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Abstract

This paper presents a feasibility investigation of integrating a hybrid photovoltaic thermal collector-solar air heater (PVT-SAH) and an air-based thermal energy storage (TES) system using phase change materials (PCMs) with rotary desiccant cooling systems for residential applications. The PVT-SAH is used to generate both electricity and thermal energy, while the TES unit is used to solve the mismatch between energy demand for desiccant wheel regeneration and thermal energy generation from the PVT-SAH. A near-optimal design of the proposed system is first identified using the response surface method. The feasibility is then examined using three performance indicators, including Solar Thermal Contribution (STC), Supply Air Temperature Unsatisfied (SATU) factor and Supply Air Humidity Ratio Unsatisfied (SAHRU) factor. The results showed that the STC increased from 82.6% to 100.0% when the regeneration temperature decreased from 70 °C to 60 °C under the simulation days. For the regeneration temperature considered (i.e. 60 °C, 65 °C and 70 °C), the supply air temperature can always be satisfied while the SAHRU factor decreased from 24.2% to 6.0% when the regeneration temperature increased from 60 °C to 70 °C. The overall results illustrated that the PVT-SAH and PCM TES unit can be potentially used to regenerate desiccant wheels if the system is appropriately designed.

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Integrating photovoltaic thermal collectors and thermal energy storage systems using phase change materials with rotary desiccant cooling systems

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Abstract: This paper presents a feasibility investigation of integrating a hybrid photovoltaic thermal collector-solar air heater (PVT-SAH) and an air-based thermal energy storage (TES) system using phase change materials (PCMs) with rotary desiccant cooling systems for residential applications. The PVT-SAH is used to generate both electricity and thermal energy, while the TES unit is used to solve the mismatch between energy demand for desiccant wheel regeneration and thermal energy generation from the PVT-SAH. A near-optimal design of the proposed system is first identified using the response surface method. The feasibility is then examined using three performance indicators, including Solar Thermal Contribution (STC), Supply Air Temperature Unsatisfied (SATU) factor and Supply Air Humidity Ratio Unsatisfied (SAHRU) factor. The results showed that the STC increased from 82.6% to 100.0% when the regeneration temperature decreased from 70 °C to 60 °C under the simulation days. For the regeneration temperature considered (i.e. 60 °C, 65 °C and 70 °C), the supply air temperature can always be satisfied while the SAHRU factor decreased from 24.2% to 6.0% when the regeneration temperature increased from 60 °C to 70 °C. The overall results illustrated that the PVT-SAH and PCM TES unit can be potentially used to regenerate

1 desiccant wheels if the system is appropriately designed.

2 Keywords: Photovoltaic thermal collector; Phase change materials; Desiccant wheel;

3 Regeneration; Feasibility study

4 **Nomenclature**

5 A area (m^2)

6 A_{fin} fin surface area (m^2)

7 b coefficient

8 C_p specific heat ($\text{J kg}^{-1} \text{K}^{-1}$)

9 E energy (J)

10 h enthalpy (J kg^{-1})

11 h_c forced convection coefficient ($\text{W m}^{-2} \text{K}^{-1}$)

12 h_{nc} natural convection coefficient ($\text{W m}^{-2} \text{K}^{-1}$)

13 h_r radiation heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)

14 h_w wind convection coefficient ($\text{W m}^{-2} \text{K}^{-1}$)

15 H_{fin} fin height (m)

16 I_t global solar irradiation (W m^{-2})

17 k thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)

18 $\dot{m}$ mass flow rate (kg s^{-1})

19 M mass per unit area (kg m^{-2})

20 MRC moisture removal capacity (kg h^{-1})

21 q heat flux density (W m^{-2})

22 RH relative humidity

1	STC	solar thermal contribution
2	t	time (s)
3	T	temperature ($^{\circ}\text{C}$)
4	v	velocity (m s^{-1})
5	W	water content of the desiccant material (kg kg^{-1})
6	W_d	width (m)
7	x	independent variable
8	Δx	length of a control volume (m)
9	y	objective response
10	Y	humidity ratio (kg kg^{-1} dry air)
11	<i>Greek symbols</i>	
12	α	absorptivity
13	δ	thickness (m)
14	η	dehumidification effectiveness
15	θ	mass flow rate ratio of regeneration air to process air
16	ρ	density (kg m^{-3})
17	<i>Subscripts</i>	
18	a	air
19	amb	ambient
20	ap	absorber plate
21	b	before
22	bp	bottom plate

1	<i>eh</i>	electric heater
2	<i>g</i>	glass cover
3	<i>in</i>	inlet
4	<i>pro</i>	process
5	<i>pv</i>	PV plate
6	<i>reg</i>	regeneration
7	<i>sc</i>	solar contribution
8	<i>Abbreviations</i>	
9	ANOVA	Analysis of Variance
10	COP	Coefficient of Performance
11	DCOP	Dehumidification Coefficient of Performance
12	DSC	Differential Scanning Calorimetry
13	IEC	Indirect Evaporative Cooler
14	IWEC	International Weather for Energy Calculations
15	PCM	Phase Change Material
16	PV	Photovoltaic
17	PVT	Photovoltaic Thermal
18	RSM	Response Surface Method
19	SAH	Solar Air Heater
20	SAHRU	Supply Air Humidity Ratio Unsatisfied
21	SATU	Supply Air Temperature Unsatisfied
22	TES	Thermal Energy Storage

1. Introduction

Over the last two decades, rotary desiccant cooling systems have been considered as one of alternative approaches to replacing conventional vapor compression systems as such systems do not use chlorofluorocarbons and have the capability of independent temperature and humidity control (La, Dai, Li, Wang, & Ge, 2010; Jani, Mishra, & Sahoo, 2016a). Compared to traditional vapour compression systems, rotary desiccant cooling systems are more energy efficient and environmentally friendly (La et al. 2010).

Various types of rotary desiccant cooling systems and their potentials to maintain acceptable indoor thermal comfort under different climatic conditions have been studied (Jani, Mishra, & Sahoo, 2015; Jani et al. 2016a; Jani, Mishra, & Sahoo, 2016b; Jani, Mishra, & Sahoo, 2016c; Khalid, Mahmood, Asif, & Muneer, 2009; Baniyounes, Liu, Rasul, & Khan, 2013). For instance, White, Kohlenbach, and Bongs (2009) evaluated the performance of a 100% fresh air solar desiccant cooling system. It was found that the indoor thermal comfort could be maintained at or near the acceptable levels using the proposed system under the Melbourne and Sydney climatic conditions. Enteria et al. (2010) developed a solar cooling and heating system with double cross-flow heat exchangers. The experimental results indicated that the amount of cooling produced by the system was not proportional to the thermal energy supplied to the desiccant cooling system and the regeneration temperature of 65 °C was enough to support the dehumidification process. Baniyounes, Liu et al. (2013) investigated the performance of a solar assisted absorption cooling system and a solar assisted rotary desiccant cooling system for an institutional building. The simulation results showed that the desiccant cooling system achieved a higher solar fraction than the absorption cooling

1 system. Baniyounes, Rasul, and Khan (2013) experimentally investigated the performance of
2 a solar desiccant cooling system. The results showed that the proposed system could achieve
3 18% energy savings, in comparison to a conventional heating, ventilation and air-conditioning
4 system. Zeng, Li, Dai, and Xie (2014) numerically investigated the annual performance of a
5 solar hybrid one-rotor two-stage desiccant cooling and heating system under the weather
6 condition of Shanghai, China. The simulation results showed that about 60% of humidity load
7 could be handled by the proposed system and 40% of the heating demand in winter could be
8 covered by solar energy. Mei, Infield, Eicker, Loveday, and Fux (2006) investigated a solar
9 rotary desiccant cooling system, in which the regenerated air was first heated by a PV façade
10 and PV sheds in parallel and then further heated by solar air collectors. The simulation results
11 indicated that the average thermal COP of the desiccant cooling system in the summer period
12 was 0.518 and a regeneration temperature of about 70 °C can be provided by the proposed
13 PV-solar air heating system. A similar desiccant cooling system with air-based PVT and solar
14 air collectors connected in series to regenerate a desiccant wheel was studied by Beccali,
15 Finocchiaro, and Nocke (2009). The results showed that the majority (up to 90%) of the
16 cooling demand of the case study office can be covered by the solar-assisted desiccant cooling
17 system. In the aforementioned studies, the heated air generated from solar systems was
18 directly used to regenerate rotary desiccant cooling systems. However, these systems cannot
19 satisfy building cooling demand during the nighttime.

20 Thermal energy storage (TES) using phase change materials (PCMs) can be used as an
21 alternative solution to solve the discrepancy between thermal energy demand and supply.
22 PCMs with the ability to provide high energy storage densities and the characteristics to store

thermal energy at a relatively constant temperature have attracted increasing attention for developing high-performance buildings (Ma, Lin, & Sohel, 2016). For instance, Fiorentini, Wall, Ma, Braslavski and Cooper (2017) presented an air-conditioning system, in which an air-based PCM TES unit was used to regulate the thermal energy demand of a house and thermal energy generation from the PVT collectors and a model predictive strategy was used to control the operation of this system. A mathematical model for a PCM-air heat exchanger was developed by Dolado, Lazaro, Marin, and Zalba (2011) to examine the PCM melting and solidification characteristics. The results indicated that increasing the rugosity of PCM encapsulation can significantly improve the convective heat transfer between the air and the PCM. Lin, Ma, Sohel, and Cooper (2014) developed a PVT-PCM integrated ceiling ventilation system for space heating and cooling. It was found that this system can effectively improve the indoor thermal comfort of passive buildings without using air-conditioning systems in winter conditions. Charvát, Klimeš, and Ostrý (2014) experimentally and numerically investigated a PCM TES unit for solar air systems. It was shown that the PCM in the first row of the panels was fully melted in less than one hour while it took several hours to melt the PCM in the last row of the panels. The results from the above studies demonstrated that PCMs could play an essential role in effective building thermal energy management.

This study presents a feasibility study of integrating a hybrid photovoltaic thermal collector-solar air heater (PVT-SAH) and a PCM TES unit with a rotary desiccant cooling system. The regeneration temperatures commonly used in rotary desiccant cooling systems are first briefly reviewed. The feasibility study is then performed based on three regeneration temperatures and three performance indicators including Solar Thermal Contribution (STC),

Supply Air Temperature Unsatisfied (SATU) factor and Supply Air Humidity Ratio Unsatisfied (SAHRU) factor. The novelty of this present work is to use a PCM TES unit to solve the discrepancy between the thermal energy demand and the thermal energy generation of PVT-SAH systems to maximize the solar thermal contribution to the regeneration of rotary desiccant cooling systems.

2. Overview of regeneration temperatures used in rotary desiccant cooling systems

In a rotary desiccant cooling system, the cooling process is achieved by removing the moisture from the process air using a desiccant wheel and reducing the temperature of the process air using evaporative cooling or other cooling technologies.

The desiccant wheel is therefore one of the key components in a rotary desiccant cooling system. A desiccant wheel generally consists of a large number of air channels whose walls coated or impregnated with desiccant materials (Ge, Ziegler, & Wang, 2010). The cross-section of a desiccant wheel can be divided into the process side and the regeneration side. The moisture of the process air is adsorbed by the desiccant material when it passes through the process side of the desiccant wheel and at the same time, the regeneration air flows through the regeneration side where the desiccant material is heated and dehumidified.

The regeneration air temperature (i.e. regeneration temperature) is one of the significant variables influencing the performance of a rotary desiccant cooling system. In general, the desiccant materials can capture more moisture from the process air if the materials are more deeply dried under a higher regeneration temperature. Moisture removal capacity (MRC), which represents the mass flow rate of the moisture removed from the process air by a rotary

desiccant cooling system (Angrisani, Minichiello, Roselli, & Sasso, 2012), increases with the increase of the regeneration temperature as the desorption process in the desiccant matrix is endothermic (Sheng et al., 2014). A study showed that the dehumidification effectiveness (η), which represents the ratio between the real and the ideal dehumidification capability of a desiccant wheel (Angrisani et al., 2012), increased from 0.54 to 0.58 when the regeneration temperature increased from 60 °C to 90 °C (Eicker et al., 2012). The study from Ali, Vukovic, Sahir, & Basciotti (2013) showed that the increase rate of MRC of a rotary desiccant cooling system decreased with increasing regeneration temperature. For a rotary desiccant cooling system, the maximum dehumidification coefficient of performance (DCOP), which is the ratio of the latent heat contained in the adsorbed moisture to the thermal energy required to produce the high-temperature regeneration air, can be obtained at a certain regeneration temperature (Ge et al., 2010). If the regeneration temperature is beyond this value, the energy cost required to produce a higher temperature of the regeneration air will be higher than the benefit resulted from the increased moisture removal. For desiccant wheels using composite desiccants, the maximum DCOP can be generally obtained when the regeneration temperature was around 85 °C (Ge et al., 2010). Therefore, there is an optimal regeneration temperature which can maximize the overall performance of a rotary desiccant cooling system.

Table 1 summarizes the regeneration temperatures used in the rotary desiccant cooling systems studied. It can be seen that the regeneration temperature studied varied between 35 °C and 120 °C. Several studies showed that the dehumidification efficiency of rotary desiccant cooling systems can be generally acceptable when the regeneration temperature was in the range of 60-70 °C.

3. System description and research method

3.1. System description

A schematic of a rotary desiccant cooling system with a hybrid PVT-SAH and a PCM TES unit is illustrated in Fig. 1. In this system, the hybrid PVT-SAH, in which the PVT collector and the SAH are connected in series, was used for both electricity and low-grade thermal energy generation. The thermal energy collected from the PVT-SAH can be used to drive the desiccant wheel regeneration in cooling conditions or for space heating in heating conditions. The electricity generated by the PV plate can be used to power the electric heater and fans used in the system and the excessive electricity can be stored in a battery or exported to the grid. The use of such a hybrid system is to achieve a relatively high air temperature while still maintaining necessary electricity generation and reducing the system operation cost (Beccali, et al., 2009). An air-based PCM TES unit (see Fig. 2) with a similar configuration as that reported by Charvát et al. (2014) was used to regulate the discrepancy between the thermal energy generated from the PVT-SAH and the thermal energy demand for the desiccant wheel regeneration. The PCM TES unit consisted of a number of PCM layers arranged in parallel and each PCM layer consisted of multiple PCM panels. A desiccant wheel, a heat recovery unit and an indirect evaporative cooler (IEC) were used to condition the process air.

This proposed system can be used for both daytime and nighttime cooling dependent on the building cooling demand. It can also be used for winter space heating. Table 2 describes the main potential operation modes.

In the process air side, the return air from the indoor space is first mixed with the fresh air

and then used as the process air in order to improve the system efficiency. The process air is first dehumidified by the desiccant wheel and then cooled by a heat recovery unit and an IEC. Under the operation mode IV, the regeneration air is first processed by the heat recovery unit and then by the PCM TES unit while under the operation modes I and II, the outlet air from the heat recovery unit is discharged to ambient directly. It is worthwhile to note that a bypass was designed for the PCM TES unit in order to avoid the overheating of the regeneration air.

3.2. Research method

The overall procedure used for the feasibility study of the proposed system is illustrated in Fig. 3, which consisted of three steps including setup of the test, performance evaluation and optimization, and the feasibility analysis. In the first step, a virtual simulation system of the proposed system was developed using TRNSYS (Klein, 2010), which will be introduced in Section 4, and the key performance indicators to be used for performance evaluation were determined. In the second step, the key factors governing the performance of the PVT-SAH and PCM TES unit were first determined based on the results from the previous studies (Dolado et al., 2011; Duffie, & Beckman, 2013; Fan, Kokogiannakis, Ma, & Cooper, 2017). The simulation cases were then designed and executed using the response surface method. Based on the simulation data, the response surface models were developed and the optimal combination of the factors was determined. In the third step, a confirmation test with the optimal combination of the factors identified was carried out and the feasibility of the proposed system for desiccant wheel regeneration was analysed.

3.2.1. Selection of performance indicators

In this study, three performance indicators, including Solar Thermal Contribution (STC),

Supply Air Temperature Unsatisfied (SATU) factor and Supply Air Humidity Ratio Unsatisfied (SAHRU) factor, were used to evaluate the performance of the proposed system under dynamic working conditions. STC is defined as the ratio of the total amount of thermal energy that can be provided by the hybrid PVT-SAH and PCM TES unit for desiccant wheel regeneration to the total thermal energy needed for desiccant wheel regeneration under a required regeneration temperature and a time period of concern, as expressed in Eq. (1) (Enteria et al., 2011). The thermal energy supplied by the PVT-SAH and PCM TES unit is determined by Eq. (2), in which the air temperature before the electric heater ($T_{b,eh,a}$) can be the temperature of the outlet air from the PVT-SAH or from the PCM TES unit dependent on which operation mode was used for desiccant wheel regeneration. The power consumption of the auxiliary electric heater is determined by Eq. (3) if the outlet air temperature from the PVT-SAH or PCM TES unit was less than the required regeneration temperature. The regeneration air flow rate ($\dot{m}_{reg,a}$) was calculated using Eq. (4) based on the configuration of the desiccant wheel.

$$STC = \frac{E_{sc}}{E_{sc} + E_{eh}} \quad (1)$$

$$E_{sc} = \int_{t_1}^{t_2} [\dot{m}_{reg,a,i} C_{p,a} (T_{b,eh,a,i} - T_{a,in,i})] dt \quad (2)$$

$$E_{eh} = \int_{t_1}^{t_2} [\dot{m}_{reg,a,i} C_{p,a} (T_{reg,a} - T_{b,eh,a,i})] dt \quad (3)$$

$$\dot{m}_{reg,a} = \theta \dot{m}_{pro,a} \quad (4)$$

where E is the thermal energy, $\dot{m}$ is the mass flow rate, T is the temperature, C_p is the specific heat, t is the time, θ is the ratio of the mass flow rate of the regeneration air to that of the process air, t_1 and t_2 are the start time and end time respectively, and the subscripts a , b , in , pro , reg , sc , and eh indicate air, before, inlet, process, regeneration, solar contribution and

electric heater, respectively. It is noted that $T_{a,in}$ in Eq. (2) represents the temperature of the ambient air to the hybrid PVT-SAH system when the desiccant wheel is directly regenerated by the air supplied by the PVT-SAH (operation modes I and II). Otherwise, it represents the inlet air temperature of the PCM TES unit if the system is operated under the operation mode IV.

SATU factor and SAHRU factor were used to evaluate the performance of the rotary desiccant cooling system under different regeneration temperatures. SATU factor was defined as the ratio of the accumulated time that the supply air temperature is above the required set-point to the total time of the desiccant cooling system in operation. Similarly, SAHRU factor was defined as the ratio of the accumulated time that the supply air humidity ratio is above the required set-point to the total time of the desiccant cooling system in operation.

3.2.2. Performance investigation and optimization using response surface method

In this study, a range of simulation exercises was designed and executed using the response surface method (RSM) to determine the near-optimal values of the main parameters of the hybrid PVT-SAH and PCM TES unit. The RSM is a set of mathematical and statistical techniques to optimize a response of interest which is influenced by a number of variables (Montgomery, 2008). As the primary focus of this study was to examine the feasibility of using the hybrid PVT-SAH and PCM TES unit to regenerate desiccant wheels, the STC was therefore selected as the response of the hybrid PVT-SAH and PCM TES unit. The SATU factor and SAHRU factor were only used to evaluate the performance of the rotary desiccant cooling system and were not considered in the RSM. The STC was investigated with three levels (-1, 0, 1) for each main parameter and the Face Centered Central Composite Design

(Montgomery, 2008; Subasi, Sahin, & Kaymaz, 2016) was used for the simulation design. Based on the simulation results, a response surface model was generated through the stepwise regression method, which was then used to predict the response of STC based on the independent variables under various conditions. The general form of the response surface model is presented in Eq. (5) (Wang, Lan, & Li, 2014).

$$y = b_0 + \sum_{i=1}^j b_i x_i + \sum_{i=1}^k b_{ii} x_i^2 + \sum_{i=1}^{i < j} \sum_j b_{ij} x_i x_j \quad (5)$$

where y is the predicted response, x_i and x_j are the independent variables, and b_0 , b_i , b_{ii} , and b_{ij} are the coefficients.

Analysis of variance (ANOVA) was used to evaluate the fitness of the model. The response surfaces were generated to visualize the individual and interactive effects of the main parameters on the response. The optimal combination of the main parameters to maximize the response (i.e. STC) was then determined. The RSM process was implemented using the trial version of Design Expert (Design Expert, 2010).

The PCM type, the length of the PCM TES unit, the size of the air gap between the glass cover and the PV plate/absorber plate in the PVT-SAH, and the air flow rate of the PVT-SAH, were considered as the major parameters in this study based on the following considerations. The PCM melting temperature and thermophysical properties will directly influence the amount of thermal energy that can be charged into the TES unit. The length of the PCM TES unit was used to control the total amount of the PCM used and therefore the storage capacity of the TES unit. The air flow rate of PVT-SAH influences its heat transfer performance and outlet air temperature. The air gap between the PV plate/absorber plate and the glass cover influences the heat loss from the PVT-SAH to the ambient environment. It is noteworthy that

the air flow rate of the TES unit during the charging process was the same as that of the PVT-SAH while that during the discharging process was determined based on the building cooling demand.

4. System modelling

4.1. Hybrid PVT-SAH

The hybrid PVT-SAH system considered in this study is shown in Fig. 4, which consisted of a glass cover, a PV plate, an absorber plate, a bottom plate, and a number of fins. The fins were deployed between the bottom plate and absorber plate along the air flow direction to enhance the heat transfer performance. A dynamic model developed in a previous study (Fan et al., 2017) was used to simulate the performance of the hybrid PVT-SAH system. In this model, the PVT-SAH was first discretized into multiple control volumes along the air flow direction, and the PVT collector was further divided into 6 nodes perpendicular to the flow direction, including the glass cover, PV plate, absorber plate, fins, fluid air, and the bottom plate, while the SAH was divided into 5 nodes perpendicular to the flow direction by excluding the node for the PV plate. The key governing equation of the glass cover node in the PVT section is described in Eq. (6). The governing equations of the PV plate, the solar absorber plate, the fins, and the bottom plate can be described using the same way. The governing equation for the air flowing through the air channel is given by Eq. (7).

$$AC_{p,g}M_g \frac{\partial T_{g,i}}{\partial t} = \alpha_g I_t A + h_{nc} A (T_{pv,i} - T_{g,i}) + h_{r,pv-g} A (T_{pv,i} - T_{g,i}) + h_w A (T_{amb} - T_{g,i}) + h_{r,g-sky} A (T_{sky} - T_{g,i}) \quad (6)$$

$$C_{p,a} \rho_a \Delta x (W_a H_{fin}) \frac{\partial T_{a,i}}{\partial t} + C_{p,a} \dot{m}_a \Delta x \frac{\partial T_{a,i}}{\partial x} = h_{c,ap-a} A (T_{ap,i} - T_{a,i}) + h_{c,bp-a} A (T_{bp,i} - T_{a,i}) + 2h_{c,fin-a} A_{fin} (T_{fin,i} - T_{a,i}) \quad (7)$$

where A is the area, M is the mass per unit area, α is the absorptivity, I_t is the global solar irradiation, h_{nc} is the natural convection coefficient, h_r is the radiation heat transfer coefficient, h_w is the wind convection coefficient, ρ is the density, W_d is the PVT-SAH width, Δx is the length of the control volume, H_{fin} is the fin height, h_c is the forced convection coefficient, A_{fin} is the surface area of fins for a control volume in contact with flowing air, and the subscripts amb , g , pv , ap , and bp indicate ambient, glass cover, PV plate, absorber plate and bottom plate, respectively.

Similarly, the governing equations for each node of the SAH can be developed. By employing the Crank-Nicolson scheme, the above equations were discretized and solved. More details of this model can be found in Fan et al. (2017).

4.2. PCM TES unit

A mathematical model for the PCM TES unit, which considered the hysteresis phenomenon but did not consider the supercooling during the phase change process, was developed using the enthalpy method based on the three main assumptions: a) the conduction heat transfer within the PCM panel is one-dimensional and perpendicular to the air flow direction; b) there is no natural convection in the air within the TES unit and; c) the convective heat transfer within the liquid PCM is negligible.

The schematic of the nodes in modelling the PCM TES unit is illustrated in Fig. 5. The PCM TES unit was divided into multiple sections containing both air nodes and PCM panels along the air flow direction. The governing equations for the energy balance of PCM panels and air flow are expressed in Eqs. (8) and (9), respectively.

The Gnielinski equation was used to calculate the Nusselt number of the turbulent flow

(Lin et al., 2014; Gnielinski, 1976), and the Nusselt number of the laminar flow was determined based on the aspect ratio of air channels (Bergman, Incropera, DeWitt, & Lavine, 2011). The governing equations were solved based on the enthalpy-temperature relationship of the PCM, which can be obtained from the Differential Scanning Calorimetry (DSC) test or from the data sheets provided by manufacturers.

$$\rho_{PCM} \frac{\partial h_{PCM}}{\partial t} = k_{PCM} \frac{\partial^2 T_{PCM}}{\partial y^2} \quad (8)$$

$$\rho_a C_{p,a} \left(\frac{\partial T_a}{\partial t} + v_a \frac{\partial T_a}{\partial x} \right) = \frac{q_{up} + q_{down}}{\delta_a} \quad (9)$$

where h is the enthalpy, k is the thermal conductivity, v is the velocity, q is the heat flux density, and δ is the thickness.

4.3. Rotary desiccant cooling system

The model of the rotary desiccant cooling system consisted of a heat recovery unit, an IEC, and a desiccant wheel. The heat recovery unit was modelled using TRNSYS component Type 760, in which a constant heat transfer effectiveness was assumed. The IEC was modelled using Type 757 with a constant effectiveness and without considering the water consumption. In this model, the outlet air condition from the IEC was determined based on the assumption that the secondary air stream process is a constant wet-bulb temperature process. The gas side and solid side resistance model for the desiccant wheel developed by Ge et al. (2010) was used in this study. The regular density silica gel was used as the desiccant material and the equilibrium relative humidity (RH) on its surface was described by Eq. (10) (Pesaran, & Mills, 1987).

$$RH = 0.0078 - 0.05759W + 24.16554W^2 - 124.478W^3 + 204.226W^4 \quad (10)$$

where W is the water content of the desiccant material.

5. Results and discussion

5.1. Validation of the PCM TES unit model

The effectiveness of the PCM TES unit model was validated using the experimental data presented by Lopez, Kuznik, Baillis, and Virgone (2013), in which the inlet and outlet temperatures of the PCM TES unit were measured using type-K thermocouples with an accuracy of ± 0.4 °C. The model developed was modified to have the same configuration as that of the experimental setup used by Lopez et al. (2013) through reducing the number of the air channels and assuming that the TES unit was well-insulated.

A comparison of the model simulation results with the experimental data reported by Lopez et al. (2013) under the air flow rate of $240 \text{ m}^3 \text{ h}^{-1}$ is presented in Fig. 6. The simulated outlet air temperature from the PCM TES unit under both charging and discharging processes generally matched well with that of the experimental data probably due to the consideration of the hysteresis phenomenon during the phase change process, indicating that the model developed can provide an acceptable estimation.

5.2. Setup of the simulation tests

In this study, the proposed system was assumed to be used to condition a Solar Decathlon (SD) house. The house was divided into two thermal zones, including the living space of 43.0 m^2 and the sleeping space of 23.0 m^2 . More details of the SD house can be found in Fiorentini (2016). The simulation study was carried out based on the working days. The house was assumed to be occupied with two residents and the schedule of the occupant activities is described in Table 3. It was assumed that the hybrid PVT-SAH was installed on the north roof of the house with a total area of 24 m^2 and a roof slope of 18.4° . It is worthwhile to note that a

larger area of PVT-SAH could bring more benefits. The performance of the proposed system was evaluated under the operation modes III and IV as described in Table 2. It is worthwhile to note that the system should be re-designed if it is primarily operated under the other operation modes or there is a cooling demand during both daytime and nighttime.

Five consecutive summer days under Brisbane (Australia) weather conditions with relatively higher temperatures and humidity ratios were selected to investigate the feasibility of using the hybrid PVT-SAH and PCM TES unit for desiccant wheel regeneration as Brisbane has reasonably abundant solar radiation and the humidity ratio is relatively high. The weather data used in the simulation were the International Weather for Energy Calculations (IWECC) data. Fig. 7 shows the ambient air temperature, humidity ratio, and total horizontal solar radiation over the selected summer days in Brisbane.

The SD house model developed using DesignBuilder (DesignBuilder, 2017) in a previous study (Fiorentini, 2016) was used to simulate the cooling demand of the house during the simulation days. The required supply air flow rate and the process air flow rate were then determined based on the flow ratio between the primary and secondary air of the IEC. The regeneration air flow rate was determined using Eq. (4). The regeneration air was assumed to be heated from the ambient air temperature to a required regeneration temperature using the heat recovery unit, the PCM TES unit and the auxiliary electric heater during the nighttime.

Three scenarios with different regeneration temperatures of 60 °C, 65 °C, and 70 °C were used to evaluate the effect of the regeneration temperature on the performance of the proposed system in terms of the three performance indicators used. The simulation plans were designed for each scenario using the RSM. The levels of the four main parameters used in each

scenario are specified in Table 4. The PCM type was considered as a categorical variable while the length of the PCM TES unit, the air gap of the PVT-SAH and the air flow rate were regarded as continuous variables. In principle, the levels of the PCM types should be different for different regeneration temperatures. However, the same PCM types were used for Scenarios A and C in this study mainly based on the consideration of the outlet air temperature of the PVT-SAH. The levels of the other parameters were considered as the same for the three different scenarios. The thermophysical properties of the PCMs used are summarized in Table 5 and an example of the enthalpy-temperature relationship of the RT70HC is presented in Fig. 8. During the simulation, the initial temperature of the PCM in the TES unit was set as 50 °C to ensure that there was no latent thermal energy stored in the PCM. The range of the size of the air gap between the glass cover and the PV plate/absorber plate was determined based on the results of a previous study (Duffie et al., 2013). The range of the air flow rate was selected in order to obtain a reasonably high outlet air temperature from the PVT-SAH system. The length of the PCM TES unit was roughly determined based on the house cooling demand and the average thermal COP (i.e. 0.518) of a rotary desiccant cooling system reported in a previous study (Mei et al., 2006).

The details of the hybrid PVT-SAH and PCM TES unit and the rotary desiccant cooling system used in this study are summarized in Table 6. To avoid a large pressure drop in the PCM TES unit and improve the convective heat transfer between the air flow and the PCM panel, the size of the air channels in the PCM TES unit was set as 10 mm based on the value recommended in a previous study (Dolado et al., 2011). The width of the TES unit was set as the same as the width of the PCM panel used (Rubitherm GmbH, 2016). The number of PCM

layers was calculated based on the height of the TES unit which was set as 0.6 m assuming that the unit will be placed under the suspended floor of the house. The channel depth of the PVT-SAH was set as 25 mm based on the result from a previous study (Fan et al., 2017). The ratio of the length of the PVT to that of the PVT-SAH was determined as 0.6 to ensure an acceptable amount of electricity generation and also to achieve a relatively high outlet air temperature from the PVT-SAH for desiccant wheel regeneration. The ratio of the regeneration side area to that of the process side and the ratio of the regeneration air flow rate to that of the process air were set as 1:1. The resulted simulation design for the scenario A is summarized in Table 7 and the simulation designs for Scenarios B and C were similar to that of Scenario A and were therefore not provided.

5.3. Results from the performance simulation

Table 7 summarizes the solar thermal contribution (STC) for all simulation exercises designed for Scenario A when the regeneration temperature was 65 °C. It can be seen that the STC was in the ranges of 57.9-93.0%, 52.2-95.7% and 33.3-91.4% when the PCMs of RT60, RT65 and RT70HC were used.

A response surface model was then generated for each PCM type under the regeneration temperature of 65 °C based on the simulation results. Through the step-wise regression, the general response surface model for three PCMs determined is expressed in Eq. (11) and the coefficients used are summarized in Table 8.

$$STC = b_0 + b_1x_1 + b_2x_2 + b_3x_3 + b_{11}x_1^2 + b_{22}x_2^2 + b_{12}x_1x_3 \quad (11)$$

where x_1 , x_2 , and x_3 stand for the length, air gap, and air flow rate, respectively.

Fig. 9 a-c) present the variation in STC when changing the PCM TES length and the air

gap between the glass cover and PV plate/absorber plate with different PCM types under the air flow rate of 500 kg h^{-1} and the regeneration temperature of 65°C while Fig. 9d) presents the variation in STC with the change of the PCM TES length and air flow rate while keeping the air gap of 16.0 mm, PCM type of RT65, and regeneration temperature of 65°C . It can be observed that the STC increased with the increase of PCM TES length under the same PCM type and the air gap. The STC first increased and then decreased with increasing air gap under the same PCM type and the PCM TES length. The optimal value of the air gap which can maximize the STC was around 14.0-18.0 mm for different types of PCMs considered. The highest STC under the flow rate of 500 kg h^{-1} was achieved when using RT65 among the three PCMs tested. From Fig. 9d), it can be seen that the STC increased with the decrease of the flow rate under a given PCM TES unit length, which was resulted from the increase in the outlet air temperature of the PVT-SAH. Therefore, a relatively small charging air flow rate was suggested in this proposed system.

Based on the response surface model, the optimal combination of the parameters to maximize the STC of the hybrid PVT-SAH and PCM TES unit with the regeneration temperature of 65°C was determined and the results are presented in Table 9. The predicted STC was 95.2% and the result from the confirmation test was 96.5% with a relative error of 1.3%. Similar procedures were also carried out for Scenarios B and C to generate the response surface models and the similar trends of STC as those of Scenario A were also observed. The optimal designs identified for Scenario B and C are also summarized in Table 9 while the detailed results were not provided in order to save the page size. It is worthwhile to note that due to the prediction error of the response surface model, the STC predicted under some

conditions were slightly higher than 100%. However, in principle, this is not possible. From Table 9, it is interesting to note that RT65 was selected as the optimal PCM for the three different scenarios with different regeneration temperatures. This was mainly due to the impact of the outlet air temperature from the PVT-SAH. It should be noted that the near-optimal designs identified based on the five selected summer days might not be the near-optimal solutions if the optimization was carried out based on the whole cooling season.

The simulation results based on the optimal values identified (i.e. Table 9) for Scenario A are presented in Fig. 10. It can be seen that the supply air temperature and humidity ratio can be generally controlled below 20 °C and 0.008 kg kg⁻¹ dry air, respectively. During the majority of the test period, the heat from the hybrid PVT-SAH and the PCM TES unit can satisfy the heating demand for desiccant wheel regeneration when the regeneration temperature was 65 °C.

The supply air conditions of the rotary desiccant cooling system under the regeneration temperatures of 60 °C, 65 °C, and 70 °C and under the optimal values identified (i.e. Table 9) over the selected summer days are presented in Fig. 11. It can be seen that the humidity ratio decreased with the increase of the regeneration temperature, indicating that the moisture removal capacity of the desiccant wheel increased with the increase of the regeneration temperature. During the simulation period, the supply air temperature was always maintained lower than the desired condition (i.e. 20 °C), indicating that the sensible load was fully satisfied and therefore the SATU factors for three cases were zero. The humidity ratio of the supply air was occasionally higher than the desired condition (i.e. 0.0075 kg kg⁻¹ dry air), in particular for the regeneration temperature of 60 °C. The SAHRU factors under the

regeneration temperatures of 60 °C, 65 °C and 70 °C were 24.2%, 11.6% and 6.0% while the STCs were 100%, 96.5%, and 82.6%, respectively. From Fig. 11, it can also be seen that a lower supply air temperature and a lower humidity ratio than the required conditions were provided during the majority of the test period. This indicated that the use of optimal control by appropriately varying the variables such as the regeneration temperature and supply air flow rate may further increase the STC and reduce the SAHRU factor. The STC could also be increased by increasing the length of the PVT-SAH to increase its outlet air temperature. However, the decision should be made based on the detailed cost-benefit analysis.

Based on the above results, it can be concluded that the use of the proposed hybrid PVT-SAH and PCM TES unit to drive the desiccant wheel regeneration is technically feasible under Brisbane weather conditions if the system is appropriately designed and the regeneration temperature is properly selected.

6. Conclusions

This paper investigated the feasibility of using the hybrid PVT-SAH and PCM TES unit to drive the regeneration of the rotary desiccant cooling system for residential applications. The feasibility analysis was carried out based on a Solar Decathlon house under Australian Brisbane summer weather conditions in terms of the performance indicators of Solar Thermal Contribution (STC), Supply Air Temperature Unsatisfied (SATU) factor and Supply Air Humidity Ratio Unsatisfied (SAHRU) factor.

The results showed that the sensible cooling load can be satisfied under all three scenarios with the regeneration temperatures of 60 °C, 65 °C and 70 °C and the SAHRU factor was decreased from 24.2% to 6.0% when the regeneration temperature increased from

60 °C to 70 °C. The near-optimal values of the key parameters considered for the hybrid PVT-SAH and PCM TES unit were identified using the response surface method (RSM) and it was found that the STCs of using the hybrid PVT-SAH and PCM TES unit to drive the regeneration of the rotary desiccant cooling system were 100%, 96.5% and 82.6% under different regeneration temperatures of 60 °C, 65 °C and 70 °C, respectively. The results from this study demonstrated the technical feasibility of using the hybrid PVT-SAH and PCM TES unit to drive the desiccant wheel regeneration under Australian Brisbane weather conditions if the system is well designed. The performance could be further improved through the use of optimal control strategies.

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Table 1 Summary of the desiccant wheel parameters, operating conditions and regeneration temperatures reported in previous studies

Reference	Desiccant wheel parameters				Operating Parameters						Regeneration temperature (°C)	Performance indicators		
	Diameter (m)	Thickness (m)	Area ratio ¹	Desiccant material ²	Stage	$T_{pro,a,in}$ (°C)	$Y_{pro,a,in}$ (kg kg ⁻¹)	Rotation speed (r h ⁻¹)	Flow ratio ³	$\dot{m}_{pro,a}$ (kg h ⁻¹)		MRC (kg h ⁻¹)	DCOP	η
Jia, Dai, Wu, & Wang (2016)	0.40	0.20	1:3	SG	1	30.0	0.0160	-	1:3	900	60.0-120.0	0.90-8.55	-	0.06-0.59
				CD		30.0	0.0160	-	1:3	900	60.0-120.0	1.98-10.6	-	0.14-0.74
Ge, Li, Wang, & Dai (2009)	0.26	0.10	1:3	CD	2	35.0	0.0014	8	1:2.4	432	50.0-90.0	1.38-3.63	-	0.23-0.60
						30.0	0.0016					1.74-3.91	-	0.25-0.56
						35.0	0.0023					1.99-6.48	-	0.20-0.65
Angrisani et al. (2012)	0.70	0.20	2:3	SG	1	31.6	0.0013	12	1:1	960	37.0-72.0	1.7-5.1	0.25-0.95	0.15-0.46
Eicker et al. (2012)	0.87	0.14	1:1	SG	1	32.0	0.0120	85	1:0.75	2400	60.0-90.0	9.60-13.6	-	0.54-0.58
	0.895	0.25	1:1	LC	1	32.0	0.0120	24	1:0.75	2400	45.0-70.0	8	-	0.17-0.23
Ali et al. (2013)	0.32	0.20	1:1	SG	1	31.0	0.0100	36	1:1	346	45-120	0.49-2.02	-	0.15-0.62
Sheng et al. (2014)	0.45	0.20	1:3	SG	1	28.2-4	0.0140	15	1.2:1	325	58.1-64.8	0.98-1.34	0.21-0	0.262-0.35
						0.2			-1:1				.27	9
De Antonellis, Intini, Joppolo, Molinaroli, & Romano (2015)	0.60	0.20	1:1	SMS	1	11.9-1	0.0045-	15.5	1:1	1078-10	35.1-66.0	2.59-4.24	-	0.52-0.85
						2.9	0.0046			83				
De Antonellis, Intini, & Joppolo (2015)	0.60	0.20	1:1	SMS	1	25.0	0.0120	15	1:1	1100	44.4-78.6	2.48-6.00	-	0.20-0.49

¹ Area ratio – ratio of the area of the regeneration side to that of the process side of a desiccant wheel.

² CD - composite desiccant; LC - lithium chloride; SG - silica gel; SMS - synthesized metal silicate.

³ Flow ratio – ratio of the flow rate of the regeneration air to that of the process air.

Table 2 Operation modes of the integrated desiccant cooling system

Mode	Description	Working principle
Mode I	Regeneration with PVT-SAH direct supply	The heated air from the PVT-SAH is directly used for desiccant wheel regeneration. The electric heater is used if the air temperature cannot reach the required setting.
Mode II	Regeneration with PVT-SAH direct supply & TES charging	A fraction of the heated air from the PVT-SAH is used for the desiccant wheel regeneration and the rest is used for the TES charging.
Mode III	TES charging	The heated air from the PVT-SAH is used to charge the PCM TES unit if there is no demand during the daytime.
Mode IV	Regeneration with the TES discharging	The PCM TES unit is discharged for desiccant wheel regeneration if there is a cooling demand during nighttime.
Mode V	Space heating	The heated air from the PVT-SAH is directly used for space heating or is directed into the TES unit (Note: may need to use another PCM with different melting temperature) and then used later for space heating.

Table 3 Occupants schedules used in the simulation

Time	Living space		Sleeping space		Status of PCM TES unit
	Occupant	Air-conditioned	Occupant	Air-conditioned	
8:00- 17:00	0	No	0	No	Charging
17:00 - 23:00	2	Yes if needed	0	No	Discharging if needed
23:00 - 8:00	0	No	2	Yes if needed	Discharging if needed

Table 4 Levels of the main parameters considered in the RSM design

Variables		Levels		
		-1	0	+1
PCM type	Scenario A ($T_{reg}=65\text{ }^{\circ}\text{C}$)	RT60	RT65	RT70HC
	Scenario B ($T_{reg}=60\text{ }^{\circ}\text{C}$)	RT55	RT60	RT65
	Scenario C ($T_{reg}=70\text{ }^{\circ}\text{C}$)	RT60	RT65	RT70HC
Length of the PCM TES unit (m)		3.0	5.0	7.0
Size of the air gap ¹ (mm)		5.0	12.5	20.0
Air flow rate (kg h^{-1})		500	750	1000

¹ The size of the air gap between the glass cover and the PV plate/absorber plate of the PVT-SAH.

Table 5 Thermophysical properties of the PCMs considered (Rubitherm GmbH, 2016)

Parameter	RT55	RT60	RT65	RT70HC
Melting range (°C)	51-57	55-61	57-68	69-71
Congeealing range (°C)	56-57	61-55	67-58	71-69
Specific heat (kJ kg ⁻¹ K ⁻¹)	2.0	2.0	2.0	2.0
Thermal conductivity (W m ⁻¹ K ⁻¹)	0.2	0.2	0.2	0.2
Heat storage capacity (kJ kg ⁻¹)	170	160	150	260
Density solid/liquid (kg m ⁻³)	880/770	880/770	880/780	880/770

Table 6 Specifications of the rotary desiccant cooling system, hybrid PVT-SAH and PCM TES unit

Component	Parameter	Value	Source
PVT-SAH	Length (m)	6.0	-
	Width (m)	4.0	Roof width
	Slope (degree)	18.4	Roof slope
	Ratio of PVT to SAH	6:4	-
Desiccant wheel	Diameter (m)	0.4	-
	Thickness (m)	0.2	-
	Rotation speed (r h ⁻¹)	12	-
	Desiccant	Silica gel (RD)	-
	Area ratio of regeneration side to process side	1:1	-
	Flow rate ratio of regeneration air to process air	1:1	-
PCM TES unit	Width (m)	0.44	Charvát, et al. (2014)
	Thickness of the PCM panel (mm)	20	Charvát, et al. (2014)
	Air channel (mm)	10	Dolado et al. (2010)
	Number of PCM layers	20	Based on the height of 0.6 m
	Rugosity (mm)	0.25	Dolado et al. (2010)
Indirect evaporative cooler	Wet-bulb temperature effectiveness	0.7	White et al. (2009)
	Flow ratio of secondary air to primary air	0.375:1.0	White et al. (2009)
Heat recovery unit	Effectiveness	0.8	White et al. (2009)

Table 7 Simulation design and simulation results - Scenario A

Length (m)	Air gap (mm)	Flow rate (kg h ⁻¹)	STC (%)		
			RT60	RT65	RT70HC
3	5.0	500	71.5	69.3	61.7
7	5.0	500	91.0	92.1	86.4
3	20.0	500	75.3	73.3	67.6
7	20.0	500	93.0	95.7	91.4
3	5.0	1000	57.9	52.2	33.3
7	5.0	1000	78.2	72.9	64.4
3	20.0	1000	61.0	56.7	36.9
7	20.0	1000	82.8	78.1	68.9
3	12.5	750	66.5	64.4	51.2
7	12.5	750	88.3	87.3	77.8
5	5.0	750	76.9	73.8	63.8
5	20.0	750	81.0	78.2	69.3
5	12.5	500	90.2	87.3	80.0
5	12.5	1000	73.6	69.8	55.0
5	12.5	750	80.9	78.0	69.0

Table 8 Coefficients of the response surface models - Scenario A

	b_0	b_1	b_2	b_3	b_{11}	b_{22}	b_{12}	R^2
RT60	54.7946	11.2848	1.0088	-0.0338	-0.7253	-0.0290	1.36E-3	0.9912
RT65	55.9970	11.7443	1.0088	-0.0420	-0.7253	-0.0290	1.36E-3	0.9943
RT70HC	51.0256	13.1424	1.0088	-0.0582	-0.7253	-0.0290	1.36E-3	0.9954

Table 9 Optimal designs identified for Scenarios A, B and C

	Scenario A	Scenario B	Scenario C
Length (m)	7.0	7.0	7.0
Air flow rate (kg h ⁻¹)	500	500	500
Air gap (mm)	17.4	16.9	16.4
PCM type	RT65	RT65	RT65
STC (RSM model)	95.2%	100.5%	85.6%
STC (confirmation test)	96.5%	100.0%	82.6%

Figure captions

Fig. 1 Schematic of a desiccant cooling system with integrated hybrid PVT-SAH and PCM TES unit.

Fig. 2 Scheme of the air-based PCM TES unit.

Fig. 3 Outline of the research method employed in this study.

Fig. 4 The configuration of the hybrid PVT-SAH.

Fig. 5 Schematic of the nodes in the modelling of the PCM TES unit.

Fig. 6 Comparison between simulation results and experimental data under the air flow rate of $240 \text{ m}^3 \text{ h}^{-1}$.

Fig. 7 Weather conditions of the selected consecutive five summer days in Brisbane.

Fig. 8 Enthalpy-temperature relationship of the PCM RT70HC (Rubitherm GmbH, 2016).

Fig. 9 Variation in STC with the changes of the key parameters under the regeneration temperature of 65°C .

Fig. 10 Results of the confirmation test using the optimal values identified - Scenario A.

Fig. 11 Supply air conditions under different regeneration temperatures.

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a) RT60 and flow rate of 500kg h⁻¹

b) RT65 and flow rate of 500kg h⁻¹

c) RT70HC and flow rate of 500kg h⁻¹

d) RT65 and air gap of 16.0 mm

Fig. 9 Variation in STC with the changes of the key parameters under the regeneration temperature of 65 °C.

a) Air temperature

b) Air humidity ratio

c) Thermal energy demand and supply

Fig. 10 Results of the confirmation test using the optimal values identified - Scenario A.

Fig. 11 Supply air conditions under different regeneration temperatures.