# Investigation of a Ground Coupled Photovoltaic Thermal Desiccant Cooling and Heating (GPVTDCH) System

Jinyi Guo, Jose I. Bilbao and Alistair B. Sproul

School of Photovoltaic and Renewable Energy Engineering, UNSW Sydney

Tyree Energy Technology Building (TETB) UNSW Sydney, Kensington NSW 2052, Australia

## Abstract

This study investigates the performance of a novel solar desiccant cooling and heating cycle, that uses ground water as the cooling source and photovoltaic thermal (PV/T) collectors as the heating source, to provide annual space air conditioning (fresh air ventilation, dehumidification, cooling and heating) for an office building in a humid subtropical climate of Sydney, Australia.

In the dehumidification and cooling mode, the hot and humid outdoor air is pre-cooled, dehumidified and cooled by the ground coupled PV/T desiccant cooling and heating (GPVTDCH) system before it is supplied to indoor. This process enables the desiccants to be regenerated by a low grade heat source (temperatures less than 60°C). In the heating mode, the cold outdoor air is pre-heated by the heat exchanger, and then been further heated by the flat plate PV/T collectors before passing directly to indoor.

To analyse the performance of the GPVTDCH system, a simulation model of the system and the office building was developed in TRNSYS. In addition, the energy consumption of fans and pumps were also sized and included in the TRNSYS model. This is an improvement to the previous work presented by the authors (Guo et al. 2014). Therefore, the performance of the GPVTDCH system can be better estimated for air ventilation, dehumidification, cooling and heating for an office building.

The GPVTDCH system annual system coefficient of performance (COP) and the reduction in the required heating and cooling energy consumption (sensible and latent) of the office building was evaluated for maintaining the indoor air conditions within the 20-26°C and 4-12 g $kg^{-1}$ specific humidity for 95% of the operating time in a year (Standards Australia 2016). The simulation results show that the GPVTDCH system can supply satisfied air conditions with a peak annual system cooling COP of 10.1 and system heating COP of 14.9, and up to 70% reduction in the required heating and cooling energy consumption for the building.

*Keywords: cooling, solar energy, photovoltaic thermal (PV/T) collectors, desiccant dehumidification and cooling, ground cooling, low temperature desiccants*

## 1. Introduction

The energy consumption of air conditioning systems accounts for 40% of the total energy consumption of a building with an increasing trend globally (Perez-Lombard et al. 2008). This becomes a global concern as the majority of the current commercially available air conditioning systems are powered by fossil fuel generated electricity.

Utilising solar energy as the main energy source to drive an air conditioning system is a long-term goal in the search for an alternative air conditioning solution (IEA 2016). Recently, some researchers investigated the utilisation of photovoltaic thermal technology for desiccant air conditioning systems. This further improves the system functionality in producing both air conditioning and electricity generation by the PV modules. Ultimately, a PV/T driven desiccant air conditioning system could provide space air conditioning (ventilation, heating and cooling, humidity control) and electricity generation to meet the building energy demand.

One of such an example is reported by Mei et al. (2006) and Eicker et al. (2010) on an installation of an air type PV/T desiccant air conditioning system in a library in Mataro (Spain). The system produced supply air temperatures between 15-17°C with an average of 5 g $kg^{-1}$ in dehumidification. It demonstrated the possibility


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of utilising solar energy to provide 84% of the system energy consumption. From a simulation analysis in TRNSYS, Beccali et al. (2009) showed that for two different occupant spaces (an office and a lecture room), a PV/T desiccant air conditioning system supplied air with temperatures of 21-24°C and an average moisture removal of 6-7 g $kg^{-1}$ with primary energy saving of 43%-55%. In the above studies, the regeneration temperatures were above 70°C from the use of solar thermal collectors. When PV/T collectors were used as the main heat source, low dehumidification performance was found. This was due to insufficient regeneration of the desiccants with heat source temperatures between 40-50°C from the PV/T collectors alone. Therefore, matching heat source temperatures from PV/T collectors to the desiccant regeneration process is critical to maximising the contribution of PV/T collectors as the main energy source in a solar desiccant air conditioning system. In addition, the space heating function was not described in detail in all of the above studies to show the overall benefits of utilising PV/T collectors in a desiccant air conditioning system.

For the full air conditioning function of a solar desiccant air conditioning system driven by solar thermal collectors, Hand et al. (2016) provided more insight on the heating performance. It was shown that the system can provide air ventilation, air-conditioning and domestic hot water heating. However, it was found that the system only saved 21% of the gas consumption for water heating and reduced 34% of the energy needed for cooling. Even though this study demonstrated the practical application of a solar water heating system with a desiccant air conditioning system (with direct evaporative cooling) to supply hot water and air conditioning to a building, further improvement is needed to increase the energy saving.

Recently, a study on a residential scale solar PV/T driven desiccant cooling prototype was reported by Finocchiaro et al. (2016). This prototype consisted of a desiccant packed bed of silica gel and an indirect evaporative cooling (IEC) unit. In summer, the system achieved satisfied dehumidification and cooling performance with a high energy efficiency ratio (EER) of 12.8. The integrated PV/T and solar thermal air collector was able to provide all the thermal energy required and 75% of the electrical energy during the cooling operation. In winter, the air from the PV/T collector was directly used for space heating. This system has much higher EER of 36.9 in which, the PV/T collector provided all the required thermal energy and 94% of the electrical energy for the building. This study highlighted the benefits of utilising PV/T driven desiccant air conditioning to achieve high performance in space cooling and heating.

From previous demonstrations, the feasibility of utilising PV/T technology in a desiccant air conditioning system in space heating and cooling application is shown. Further work is needed to assess the benefits and limitations of system performance of such a system in a building, in particular, the space heating function. In this study, a ground coupled PV/T desiccant air conditioning (GPVTDCH) system is investigated theoretically. The ambient air is pre-treated to enhance the desiccant air conditioning performance and allowing the use of PV/T collectors as the main heat source at low heat source temperatures. In addition, all required thermal and electrical energy for the system operation is provided by the flat plate PV/T collectors (refer to as "PV/T collector" from now on).

## 2. System Modeling and Method

### 3.1. GPVTDCH system description

The schematic diagram of the GPVTDCH system shown in Fig 1 outlines the air flow path for a) cooling and b) heating operations. As shown in Fig 1a. it consists of three sub-systems, including: (i) a PV/T water heating system (solid red line), (ii) a desiccant cooling and heating system (blue and red dash lines), and (iii) an open-loop ground water cooling and heating system (blue solid line).

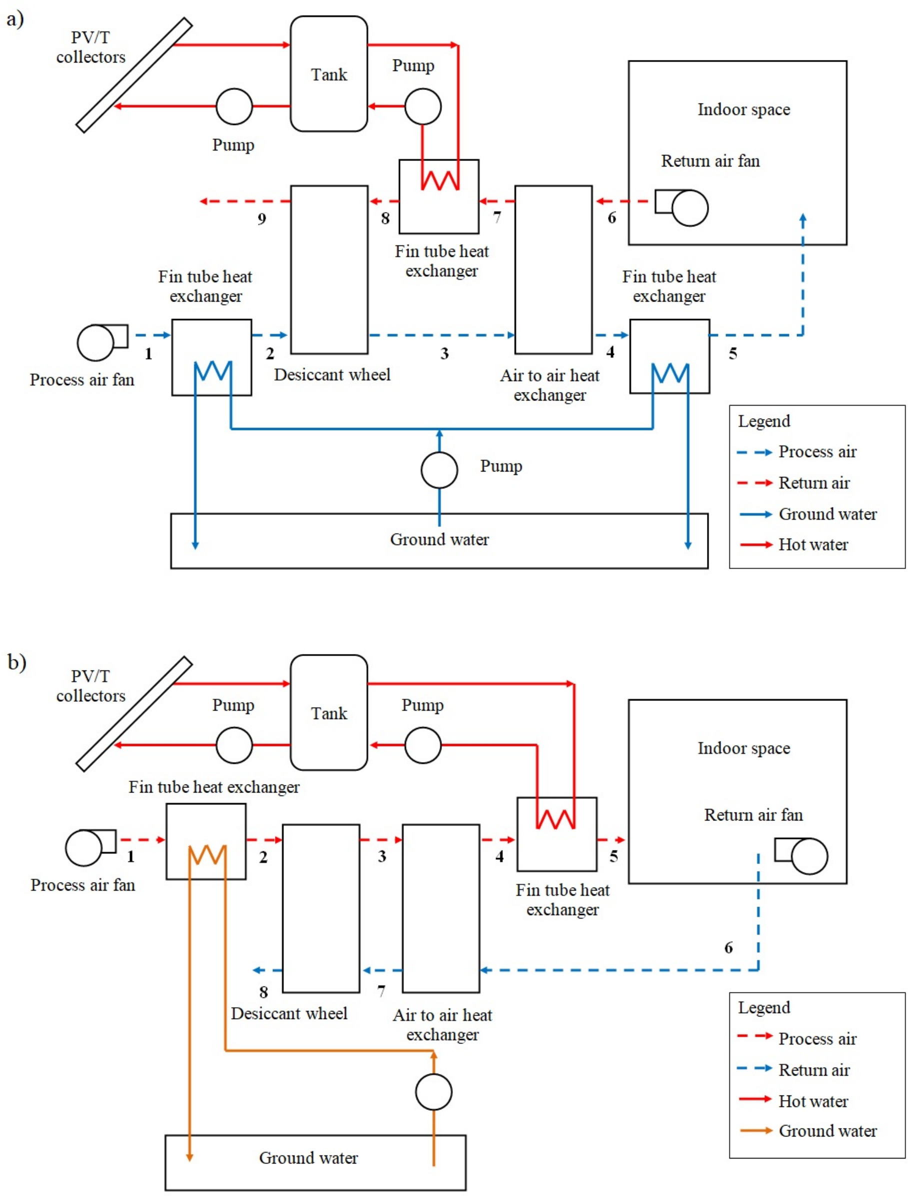


**Fig 1 - Air flow path on the schematic diagram of the GPVTDCH system a) for air dehumidification and cooling and b) for air heating operation**

During the cooling and dehumidification operation (shown in Fig 1a), the outdoor air in the process air stream (blue dash line) is pre-cooled via a ground water cooling system with a fin tube heat exchanger (1-2). The pre-cooled air is then dehumidified by a rotary solid desiccant wheel (2-3). After which the dehumidified air is cooled by a air to air heat exchanger (3-4) with the return air and then by the ground water cooling system (4-5). The cool and dry air is delivered to the space to condition the indoor air. On the other hand, in the return air stream (red dash line), the indoor air is heated by the air to air heat exchanger (6-7) and then the PV/T water heating system via the fin tube heat exchanger (7-8). The hot air (desiccant regeneration air) passes through the desiccant wheel to regenerate the desiccants (8-9).

During heating operation (show in Fig 1b), the outdoor air in the process air stream (red dash line) is pre-heated by the heat exchangers (1-4). It is further heated by the PV/T collector (4-5) for the space heating. The desiccant wheel is bypassed if no dehumidification is required.

### 3.2. System and building modelling

The GPVTDCH system was modelled in TRNSYS to evaluate its annual performance. A validated water type PV/T collector model from Bilbao and Sproul (2012), called Type 850, was used. For the desiccant wheel, the numerical model developed and validated by Goldsworthy and White (2011) was used. Both of the above component models have been validated against experiment measurements to represent the physical performance of the key components. For other components, the heat exchanger was modelled in Type 5, the ground water temperature was modelled by Type 501, the pump was computed in Type 110 and the fan was represented by Type 111 in TRNSYS.

The size of the GPVTDCH system components are outlined in Tab 1. The PV/T collector was designed to operate at a low mass flow rate per unit collector area, to deliver temperature water above 40°C for the desiccant regeneration process. For the sizing of the PV/T collector, a sensitivity analysis by Beccali et al. (2009) shows that the heating and cooling performance of a solar desiccant heating and cooling system increases with the increase in the collector area while the system energy consumption decreases. Therefore, a relative large PVT collector area was sized for the commercial office building which is similar to the size of the solar collector and building area studied by Beccali et al. (2009). Furthermore, the rotational speed of the desiccant wheel was designed at 10 RPH for the given desiccant wheel thickness of 0.2 m.

**Tab 1 - Summary of main system parameters**

| Component | Parameter | Values |
|---|---|---|
| Flat plate PV/T collector | Collector area ($m^2$) | 51 |
| | Tilt angle | 15 |
| | Mass flow rate per unit collector area ($kg\ s^{-1}\ m^{-2}$) | 0.01 |
| Desiccant wheel | Wheel thickness (m) | 0.20 |
| | Rotational speed (RPH) | 10 |
| | Desiccant | Silica gel |
| Heat exchanger | Characteristics | Fin tube type |
| | Dimension L x W x H (m) | 1 x 1 x 1 |
| Hot water pump | Type | Groundfos MAGNA |
| Bore water pump | Type | Groundfos SQE |
| Air fan | Type | PowerLine EC |
| Hot water storage tank | Capacity (L) | 7,000 |
| | Tank loss coefficient ($W\ m^{-2}\ K^{-1}$) from Cruickshank and Harrison (2010) | 1.26 |
| Other | Depth of the ground water (m) | 30 |

To control the GPVTDCH system operation, the PV/T water heating system and the ground coupled desiccant cooling system have independent controls. The PV/T water heating system has simple control logic. It operates when the inlet fluid temperature is 5°C lower than outlet fluid temperature and it turns off when the tank water temperature is 5°C higher than the outlet fluid temperature of the PV/T collector.

The GPVTDCH system has four operation modes as following:

- Cooling and dehumidification mode - air dehumidification and cooling
- Sensible cooling mode - air cooling only
- Veniltation mode - air ventilation via the heat exchanger

- Sensible heating mode - air heating only

The operation of each mode is controlled by both outdoor and indoor conditions. The sensible cooling mode is ON when either outdoor temperatures or indoor temperatures are greater than the upper temperature set point ($T_{set,up}$) and outdoor specific humidity and indoor specific humidity are lower than the upper specific humidity set point ($\omega_{set,up}$). The dehumidification and cooling mode is ON when either outdoor temperatures or indoor temperatures are greater than $T_{set,up}$ and the outdoor specific humidity and indoor specific humidity are greater than $\omega_{set,up}$. The sensible heating mode is ON when either outdoor temperatures or indoor temperatures are lower than the lower temperature set point ($T_{set,low}$) and outdoor specific humidity and indoor specific humidity are higher than the lower specific humidity set point ($\omega_{set,low}$). Tab 2 summarizes the operation and control.

**Tab 2 - Summary of GPVTDCH system operation and control**

| Mode | Ground loop | Heat exchanger | Desiccant wheel | PV/T | Condition |
|---|---|---|---|---|---|
| Cooling and dehumidification | On | On | On | On | $T > T_{set,up}$<br>$\omega > \omega_{set,up}$ |
| Cooling | On | On | Off | Off | $T > T_{set,up}$<br>$\omega < \omega_{set,up}$ |
| Ventilation | Off | On | Off | Off | $T_{set,low} < T < T_{set,up}$<br>$\omega_{set,low} < \omega < \omega_{set,up}$ |
| Heating | On | On | Off | On | $T < T_{set,low}$ |

To evaluate the feasibility of the GPVTDCH system as an HVAC system for a building, it is necessary to link it to a building model. A well-shaded office with 10 occupants and building was used. Following the requirements of the AS/NZS 1668.2:2012 (Standards Australia 2012), a minimum 100 m$^2$ floor area was thus calculated for the 10 occupants and it was assumed a ceiling height of 3 meters, for a total internal volume of 300 m$^3$. The office was assumed to operate from 8 am to 6 pm on Monday to Friday, excluding eleven public holidays in Australia, for a total of 249 days in a year. This schedule was applied to the building operation when calculating internal heat and moisture gains.

To model this simple office building in TRNSYS, Type 660a was considered. This modelled a simple lumped capacitance single zone structure subject to internal gains. It neglected solar gains and assumed an overall U value for the entire structure. The U-value and capacitance value of the building in Tab 3 was input to the Type 660a to present the performance of the modelled office space. Key parameters including building characteristics, internal heat gain due to infiltration, lighting, equipment and occupants, as well as moisture gain are summarised in Tab 3. For this model, the energy consumption to maintain the building at indoor temperatures between 20-26°C and specific humidity between 4-12 g/kg during operational hours was 62 kWh/m$^2$.annum. Note that using the National Australian Built Environment Rating System (NABERS), the modelled building corresponds to a 5-5.5 NABERS star, a rating that corresponds to the minimum design benchmark requirement for a new office building in Sydney. The building model was linked to the GPVTDCH system model in TRNSYS.

**Tab 3 - Summary of the simple building model for an office (all terms are defined in the Appendix)**

| | | | | |
|---|---|---|---|---|
| **Building characteristics** | Floor area (m$^2$) | 100 | Height (m) | 3 |
| | Surface (m$^2$) | 320 | Volume (m$^3$) | 300 |
| | Building capacitance (kJ K$^{-1}$) | 45780 | $U_{eff}$ (W m$^{-2}$ K$^{-1}$) | 0.5 |

| | | | | |
|---|---|---|---|---|
| **Infiltration** | Infiltration rate (ACH) (Egan) | 0.7 | $\dot{m}_{\text{infil}}$ (kg hr$^{-1}$) | 253.1 |
| **Lighting** | $\dot{q}_{\text{light}}$ (W m$^{-2}$) (ASHRAE 2013) | 10.6 | $\dot{Q}_{\text{light}}$ (W) | 1056 |
| **Equipment** | $\dot{q}_{\text{equip}}$ (W m$^{-2}$) (ASHRAE 2013) | 6.5 | $\dot{Q}_{\text{equip}}$ (W) | 1001 |
| **People** | $\dot{q}_{\text{people}}$ (W person$^{-1}$) (ASHRAE 2013) | 75 | $\dot{Q}_{\text{people}}$ (W) | 750 |
| **Humidity gain** | $\dot{q}_{\text{latent}}$(W person-1) (ASHRAE 2013) | 55 | $\dot{Q}_{\text{latent}}$(W) | 550 |
| | $\dot{\omega}_{\text{gain.person}}$ (kg hr$^{-1}$ person$^{-1}$) | 0.088 | $\dot{\omega}_{\text{gain}}$ (kg hr$^{-1}$) | 0.88 |

Sydney (latitude 33.9°S) with humid subtropical climate classified by Köppen climate classification was selected to evaluate the annual performance of the GPVTDCH system. According to the Meteonorm TMY2 data files from TRNSYS library, Sydney has an annual average temperature of 17.9°C and specific humidity of 9.2 g kg$^{-1}$. The TRNSYS model of the GPVTDCH system was simulated to provide air ventilation, dehumidification and cooling, heating in a year. The GPVTDCH system COP and the reduction in the thermal to maintain the the simulated office space 20-26°C and specific humidity between 4.3-12.1 g kg$^{-1}$, for more than 95% of the operating time, as specified in the AS5389:2016 (Standards Australia 2016) were analysed. The thermal (sensible and latent) load was calculated by Type 660a in TRNSYS.

## 3. Results and Discussion

### 3.1. Performance indicators

To evaluate performance of the GPVTDCH system, several different performance indicators were used. All terms are defined in the Appendix.

The annual system COPs for cooling and heating were defined by equation 1.

$$\text{COP}_{\text{cool}} = \frac{\text{Total cooling aenergy provided in a year}}{\text{Total auxiliary energy consumed in a year}} = \frac{\sum_1^n \dot{Q}_{\text{cool}}\Delta t}{\sum_1^n \dot{Q}_{\text{aux,total}}\Delta t} \quad \text{(eq. 1a)}$$

$$\text{COP}_{\text{heat}} = \frac{\text{Total heating energy provided in a year}}{\text{Total auxiliary energy consumed in a year}} = \frac{\sum_1^n \dot{Q}_{\text{cool}}\Delta t + \sum_1^n \dot{Q}_{\text{heat}}\Delta t}{\sum_1^n \dot{Q}_{\text{aux,total}}\Delta t} \quad \text{(eq. 1b)}$$

where

The cooling capacity ($\dot{Q}_{\text{cool}}$) presents the cooling power of the system, expressed as:

$$\dot{Q}_{\text{cool}} = \dot{m}_{\text{sa}}(h_{\text{am}} - h_{\text{sa}}) \quad \text{(eq. 2)}$$

The heating capacity ($\dot{Q}_{\text{heat}}$) presents the heating power of the system, expressed as:

$$\dot{Q}_{\text{heat}} = \dot{m}_{\text{sa}}(h_{\text{sa}} - h_{\text{am}}) \quad \text{(eq. 3)}$$

The total auxiliary power was calculated as:

$$\dot{Q}_{\text{aux,total}} = \dot{W}_{\text{pumps}} + \dot{W}_{\text{fans}} + \dot{Q}_{\text{reg.aux}} \quad \text{(eq. 4)}$$

In the study, the auxiliary heating energy $\dot{Q}_{\text{reg,aux}}$ for the desiccant regeneration process was assumed to be provided by a resistance heater with electrical COP of 1. $\Delta t$ is the hour where the GPVTDCH system is operating.

The reduction in the thermal (heating and cooling) load of the office space by the GPVTDCH system is calculated as:

$$\text{Thermal load reduction} = \frac{\sum_1^n \dot{Q}_{\text{office,sen,GPVTDCH,on}}\Delta t + \sum_1^n \dot{Q}_{\text{office,lat,GPVTDCH,on}}\Delta t}{\sum_1^n \dot{Q}_{\text{office,sen,GPVTDCH,off}}\Delta t + \sum_1^n \dot{Q}_{\text{office,lat,GPVTDCH,off}}\Delta t} \quad \text{(eq. 5)}$$

where $\dot{Q}_{\text{office,sen,GPVTDCH,off}}$ and $\dot{Q}_{\text{office,lat,GPVTDCH,off}}$ were calculated by Type 660a in TRNSYS. This type calculates the energy required to maintain the indoor conditions within the range of 20-26°C for indoor temperatures and 4.3-12.1 g kg$^{-1}$ for indoor specific humidity as been specified by the AS5389:2016 (Standards Australia 2016). The office building model was simulated with and without the operation of the GPVTDCH system to compare the reduction in the heating and cooling energy consumption required in a year.

### 3.2. Air heating, dehumidification and cooling process by the GPVTDCH system

To understand the air conditioning process of the GPVTDCH system in air cooling and heating operation, the air conditions (dry bulb temperatures and specific humidity) after various components are shown on the psychrometric diagram.

Fig 2 shows the air conditions conditioned by the GPVTDCH system at an air flow rate of 3,000 kg hr$^{-1}$ during the air dehumidification and cooling mode on a psychrometric diagram. The blue line represents the process air stream and the red line represents the return or desiccant regeneration air stream. The black line is the boundary where the office indoor air condition is required to be kept between 20-26°C and 4.3-12.1 g kg$^{-1}$ at 95% of the time during the occupancy according to AS5389:2016 (Standards Australia 2016). The detail air conditions after each process are listed in air conditions in Tab 4.

As shown in Fig 2, the ambient air (1) with a temperature of 31.1°C is pre-cooled by the ground water to a temperature of 20.8°C. This increases the relative humidity of the inlet air to the desiccant wheel, which allows the desiccants to be regenerated at low heat source temperature of 47.2°C heated by the PV/T water circulation at (8). In comparison to a conventional desiccant dehumidification process, the pre-cooling of the entry air to the desiccant wheel can reduce the temperatures in the dehumidification process. This enhances the dehumidification performance. In addition, it also increases the relative humidity of the dehumidified air from the desiccant wheel. Since the partial pressure difference between the process air stream and the regeneration air stream need to be balanced between the dehumidification and the regeneration process, thus, lower heat source temperatures can be used for the regeneration process. Therefore, in the GPVTDCH system, the entry air to the desiccant wheel is pre-cooled by the ground water, thus low regeneration temperatures can be used in comparison to a conventional desiccant dehumidification process (Jeong et al 2011, Zhang and Niu 2003).

Nevertheless, the dehumidification process removes 5.9 g kg$^{-1}$ of moisture from the air in the desiccant wheel. In comparison to previous studies with desiccant regeneration temperatures of 70°C (Mei et al. 2006, Eicker et al. 2010, Beccali et al. 2009), the heat source temperature of GPVTDCH system to achieve the similar dehumidification performance is significantly less. In addition, it can be seen that the GPVTDCH system can provide supply air at 19.7°C and 7.6 g kg$^{-1}$, suitable for space cooling application.

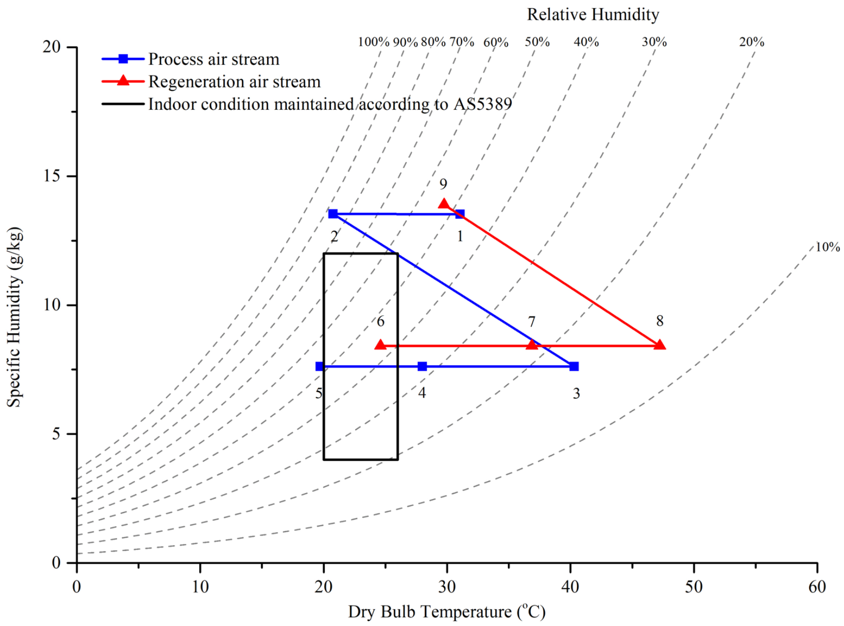


**Fig 2 - Psychrometric process of the GPVTDCH system on the psychrometric diagram for a summer day condition of Sydney (the subscript number corresponds to the process in Fig 1a)**

**Tab 4 - Air conditions of the GPVTDCH system operation for a summer day condition of Sydney (the subscript number corresponds to the process in Fig 1a)**

| Air conditions | Temperature (°C) | Specific humidity g $kg^{-1}$ |
|---|---|---|
| **Process air stream** | | |
| Ambient air (1) | 31.1 | 13.5 |
| Desiccant wheel inlet air (2) | 20.8 | 13.5 |
| Dehumidified air (3) | 40.3 | 7.6 |
| Cooled air by air to air heat exchanger (4) | 28.0 | 7.6 |
| Supply air (5) | 19.7 | 7.6 |
| **Regeneration air stream** | | |
| Indoor air (6) | 24.6 | 8.4 |
| Pre-heated air by air to air heat exchanger (7) | 36.9 | 8.4 |
| Heated air by PV/T water heating circulation (8) | 47.2 | 8.4 |
| Exhaust air from the desiccant regeneration process (9) | 29.8 | 13.9 |

Fig 3 shows the air conditions conditioned by the GPVTDCH system at an air flow rate of 3,000 kg $hr^{-1}$during the air heating mode on a psychrometric diagram. The red line represents the return or desiccant regeneration air stream. The black line is the boundary where the office indoor air condition is required to be kept between 20-26°C and 4.3-12.1 g $kg^{-1}$ at 95% of the time during the occupancy according to AS5389:2016 (Australia Standard 2016). The detailed air conditions after each process are listed in Table 5.

In the air heating process, as shown in Fig 3, the ambient air (1) with a temperature of 15.5°C is pre-heated to a temperature of 22.6°C after the air to air heat exchanger (4). The process air is further heated by the PV/T water circulation to a supply air (5) temperature of 31.3°C. Since it is sensible heating process, there is no change in the specific humidity of the air.

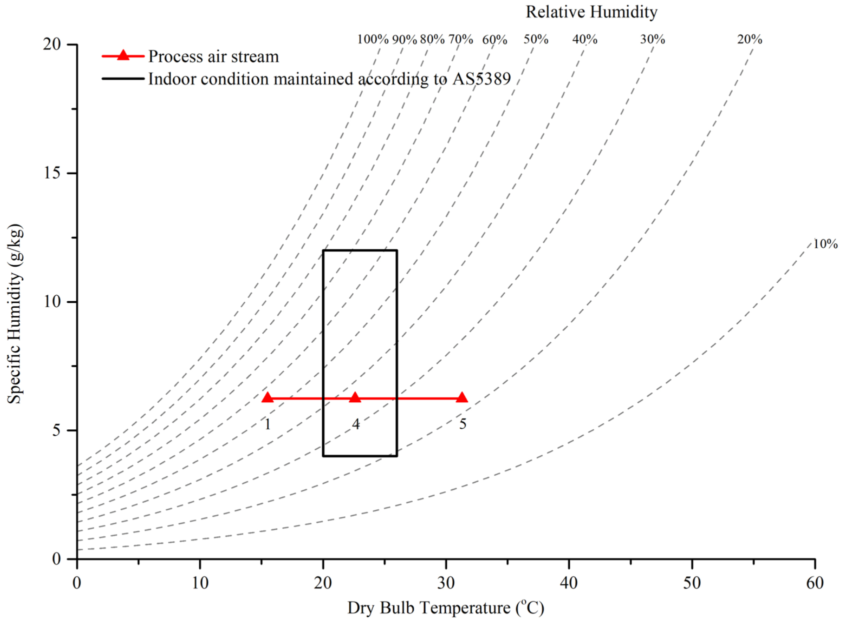


**Fig 3 - Psychrometric process of the GPVTDCH system on the psychrometric diagram for a winter day condition of Sydney (the subscript number corresponds to the process in Fig 1b)**

**Tab 5 - Air conditions of the GPVTDCH system operation for a winter day condition of Sydney (the subscript number corresponds to the process in Fig 1b)**

| Air conditions | Temperature (°C) | Specific humidity g kg$^{-1}$ |
|---|---|---|
| Ambient air (1) | 15.5 | 6.2 |
| Pre-heated air (4) | 22.6 | 6.2 |
| Supply air heated by PV/T water circulation (5) | 31.3 | 6.2 |

Overall, the above results provide a snapshot of the air dehumidification and cooling process in Fig 2 and the air heating process in Fig 3 on a psychrometric diagram. It can be seen that the GPVTDCH system can provide satisfactory supply air temperatures and specific humidity to condition the indoor space. In the next section, a more detailed analysis of the system's performance over a year is presented.

## 3.3. GPVTDCH system performance

To evaluate the energy performance of the GPVTDCH system as an air conditioning system, the coefficient of performance (COP) defined in equation 5 was used. Figure 4 shows the annual cooling COP (blue line), and heating COP (red line) of the GPVTDCH system under various flow rates from 1,000 to 7,000 kg hr$^{-1}$.

As shown in Fig 4, there is a peak in the annual system cooling COP and heating COP. This is because at a low process air flow rate (i.e. 1,000 kg hr$^{-1}$), the cooling and heating capacity of the GPVTDCH system is low as defined by equation 2 and equation 3. However, at a high process air flow rate (i.e. 7,000 kg hr$^{-1}$), the electricity consumption of the GPVTDCH system defined by equation 4 is also high. This results in a peak cooling COP value of 10.1 and heating COP value of 14.9 at the process air flow rates between 3,000 and 5,000 kg hr$^{-1}$. In addition, the annual heating COP is higher than the annual cooling COP. This is because the cooling source temperature is limited by the ground water temperature of the location at 17.9°C, whereas the heating source temperature of the PV/T collector is much higher at maximum of 58.2°C. Nevertheless, in comparison to a common compression air conditioning system with system COP of 3-6 (Otanicar et al 2012), the system COP of the GPVTDCH system is much higher. In addition, the electricity and thermal energy required to operate the

GPVTDCH system can be all provided by the PV/T collectors.

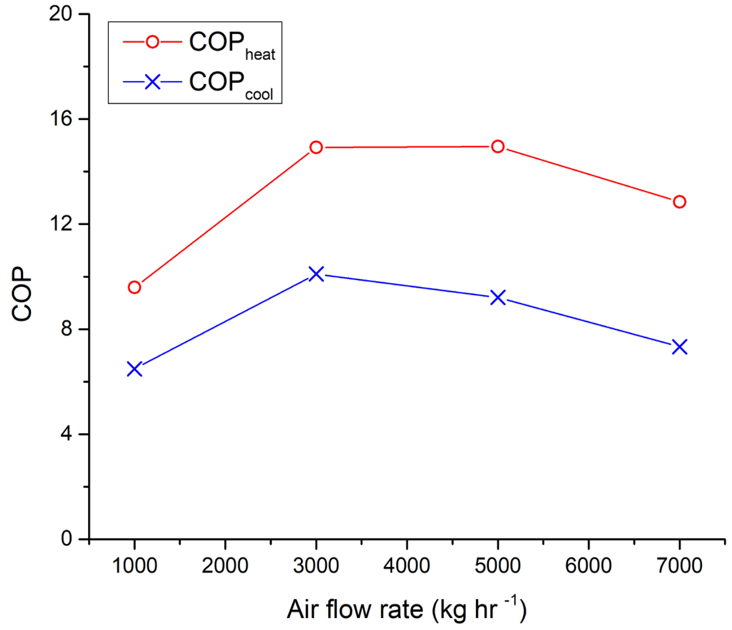


**Fig 4 - System COPs of GPVTDCH system at various process air flow rates**

To analyse the reduction in the total (sensible and latent) heating and cooling energy consumption required to maintain the indoor air condition within 20-26°C and 4-12 g kg$^{-1}$ achieved by the use of GPVTDCH system in a year, Fig 5 compares the percentage of total heating and cooling energy consumption reduction at various process air flow rates. It can be seen in Fig 5 that the reduction in the sensible heating and cooling energy consumption of the office building increases as the process air flow increases. This is because the heating and cooling capacity of the GPVTDCH system increases with the increase in the process air flow rate. Therefore, more heating and cooling energy is provided by the GPVTDCH system to offset the required thermal load. In contrast, the reduction in the latent load of the office building decreases as the air flow increases. This is because the dehumidification performance of the desiccant wheel is limited. At a high process air flow rate, there is insufficient time for the heat and mass exchange process between the air and the desiccants inside the desiccant wheel. The total reduction in the total thermal load of the office building is from 40% to 70% as the air flow rate increases from 1,000 kg hr$^{-1}$ to 7,000 kg hr$^{-1}$. This needs to be further improved by investigating appropriate controls of the process air flow rate and supply air conditions.

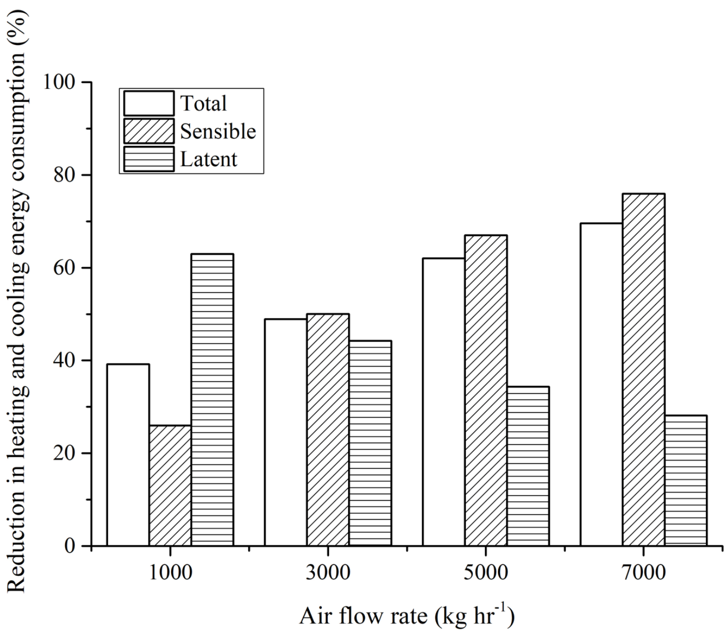


**Fig 5 - Reduction in required heating and cooling load (total, sensible and latent) to maintain the office building within 20-26°C and 4-12 g kg$^{-1}$ at various process air flow rates**

## 4. Conclusions

This study investigated the performance of a ground coupled photovoltaic thermal desiccant cooling and heating (GPVTDCH) system to provide air ventilation, dehumidification, cooling and heating for an office building in a humid subtropical climate of Sydney, Australia.

The GPVTDCH system was modelled and linked to a building model in TRNSYS. The performance of the GPVTDCH system in maintaining the indoor conditions between 20-26°C and 4-12 g kg$^{-1}$ at 95% of the operation time during the occupancy were evaluated. The key findings are:

- The GPVTDCH system can provide satisfactory supply air conditions. On a typical summer day with ambient air conditions of 31.1°C and 13.5 g kg$^{-1}$, in the cooling operation, the supply air can be dehumidified and cooled to 19.7°C and 7.6 g kg$^{-1}$ (5.9 g kg$^{-1}$ for dehumidification). While on a typical winter day with ambient air conditions of 15.5°C and 6.2 g kg$^{-1}$, in the heating operation, the supply air can be heated to 31.3°C and 6.2 g kg$^{-1}$.
- High annual system cooling COP of 10.1 and system heating COP of 14.9 were modelled for air flow rates between 3,000 kg hr$^{-1}$ and 5,000 kg hr$^{-1}$. This is higher than a common compression air conditioning system
- The GPVTDCH system can reduce 70% of the required heating and cooling energy consumption to maintain the office building with the designed air conditions AS5389:2016 (Standards Australia 2016)

Further study is required to improve the building model with consideration of solar gain and compare to a long term data of a physical building performance. In addition, an appropriate control algorithm for the process air flow rate and supply air conditions need to be investigated to improve the performance of the GPVTDCH system to provide electricity and air conditioning to a building.

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# 7. Appendix

## 7.1. Nomenclature

| | | | |
|---|---|---|---|
| $\dot{m}$ | Mass flow rate (kg hr$^{-1}$) | *Subscript* | |
| $\dot{q}$ | Rate of energy per unit of area (W m$^{-2}$) | aux | Auxiliary |
| $\dot{Q}$ | Rate of energy (W) | cool | Cooling |
| t | Time (hr) | equip | Equipment |
| $T$ | Temperature (°C) | heat | Heating |
| $U_{\mathrm{eff}}$ | Overall heat transfer coefficient (W m$^{-2}$ K$^{-1}$) | infil | Infiltration |
| $\omega$ | Specific humidity (g kg$^{-1}$) | lat | Latent |
| $\dot{\omega}$ | Rate of moisture change (kg hr$^{-1}$) | light | Lighting |
| $\dot{W}$ | Power (W) | low | Lower |
| | | sen | Sensible |
| *Abbreviations* | | set | Setting |
| EER | Energy efficiency ratio | up | Upper |
| COP | Coefficient of performance | | |
| GPVTDCH | Ground coupled PV/T desiccant cooling and heating | | |
| PV/T | Photovoltaic thermal | | |