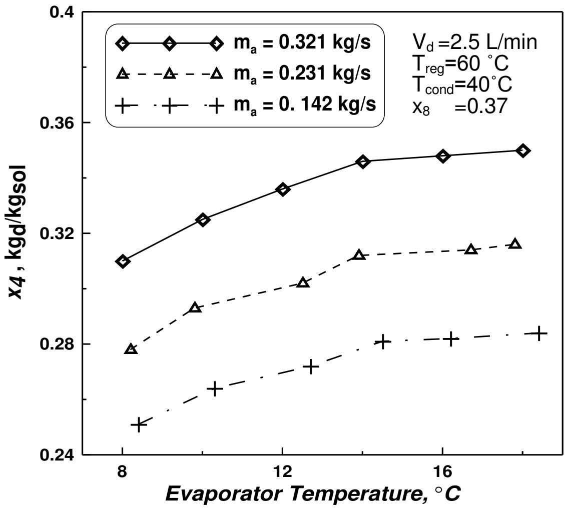

Fig. 10 Effect of T_{ev} on x_4.

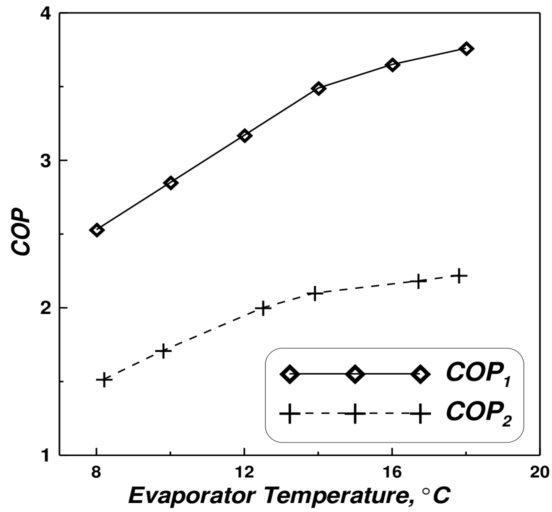

Fig. 11 Effect of evaporator temp. on *COP*.

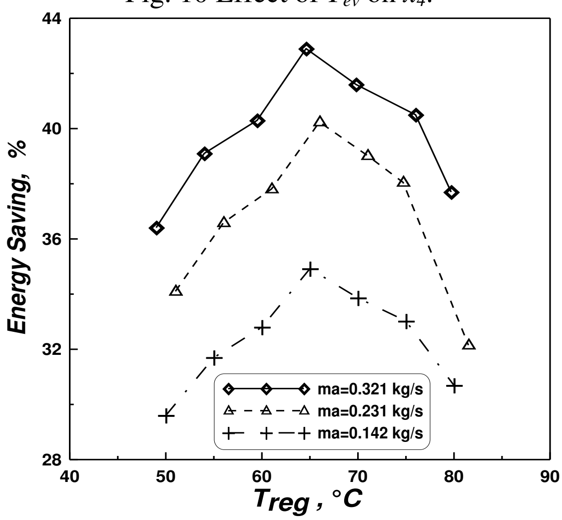

Fig. 12 Energy savings vs. T_{reg}.

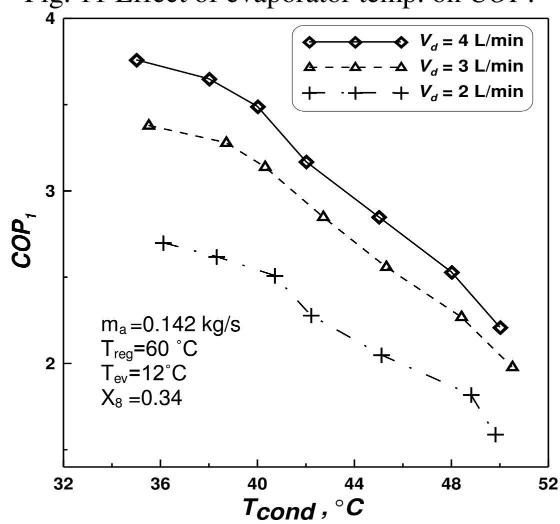

Fig. 13 Effect of T_{cond} on COP_1.

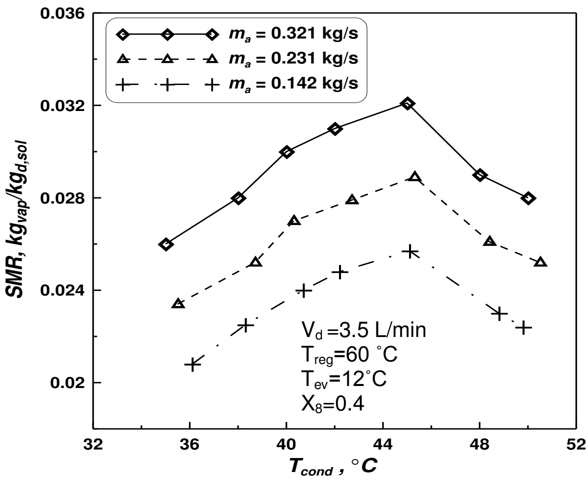

Fig. 14 Effect of T_{cond} on *SMR*.

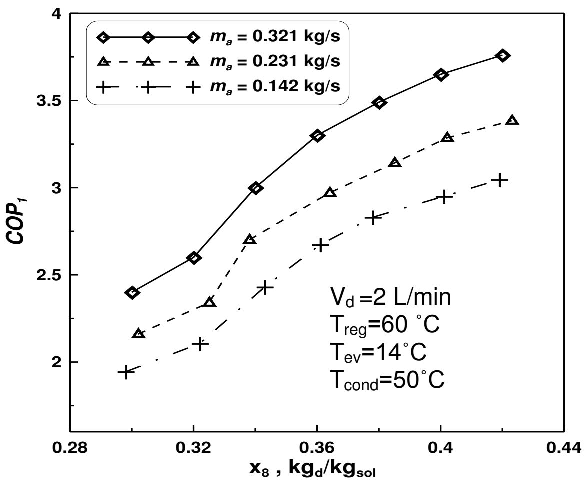

Fig. 15 Effect of x_8 on COP_1.

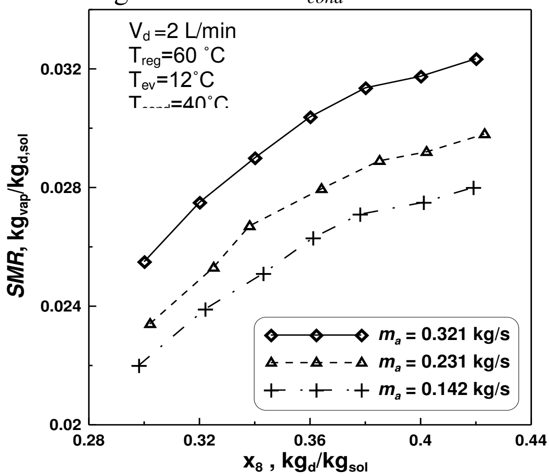

Fig. 16 Effect of x_8 on *SMR*.

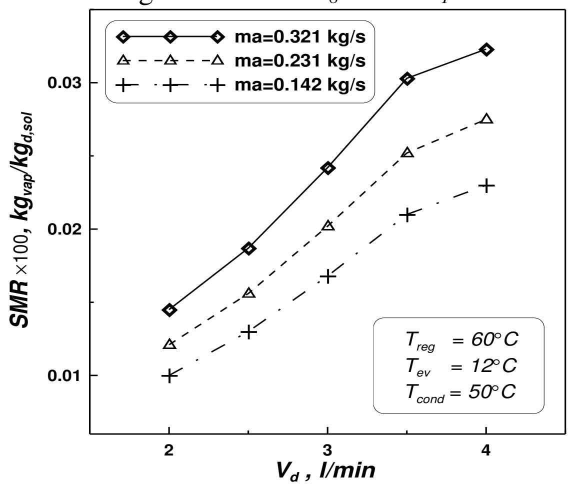

Fig. 17 Effect of V_d on *SMR*.

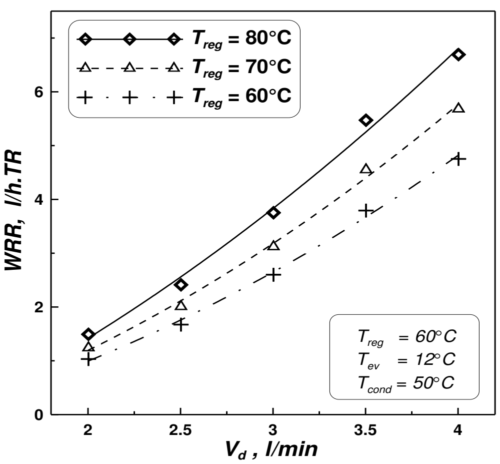

Fig. 18 Effect V_d on *WRR*.

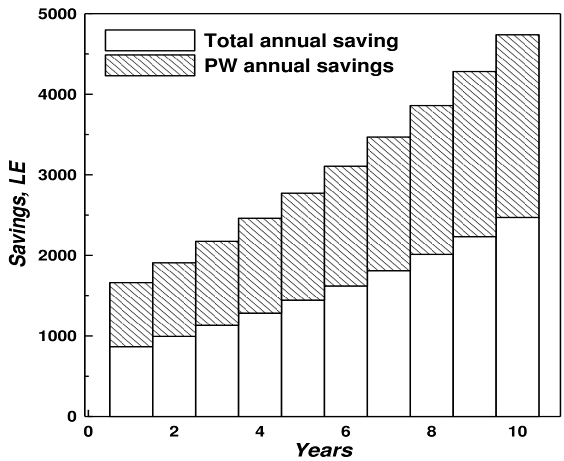

Fig. 19 Annual savings of energy cost and PW for the proposed system.

6. LIFE CYCLE COST ANALYSIS

In this section the present worth value and the payback period are used to compare the proposed HDAC system and VCS system. The life cycle N of the proposed system is assumed 10 years. The yearly loan (mortgage) payment for the proposed HDAC system can be calculated based on the present worth factor PWF as follows [12]:

$$Mortgage\ payment = Capital\ cost / PWF \tag{9}$$

$$PWF = \frac{1}{d-i}\left(1-\left(\frac{1+i}{1+d}\right)^{N}\right) \quad if \quad i \neq d \tag{10a}$$

$$PWF = \frac{N}{1+i} \quad if \ i = d \tag{10b}$$

where i is the energy interest rate and d is the market discount rate. The life cycle savings LCS (LE) may be calculated as follows:

$$LCS = PWF \times \Delta C_{RC} - \Delta C_{IC} \tag{11}$$

where ΔC_{RC} is annual running cost savings and ΔC_{IC} is the initial extra expenditure.

$$\Delta C_{IC} = C_{IC,HDAC} - C_{IC,VCS} \tag{12}$$

$$\Delta C_{RC} = C_{RC,VCS} - C_{RC,HDAC} \tag{13}$$

A simple crude method for getting a quick evaluation of the alternatives is to calculate how long it takes to recover the initial investment, i.e. the payback period PP as follows [13]

$$PP = ln\left(\frac{\Delta C_{IC} \times i}{\Delta C_{RC}} + 1\right) / ln(i+1) \tag{14}$$

The total capital costs of the proposed HDAC and VCS with reheat system have been illustrated with the available material in the Egyptian market in Table 1. In both systems, window type VCS air conditioners are used as reflected in the capital costs below.

Table 1 Capital cost of both systems.

HDAC System		VCS System with Reheat	
	Cost, LE		Cost, LE
DX unit (0.75 TR)	1200	DX unit (1.75 TR)	3600
Lithium Chloride (2 kg)	4000	Reheat coil (added)	800
Manufacturing and accessories	1000	Manufacturing and accessories	450
Overheads	300	Overheads	300
Total capital cost	**6500**	**Total capital cost**	**5150**

The running cost of the HDAC and VCS system are calculated using a thermal energy weighting factor of three, (i.e. 1 kWh of electrical energy = 3 kWh of thermal energy). The price of one electric kWh is assumed to be 0.4 LE in average, see Table 2.

Table 2 Running cost of both systems.

HDAC System			VCS System with Reheat		
	kWh/ day	Annual, LE		kWh/ day	Annual, LE
Regeneration heat (0.58 kW)	5.22	417.6	Reheat coil (2.0 kW)	18.00	1440.0
Compressor and fans (1.14 kW)	10.26	820.8	Compressor and fans (1.95 kW)	17.55	1404.0
Maintenance		200.0	Maintenance		200.0
Total annual running cost	**15.48**	**1438.4**	**Total annual running cost**	**40.86**	**3044**

The difference in capital cost ΔC_{IC} is 1350 LE as shown in Table 1 and the difference in annual running cost ΔC_{RC} is 1605.6 LE as shown in Table 2. The capital cost of the hybrid system is financed over 10 years at market discount rate d = 9% and the annual cost payments are expected to inflate at a rate of i = 8%. From Eq. (15) the mortgage payment is equal to 6500/8.8 = 738 LE. Also, the payback time is found to be 0.84 years (10 months). The payback time of the proposed system is less than one year and it will achieve an annual savings in the total running cost of about 53% compared to VCS with reheat. The total annual saving of the HDAC system reaches 2470 LE at the end of its life time. Due to the market discount rate, the annual present worth value is expected to be 2260 LE compared to the conventional VCS with reheat system. Fig. 20 shows a diagram for the annual total savings and the present worth savings. Note: the total capital cost of the HDAC system as shown in Table 1 is almost the same as that of the VCS with reheat that uses an air conditioner of the conventional split type.

7. CONCLUSIONS

A multi-purpose HDAC integrated vapor compression system of a small capacity is designed and experimentally tested. At specific design and operating conditions, some important conclusions can be summarized as follows:

- The COP_1 and *SMR* are both increased with air mass flow rate and desiccant flow rate.
- An increase of strong solution concentration will increase the COP_1 and *SMR*.
- By increasing T_{ev}; the COP_1 increases but *SMR* decreases.
- By increasing the T_{reg}; the COP_1 decreases and *SMR* increases.
- Results showed that an amount of 6.7 l/h.TR of pure water can be collected at specific design and operating conditions.
- An imperical relation for the *WRR* with V_d and T_{reg} is withdrawn.
- The COP_1 of the proposed system is found to be 68% greater than that of *VCS* with reheat at typical operating conditions.
- The HDAC system can achieve an annual energy savings of 53% and has apayback period of 10 months compared to the *VCS* with reheat.
- A HDAC system integrated with a 0.75-TR conventional air conditioner can replace a VCS with reheat with a cooling capacity of 1.75 TR in addition to a reheat coil of 2.0 kW rating.

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دراسة عملية لنظام تكييف هواء متعدد الأغراض يعمل بالمواد المازة مدمج مع دورة إنضغاط البخار

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يتناول هذ البحث دراسة عملية لنظام تكييف هواء هجين متعدد الأغراض يعمل بالمواد المازة السائلة المدمجة مع دورة إنضغاط البخار. تم تصميم المنظومة – المسماة إختصارا HDAC – لتلبية متطلبات أحمال التبريد لتكييف هواء فراغات ذات حمل كامن عالى نسبيا بالإضافة إلى إستخلاص الماء العذب من الهواء الجوى. تتكون هذه المنظومة من وحدة تجفيف تعمل بالمواد المازة السائلة مدمجة مع دورة تبريد بإنضغاط البخار. تعمل وحدة التجفيف بمحلول كلوريد الليثيوم. تم دراسة تأثير العوامل المختلفة على أداء المنظومة، مثل معدل سريان محلول كلوريد الليثيوم، ومعدل سريان الهواء المكيف، ودرجات حرارة كل من المبخر والمكثف، وتركيز المحلول المركز ودرجة حرارة عملية التنشيط.

يمكن لمنظومة HDAC إستخلاص 6.7 لتر/ساعة-طن تبريد من الماء العذب من الهواء الجوى. وحققت المنظومة معامل أداء يصل إلى 3.8 بزيادة مقدارها 68% عن دورة إنضغاط البخار التقليدية. إستطاعت المنظومة تغطية حمل تبريد مقداره 1.75 طن تبريد بإستخدام دورة إنضغاط البخار قدرتها الإسمية 0.75 طن تبريد. وأخيرا فإن المنظومة بما تحققه من وفر فى الطاقة تستطيع إسترداد تكلفتها الأولية خلال فترة عشرة شهور وبدون أى تكلفة أولية تزيد عن تلك المطلوبة لجهاز مماثل من نوع الوحدات المنفصلة المعروف. كما تؤكد النتائج على الفوائد الجمة المتوقعة من منظومة HDAC .