Journal of Engineering Research

Volume 5 | Issue 1 Article 2

2021

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Vol. 5, No.1 – 2021 Journal of Engineering Research (ERJ)

Effect of Operational and Design Parameters on Desiccant-Assisted Hybrid Air-conditioning Systems Performance

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***Abstract*- Energy savings are a major goal in our lives because energy consumption is continuously increasing. We have studied in this paper four air conditioning systems, so that each system mainly consists of a vapor compression cycle and a rotating desiccant wheel. The thermodynamic analysis for air conditioning system, the heat exchanger, the ground heat exchanger and the evaporative water spray cooler were presented. Design and operating parameters including outlet air temperature, outlet air humidity, regeneration air temperature, and desiccant wheel speed are studied to assess its effect on the performance of the systems considered. This hybrid system significantly decreases the supplied air temperature at different four Systems. When the inlet air humidity ratio and the regeneration air temperature increase, the COP_{th} of the air conditioning system decreases. When the desiccant wheel speed increased, the COP_{th} of the air conditioning system decreases. COP_{th} decreases with the increasing of the cooling load for the space and also decreases with the decreasing of the inlet air humidity ratio at process air. The results show that in these four systems, COP_{th} is the highest at system 4 which contains desiccant wheel, heat exchanger and ground source circulation achieving an improvement of about 48 % and a consequent power consumption reduction of 20 %.**

***Key words*- Hybrid air conditional, Ground source circulation, Desiccant wheel, COP_{th}.**

1. INTRODUCTION

As a result of climate change and global warming, air conditioning demand is growing rapidly. The continued use of traditional electrical air conditioning with electricity produced from combustion of fossil fuels would increase the mission of greenhouse gas and deeply intensify global warming. In exchange, there will be a further rise in air conditioning demand. Air conditioning in subtropical towns is a common provision for houses. Air conditioning, however, typically consumes half the energy consumption of buildings. Climate change is becoming increasingly evident. The most effective in desiccant air conditioning development is the desiccant material. The desiccant material used significantly influences the efficiency of the air conditioning systems [1]. Common materials of drying include activated carbon, aluminum activated, molecular sieve, silica gel, lithium chloride, calcium chloride, etc. Molecular sieve, silica gel, lithium chloride, calcium chloride, the selection of suitable desiccant materials has two principles: (1) a large-scale saturated adsorption capacity material should be available for the desiccant; (2) a desiccant material of type 1M should approach the adsorption output in the best possible way. Pennington [2], the first rotary desiccant air conditioning cycle patent has been introduced. A modified ventilation cycle is used the process ambient air for the building and ambient air for regeneration. It is obvious that, in relation to the standard ventilation cycle, thermal performances including thermal coefficient performance and special cooling capacity are decreased because both the humidity ratio and ambient air temperature are typically higher than return air. The recirculation cycle is a Pennington cycle modification, which reuse return air as process air. In this step, ambient air is used for regeneration. The humidity ratio and temperature are therefore relatively low, and that cycle's thermal COP is only 0.8[3]. There is a shortage of fresh air as a main disadvantage of the recirculation cycle. The Dunkle Cycle [4] combines the merit of the ventilation cycle with the recirculation cycle, allowing for a relatively low cold air temperature for the HE at the ventilation cycle, and providing a relatively large amount of cooling capacity for the conditioned room. The is a secondary heat exchanger. The Dunkle cycle is limited by the lack of fresh air, as with the recirculation cycle. Many cooling loads don’t need the system source for outdoor air. Fresh air should be maintained at an acceptable level to ensure favorable system performance and good indoor air quality. Implementation by Maclaine-cross [5], the introduction of SENS cycle is a simplified advanced solid desiccant cycle. A two-stage solid desiccant air conditioning system integrated with a HVAC system has been introduced by Meckler [6]. He used an enthalpy exchanger for precooling the process air by exchanging sensible and latent heat for return air without adding external heat or regeneration. Consequently, the conventional desiccant wheel is used with external heat to dehumidify air further. The enthalpy exchanger could perform around 30–50% of the dehumidification task. Henning [7] has also introduced this kind of two-stage system. In two stage desiccant cooling method, Zhang and Niu [8] discussed the use of low regenerative temperatures. Simulation results demonstrated that a lower temperature for regeneration than for a single stage cooling desiccant system was required. Mazzei et al. [9] have examined many Possible theatre and supermarket AHU configurations and HVAC systems. They compared the conventional VAC system and the hybrid systems advantages. Important operating cost savings and increased plant cost (simple 2-year payback period for supermarket), decreased energy demand and improved ambient humidity controls. Ghali et al. [10] have simulated the hybrid desiccant VAC system's transient performance in Beirut climatic conditions. The economic feasibility of combining a rotating solid desiccant wheel with the traditional VCC within the city of Beirut was studied. The combined system uses condenser and natural gas heat to regenerate the desiccant wheel to improve the economic practicability of a hybrid system. Dai. et al. [11] introduced a hybrid desiccant dehumidifier, evaporative cooling, and vapor compression air conditioning. In comparison with the vapor compression alone, the test showed that the cooling production and COP in the new hybrid system significantly increased. They showed a ± 20-30% increase in the cooling production. The energy consumption and the scale of the steam compression cycle can also be decreased. A desiccant wheel integrated vapor compression air conditioning system was developed by Subramanian et al. [12]. In order to estimate its performance in comparison with traditional and reheating systems, they studied the various supply airflow rates. The dehumidification of the air supply is increased by the desiccant wheel, which makes air conditioning less humidified. The dew point temperature of supply air in comparison with conventional systems are around 2 °C lower and the COP marginally lower by about 5%. However, the desiccant wheel system COP almost doubles the reheat system. Jia et al. [13] introduced the possibility to use a hybrid desiccant-assisted air conditions and split cooling coil, which combines the advantages of removing moisture via the desiccant and cooling coil for sensible heat removal. Experiments have been performed on a hybrid desiccant air conditioning system that combines rotary solid desiccant system and VCC. The hybrid desiccant cooling system helps in saving electricity power by 37,5 percent while process temperature and relative humidity are maintained at 30 ° C and 55 percent respectively in comparison to conventional VCC. A thermal analysis of air conditioning system was introduced by ElAgouz et al. [14] via its various components: desiccant wheel, solar collector, heat exchanger, ground heat exchanger and water spray evaporative cooler. The study simulates 3 different air conditioning cycles for different zones such as: hot-dry, warm-dry, hot-humid and warm-humid zones. The results show that the desiccant air conditioning system gives a greater thermal comfort in different climates. Sultan et al. [15] they discussed sensible and latent load of AC required for various nonhuman AC applications. They represented ideal temperature and humidity zones, the efficient AC systems are proposed and discussed for the subjected applications. In addition, thermodynamic limitation of VAC system and scope of proposed systems is also highlighted.

The performance evaluation of two kinds of crosslinked hydrophilic organic polymer sorbents (PS-I and PS-II) for desiccant air-conditioning applications by Sultan et al. [16], Optimum temperature and humidity zones are established for various air-conditioning applications which include (i) humans' thermal comfort, (ii) animals' thermal comfort, and (iii) postharvest storage of fruits/vegetables. The potential selection of efficient air-conditioning (AC) and cooling systems in order to avoid excess power consumption, mitigation of harmful refrigerants generated by the existing AC systems by Kashif et al. [17]. An evaporative–vapor compression based combined air conditioning system for providing required human comfort conditions at comparatively low cost has been presented by Chauhan [18]. Five kinds of adsorbents for desiccant air-conditioning (DAC) applications are investigated by Sultan et al [19]. Each adsorbent yield distinctive water vapor adsorption isotherm that can be categorized as type-I, type-II, type-III, type-V, and type-linear on the basis of the International Union of Pure and Applied Chemistry (IUPAC) classification. A lab-scale open-cycle solid-silica-gel-based DAC experimental apparatus was developed by Duan et al. [20]. Introduced desiccant based air-conditioning (DAC) options for livestock's thermal comfort by Niaz, et al. [21]. Mahmood et al. [22] introduced the experimentally investigates desiccant dehumidification and indirect evaporative cooling for agricultural products' storage, thermodynamic advantages of the proposed system were highlighted and compared to vapor compression systems. Noor et al. [23] introduced comprehensive details of evaporative cooling options for building air‑conditioning (AC) in Multan (Pakistan). A solid-silica-gel-based DAC system was developed, at lab-scale, for the performance evaluation of the DAC system by Sultan [24].

In this paper we present four air conditioning systems, each system consists mainly of vapor compression cycle and rotating desiccant wheel:

1- Central air conditioning system with desiccant wheel.

2- Central air conditioning system with desiccant wheel and heat exchanger.

3- Central air conditioning system with desiccant wheel and indirect evaporative cooler.

4 - Central air conditioning system with desiccant wheel, heat exchanger and ground source circulation.

We analyze the impact of important operating parameters on the performance of the systems considered, including outlet air temperature, outlet air humidity, regeneration air temperature, and desiccant wheel speed.

2. Systems Description

In the first system (system 1), as shown in Fig. 1, two air streams, air process (points 1–R) and air regeneration (points 3–5), is involved in the desiccant cooling process. The hybrid air conditioning system consists of the desiccant wheel, vapor compression cycle, and auxiliary air heater. The process airstream and the regeneration air stream are prepared, thorough ambient air. The process air (1) is dried in the desiccant wheel first. By passing it through the cooling coil, the dry air (2) is cooled. An auxiliary heater heats the regeneration air (3). The warm regeneration air (5) carries moisture from the desiccant wheel to the surroundings. This process regenerates the desiccant wheel. The second system (system 2), is schematically illustrated Fig. 2. It consists of the desiccant wheel, heat exchanger, vapor compression cycle, and auxiliary air heater. After the process air is dried in the desiccant wheel (2), it is precooled by passing it through the heat exchanger (3). The precooled air at (3) is cooled by cooling coil. The regeneration air (4) is preheated in heat exchanger. The preheated air (6) is heated at auxiliary heater. The warm regeneration air (7) carries moisture from desiccant wheel to the surroundings. In the third system (system 3), the hybrid air conditioning system consists of the desiccant wheel, heat exchanger, indirect evaporative cooler, vapor compression cycle, and auxiliary air heater, the process air is dried in the desiccant wheel (2), and it is precooled by passing it through heat exchanger (3), and by passing indirect evaporative cooler (4). The scheme of the hybrid air conditioning system is shown in Fig. 3(a). The thermodynamic process is illustrated on the psychometric chart in Fig. 3(b). The precooled air (4) is cooled by passing in cooling coil. The regeneration air is passed the same process of that in system 2. In the fourth system (system 4), the hybrid air conditioning system consists of the desiccant wheel, heat exchanger, ground source circulation, vapor compression cycle, and auxiliary air heater. The scheme of the hybrid air conditioning system is shown in Fig. 4(a). The thermodynamic process is illustrated on the psychometric chart in Fig. 4(b), the process air is passed desiccant wheel to dry and heat exchanger as the system 3 but after heat exchanger the process air is passed in ground source circulation (4). Thereafter, the precooled air (4) is cooled by passing in cooling coil. The regeneration air is passed through the same process of system 2 and system 3.

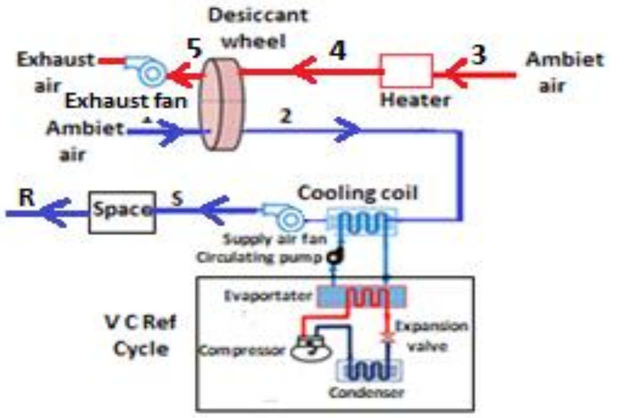

Fig. 1 Schematic of central air conditioning system with desiccant wheel (system 1).

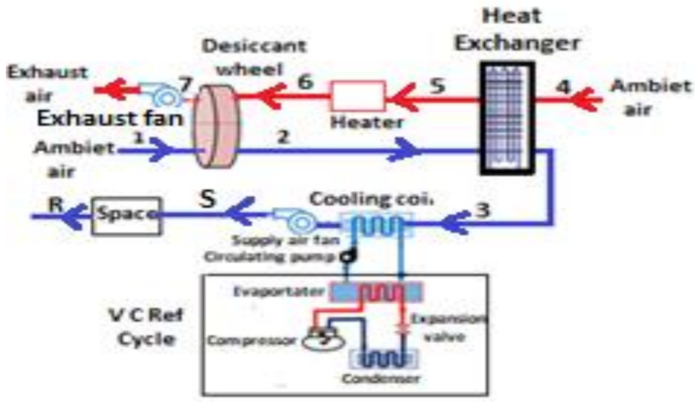

Fig. 2 Schematic of central air conditioning system with desiccant wheel / heat exchanger (system 2).

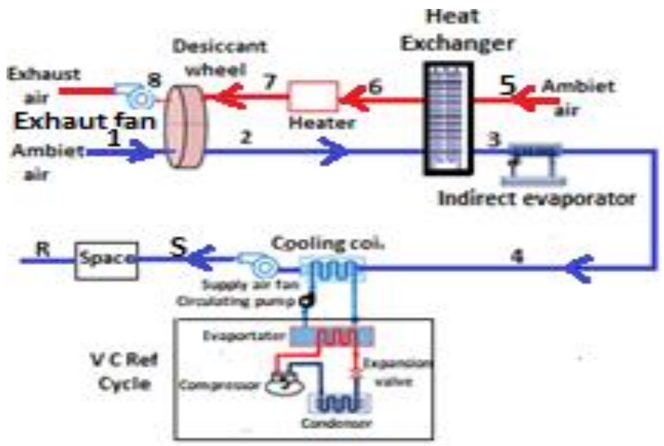

Fig. 3(a)

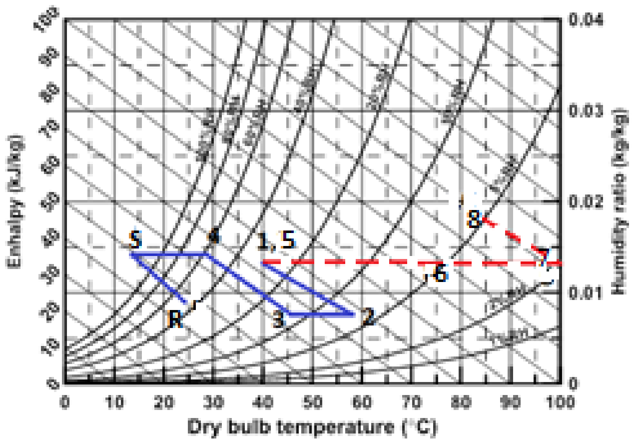

Fig. 3(b)

Fig. 3 Central air conditioning system with desiccant wheel/heat exchanger and indirect evaporative cooler (system 3). (a) schematic of sytem, (b) processes of both main air and regeneration air on psychrometric chart.

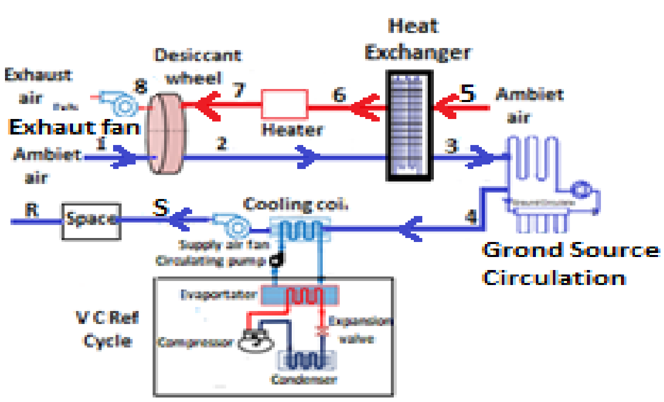

Fig. 4(a)

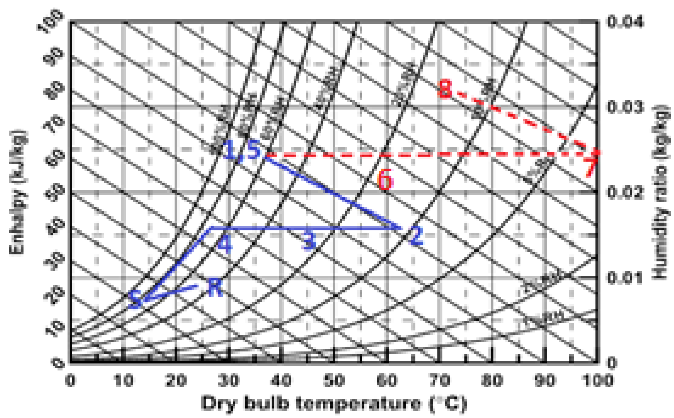

Fig. 4(b)

Fig. 4 Central air conditioning system with desiccant wheel / heat exchanger and ground source circulation (system 4). (a) schematic of system, (b) processes of both main air and regeneration air on psychrometric chart.

Table 1. The Operating Parameters

Parameter	$T_{air,in}$	$w_{air,in}$	T_{Reg}	Desiccant Wheel
	30-50 °C	20,24 g.kg^{-1}	70-120 °C	10-30 RPH

3. Governing Equations

3.1. The process air

The temperature and humidity of outlet air from the desiccant wheel is measured using the simulation software of desiccant wheel. With negligible air leakage and heat transfer to the outside, the ideal heat exchanger system is used. The effectiveness is defined as:

$$\varepsilon_{HE} = \frac{T_{HE,ci} - T_{HE,co}}{T_{HE,ci} - T_{HE,hi}} \quad \textbf{(1)}$$

The heat exchanger outlet air temperature at process air path is:

$$T_{HE,co} = T_{HE,ci} - \boldsymbol{\varepsilon_{HE}}\left(T_{HE,ci} - T_{HE,co}\right) \quad \textbf{(2)}$$

Evaporative coolers are normally rated according to their saturation effectiveness, as described by the Fouda and Melikyan equation [25]

$$\varepsilon_{EC} = \frac{T_{EC,ci} - T_{EC,co}}{T_{EC,ci} - T_{EC,wbi}} \quad \textbf{(3)}$$

Therefore, the evaporative coolers outlet air temperature is:

$$T_{EC,co} = T_{EC,ci} - \boldsymbol{\varepsilon_{EC}}\left(T_{EC,ci} - T_{EC,wbi}\right) \quad \textbf{(4)}$$

The process inlet and outlet evaporative cooler is constant enthalpy process.

A cooling coil unit with ground circulation was used in system 4. Variations in the air temperature and the solar radiation influence the ground temperature. The undisturbed ground temperature fluctuates daily and annually under the influence of these effects. A sinusoidal function Kusuda and Archenbach [26] can be used to estimate annual surface ground temperature variation. The undisturbed temperature of the ground is changes to a depth of 10 m, after which the temperature of the ground becomes constant Khalajzadeh et al. [27]. The cooling coil unit is designed to cool the air supply stream after the heat exchanger. With negligible air leakage and heat transfer to the outside, the effectiveness of ground source circulation is defined as:

$$\varepsilon_{GSC} = \frac{T_{GSC,ci} - T_{GSC,co}}{T_{GSC,ci} - T_{wi}} \quad \textbf{(5)}$$

The outlet air temperature from ground source circulation is:

$$T_{GSC,co} = T_{GSC,ci} - \boldsymbol{\varepsilon_{GSC}}\left(T_{GSC,ci} - T_{w,i}\right) \quad \textbf{(6)}$$

3.2. The regeneration air

The energy balance between the two air streams is written in system 2, using the following equation.

$$h_{HE,ci} - h_{HE,co} = h_{HE,ho} - h_{HE,hi} \quad \textbf{(7)}$$

The humidity ratio is constant at inlet and outlet heat exchanger in both process and regeneration air passes since the leakage is negligible. With the above equation, the states at outlet air in two paths were determined and the temperature and humidity ratio in inlet air in two paths are known.

3.3 Cycle performance

The cooling system's thermal Coefficient of performance, COP_{th}, was defined by the ratio of cooling load system to the sum of regeneration heat and the input of electrical power to vapor compression cycle.

$$COP_{th} = \frac{Q_l}{Q_{Reg} + P_{in}} \quad \textbf{(8)}$$

The regeneration energy gain by the air was determined from:

$$Q_{Reg} = \dot{m}_{Reg}\left(h_{Reg} - h_{hi}\right) \quad \textbf{(9)}$$

Table 2. Proceeding of system

SYSTEMS	DESICCANT WHEEL	HEAT EXCHANGER	INDIRECT EVAPORATIVE COOLER	GROND SOURCE CIRCULATION	COOLING COIL	SPACE	HEATER
System 1	$T_1 = T_\infty$ Use desiccant wheel simulation software to obtain T2	--------------------	------------------	------------------	$Q_{cc} = \dot{m}_a(h_2 - h_s)$	$Q_l = \dot{m}_a(h_R - h_s)$ Q_l was calculated from the space conditioned	$Q_h = \dot{m}_a(h_4 - h_3)$
System 2		$T_3 = T_2 - \varepsilon_{HE}(T_2 - T_4)$ $h_2 - h_3 = h_5 - h_4$	------------------	------------------	$\boldsymbol{Q_{cc} = \dot{m}_a(h_3 - h_s)}$		$Q_h = \dot{m}_a(h_5 - h_4)$
System 3		$T_3 = T_2 - \varepsilon_{HE}(T_2 - T_5)$ $h_2 - h_3 = h_6 - h_5$	$T_4 = T_3 - \varepsilon_{EC}(T_3 - T_{wb3})$	------------------	$\boldsymbol{Q_{cc} = \dot{m}_a(h_4 - h_s)}$		$Q_h = \dot{m}_a(h_6 - h_5)$
System 4		$T_3 = T_2 - \varepsilon_{HE}(T_2 - T_5)$ $h_2 - h_3 = h_6 - h_5$	------------------	$T_4 = T_3 - \varepsilon_{GSC}(T_3 - T_{w,in})$	$\boldsymbol{Q_{cc} = \dot{m}_a(h_4 - h_s)}$		$Q_h = \dot{m}_a(h_6 - h_5)$

P_{in} is the input power to run the compressor of vapor compression cycle. Table (3) shows comparison of the systems. Within Eq. (8), only thermal COP_{th} is considered. In the current study the energy required for circulator pump and ventilator fans are not taken into account. This assumption makes the results independent of the distribution system.

4. Case Study

A case study was done to investigate the performance of air conditioning systems for Central Library, Kafr El Sheikh University. Data for operating conditions were gathered through file sheet of system for analysis of the system. The cooling load for the building is estimated using 'HAP' (Hourly Analysis Program) that estimates design cooling and heating loads for this building in order to determine required sizes for HVAC system components. Ultimately, the program provides information needed for selecting and specifying equipment. The data from HAP was fed into the 'Desiccant wheel simulation' software to select the suitable desiccant unit required for the system. The desiccant units considered for the system are WSG3050X200. The unit is chosen according to the amount of air entering to the desiccant wheel. Table 4 shows the cooling and heating loads part of the building.

Table 3. Comparison of the Systems

Systems	Q_l	Q_{reg}	P_{in}
System 1	$Q_l = \dot{m}_a (h_R - h_s)$	$Q_{Reg} = \dot{m}_{Reg}(h_4 - h_3)$	By knowing cooling coil for each system, the input power for compressor was obtained from 30XW carrier catalog
System 2		$Q_{Reg} = \dot{m}_{Reg}(h_6 - h_5)$	
System 3		$Q_{Reg} = \dot{m}_{Reg}(h_7 - h_6)$	
System 4		$Q_{Reg} = \dot{m}_{Reg}(h_7 - h_6)$	

5. Numerical Procedures

The computer program was used to solve the governing equations mentioned above. A desiccant wheel simulation software was used for the analysis of desiccant cooling cycles, by knowing inlet conditions in points 1 and 6 and also the wheel properties, the simulation of the desiccant wheel can be performed. In the airstream process the simulation continues by calculating points 3, and 4 with regard to the equations described in Section 3.1. The outdoor design condition at point 4 is identified in the regeneration air stream and the heated air at the point 5 can be measured.

Table 4. The cooling and heating loads part of the building

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Aug 1500 COOLING OA DB / WB 98.1 °F / 68.8 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 37.8 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	2760 ft²	64039	-	2760 ft²	-	-
Wall Transmission	2880 ft²	13336	-	2880 ft²	16231	-
Roof Transmission	15600 ft²	144494	-	15600 ft²	47021	-
Window Transmission	2760 ft²	30421	-	2760 ft²	40572	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	110 ft²	619	-	110 ft²	825	-
Floor Transmission	15600 ft²	0	-	15600 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	18701 W	56431	-	0	0	-
Task Lighting	31200 W	99615	-	0	0	-
Electric Equipment	31200 W	100963	-	0	0	-
People	1560	327000	319800	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	836918	319800	-	104649	0
Zone Conditioning	-	822185	319800	-	109209	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	46138 CFM	0	-	46138 CFM	0	-
Ventilation Load	23400 CFM	527368	-319736	23400 CFM	625547	0
Supply Fan Load	46138 CFM	0	-	46138 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	1349553	64	-	734756	0
Central Cooling Coil	-	1349553	0	-	0	0
Central Heating Coil	-	0	-	-	734756	-
>> Total Conditioning	-	1349553	0	-	734756	0
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

6. Results and discussion

In this study, wheel speed = (10, 15, 20, 25 and 30 RPH), wheel diameter = 3050 mm, and wheel depth=200 mm are the design parameters of desiccant wheel. Ground temperature is assumed to be constant at $T_{w,\,in}$ = 20 °C and $\varepsilon_{HE} = \varepsilon_{GSC} = \varepsilon_{EC}$ = 0.7 are the effectiveness of heat exchanger, cooling coil unit, and evaporative cooler are. The mass flow rate is the same for processing and regeneration air.

Fig. 5 shows the variation of the desiccant wheel's outlet air temperature and outlet air humidity ratio due to changes inlet conditions. The speed of the desiccant wheel is kept constant at 25 RPH and constant regeneration air temperature at 100 °C. The outlet air temperature and the outlet air humidity ratio of the desiccant wheel in the process air stream increases as the inlet air temperature increases at all considered systems. With a constant inlet air temperature and high inlet air humidity ratio, the process leaves the desiccant wheel at point 2 at high temperature and contains more humidity. At the inlet air humidity ratio w_1= 24 g_w kg_{da}^{-1}, the outlet air temperature from the desiccant wheel in the process air stream is higher. At inlet air humidity ratio w_1= 20 g_w kg_{da}^{-1} the outlet air humidity ratio from the desiccant wheel in the process air stream is lower.

Fig. 6 shows the variation of desiccant wheel's outlet air temperature and outlet air humidity ratio due to changes in the regeneration air temperature and inlet air humidity ratio. The speed of desiccant wheel is kept constant at 25 RPH and inlet air temperature is constant at 40 °C. When the temperature of regeneration air increases, the temperature of outlet air from the desiccant wheel in the process air stream increases at point 2; on the other hand, the outlet air humidity ratio from desiccant wheel in the process air decreases at all considered system. This would seem to be expected as the effectiveness of desiccant wheel increases as the regeneration temperature increase. At the inlet air humidity ratio w_1= 24 g_w kg_{da}^{-1}, the outlet air temperature from the desiccant wheel in the process air stream is higher. At the inlet air humidity ratio w_1= 20 g_w kg_{da}^{-1}, the outlet air temperature from the desiccant wheel in the process air stream is lower.

Fig. 7 shows COP_{th} values for a space condition of all considered systems with different wheel speed and different inlet air humidity ratio of the inlet air at a constant inlet air temperature of 40 °C and a constant regeneration temperature of 100 °C. With the increase of the wheel speed COP_{th} increases. COP_{th} is higher at $w_{in,air}$ = 20 g_w kg_{da}^{-1} at all systems considered. At system 4 which contains desiccant wheel, heat exchanger and ground source circulation, the COP_{th} is the highest.

Fig. 8 shows COP_{th} values for a space condition of all considered systems with different inlet air temperature and different inlet air humidity ratio at the constant regeneration temperature at 100 °C and wheel speed at 25 RPH. With the increase of the inlet air temperature, COP_{th} increases. COP_{th} is higher at $w_{in,air}$ = 20 g_w kg_{da}^{-1} at all considered systems.

Fig. 9 shows also, COP_{th} values for a space condition of all considered systems with different regeneration temperature and different inlet air humidity ratio at the constant inlet air temperature of 40 °C and wheel speed of 25 RPH. With the increase of the regeneration temperature, COP_{th} decreases. COP_{th} is higher at $w_{in,air}$ = 20 g_w kg_{da}^{-1} at all considered systems.

Fig. 10 shows COP_{th} values for a space condition of all considered systems with different space cooling loads and different inlet air humidity ratio, and the constant inlet air temperature at 40 °C and the constant wheel speed at 25 RPH. As shown in Fig. 10 COP_{th} decreases when the cooling load increases. Among the systems considered, system 4, which contains desiccant wheel, heat exchanger and ground source circulation, possess the highest COP_{th} with the variation of design parameters and operational parameters, as noticed from fig. 7-10.

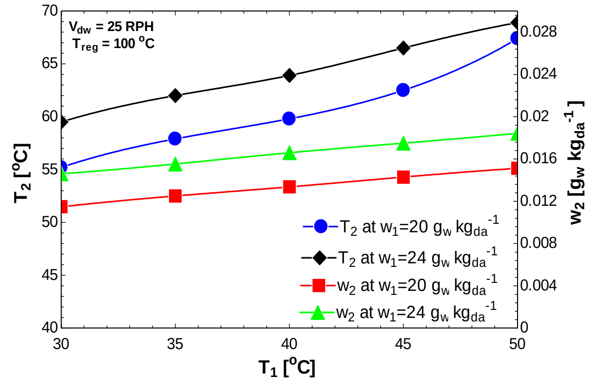

Fig. 5 The effect of inlet air conditions, on the outlet air conditions from desiccant wheel.

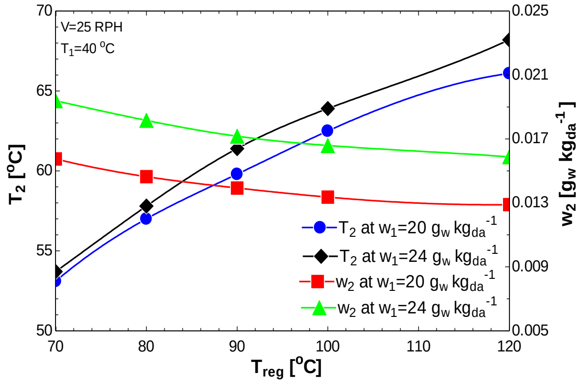

Fig. 6 The effect of regeneration air temperature, on the outlet air conditions from desiccant wheel.

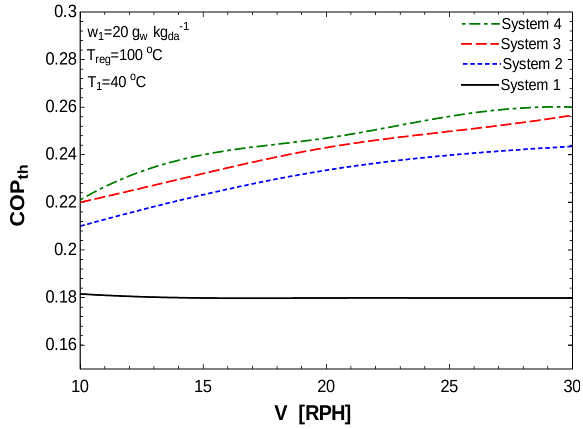

Fig. 7(a)

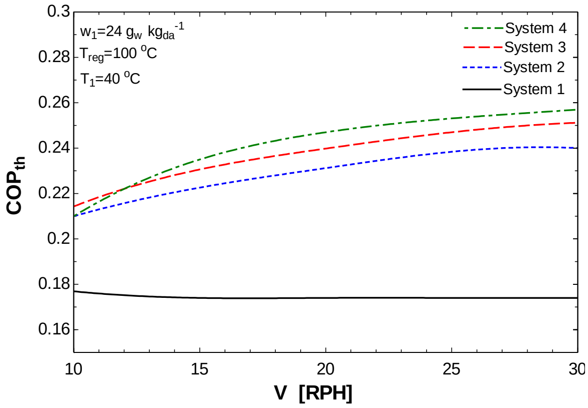

Fig. 7(b)

Fig. 7 The effect of wheel speed on the coefficient of performance, COP_{th} of all considered systems.

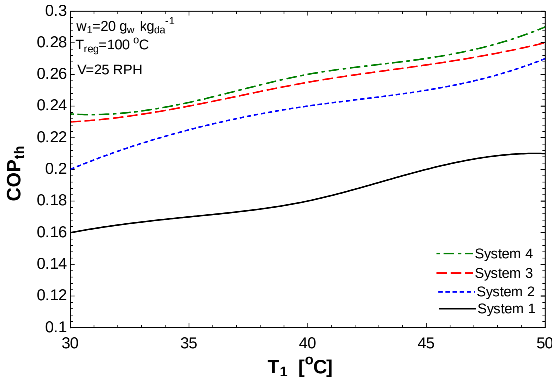

Fig. 8(a)

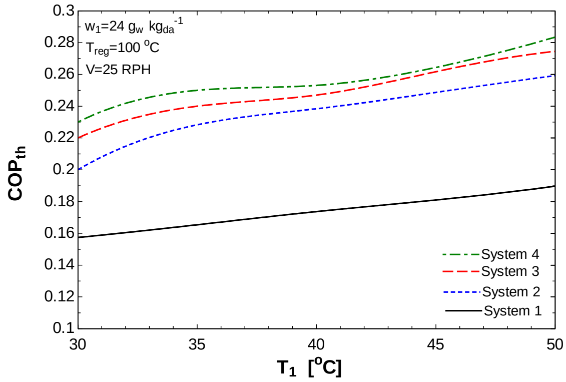

Fig. 8(b)

Fig. 8 The effect of inlet air temperature on the coefficient of performance, COP_{th} of all considered systems.

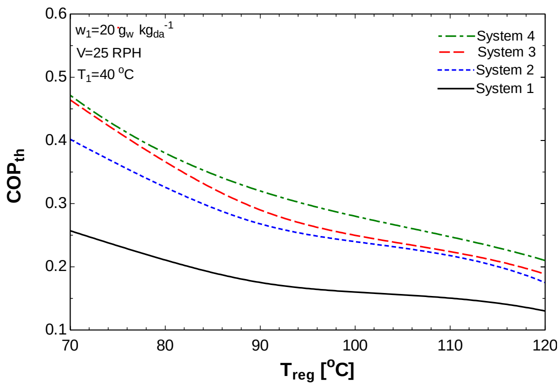

Fig. 9(a)

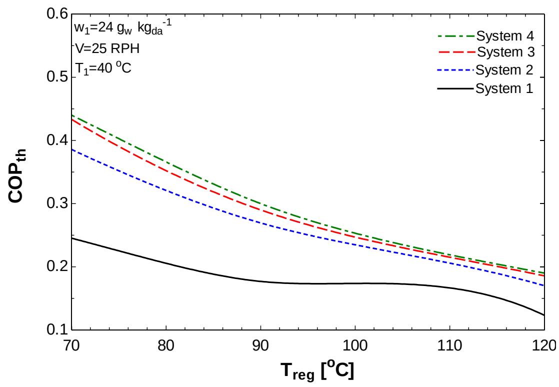

Fig. 9(b)

Fig. 9 The effect of regeneration air temperature on the coefficient of performance, COPth of all considered systems.

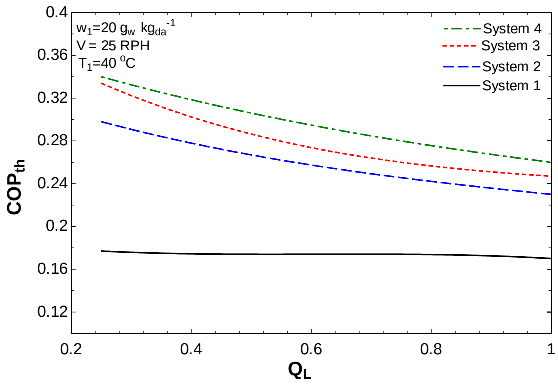

Fig. 10(a)

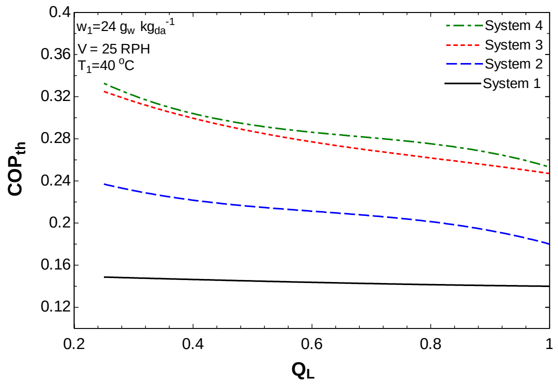

Fig. 10(b)

Fig. 10 The effect of cooling load of space on the coefficient of performance, COP_{th} of all considered systems.

7. Conclusion

In this paper, we have studied four air conditioning systems such that each system consists mainly of vapor compression cycle and rotating desiccant wheel. We analyzed the influence of important operating parameters including outlet air temperature, outlet air humidity, regeneration air temperature, and desiccant wheel speed on the performance of the systems considered. The main conclusions from the results of this study are listed as follows:

1. The outlet air temperature from the desiccant wheel in the process air stream increases with the increasing of the regeneration air temperature and also increases with the increasing of the inlet air humidity ratio.
2. The outlet humidity ratio from the desiccant wheel in the process air stream decreases with the increasing of the regeneration air temperature and increases with the increasing of the inlet air humidity ratio.
3. When the inlet air humidity ratio and the regeneration air temperature increase, the COP_{th} of the air conditioning system decreases.
4. When the desiccant wheel speed increases, the COP_{th} of the air conditioning system increases.
5. In these four systems, COP_{th} is highest at system 4 (which contains desiccant wheel /heat exchanger and ground source circulation).
6. COP_{th} is improved about 48% and a consequent power consumption reduction of 20 % at system 4.
7. COP_{th} decreases with the increasing of the cooling load for the space, and increases with the decreasing of the inlet air humidity ratio at process air.

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