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Ground Coupled Photovoltaic Thermal (PV/T) Driven Desiccant Air Cooling

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Abstract

The photovoltaic thermal (PV/T) driven desiccant air cooling process could be a good solution to the conventional air cooling cycle in terms of energy saving, where the latent cooling load would be removed adiabatically. However, problems exist where the required desiccant regeneration temperature (60°C - 80°C) often exceeds the outlet fluid temperature from standard PV/T collectors. In addition, operating the PV/T collector at high temperature reduces its electrical performance and lifetime. This study presents a desiccant air cooling process coupled with PV/T water heating cycle and a ground coupled cooling cycle which solves the above limitation. The inclusion of the ground coupled cooling increased the inlet air relative humidity and reduced the required regeneration temperature.

It was found that the required regeneration temperature increased with higher inlet air temperature and specific humidity for both conventional and ground coupled desiccant cooling cycle with pre-cooling. In comparison to the conventional desiccant cooling process, the ground coupled desiccant cooling process had lower regeneration temperature requirement and would be less affected by the inlet air conditions. Furthermore, low water flow rates would be preferred with higher outlet fluid temperature to suit the desiccant regeneration.

With defined thermal coefficient of performance (COP) and solar fraction (SF) as the main performance indicators, this ground coupled PV/T desiccant air cooling cycle was simulated under the Sydney summer climate condition in January. The results showed that the ground coupled PV/T desiccant cooling process could supply air temperature ranged between 21°C to 23°C with a constant designed specific humidity of 0.008 kg/kg. Furthermore, the climate conditions would strongly affect both the thermal COP and the solar fraction, thus auxiliary heating would be needed.

Nomenclature

A_c	collector area (m^2)
C_p	specific heat capacity (J/kg.K)
$F_{1,i}$, $F_{2,i}$	combined characteristic potentials
F_R	heat removal factor
F’	collector efficiency factor
G	solar radiation intensity (W/m^2)
h	specific enthalpy (kJ/kg)
$\dot{m}$	mass flow rate (kg/s)
n	Efficiency
n_{F1}, n_{F2}	characteristic effectiveness
$\dot{Q}$	rate of heat transfer (W)
T	temperature (°C)
$\tilde{T}_p$	mean plate temperature (°C)
U_L	overall heat loss coefficient ($W/m^2.K$)
w	specific humidity (kg/kg)

Greek Symbol

α	absorptance
β	the temperature coefficient
ε	heat exchanger effectiveness
ξ	packing factor
τ	transmittance

Subscript

am	ambient
c	cold fluid
e	electrical
h	hot fluid
in	inlet
o	outlet
reg	regeneration
e_STC	PV cell standard test condition
u	thermal

1. Introduction & Literature Review

The desiccant air cooling process could be an energy efficient solution where the latent heat is removed by the desiccants adiabatically through the adsorption process and the major energy inputs are for desiccant regeneration and fluid circulation. Many researchers have investigated the desiccant air cooling cycle theoretically and experimentally (La et al., 2010). Figure 1 shows a conventional open cycle desiccant air cooling. In the process, the outdoor air is firstly dehumidified through a desiccant wheel (1-2) and becomes hot due to the carry over and the adsorption heat. It is then cooled down via a heat exchanger (2-3) and an evaporative cooler (3-4) before entering the indoor space. On the return air side, the room air is pre-cooled by a different evaporative cooler (5-6) and cools the supply air in the heat exchanger (6-7). This air is then heated by a heat source (7-8) to regenerate the desiccants (8-9). It is evident that the major input energy to this process is heating the return air for the desiccant regeneration. Panaras et al (2010) simulated and experimentally measured the performance of this cycle with simple mathematical equations. They reported that with both humidifiers turned on, the temperature and specific humidity of the supply air were in the range between 15 ^{o}C to 20 ^{o}C and 0.008-0.012 kg/kg.

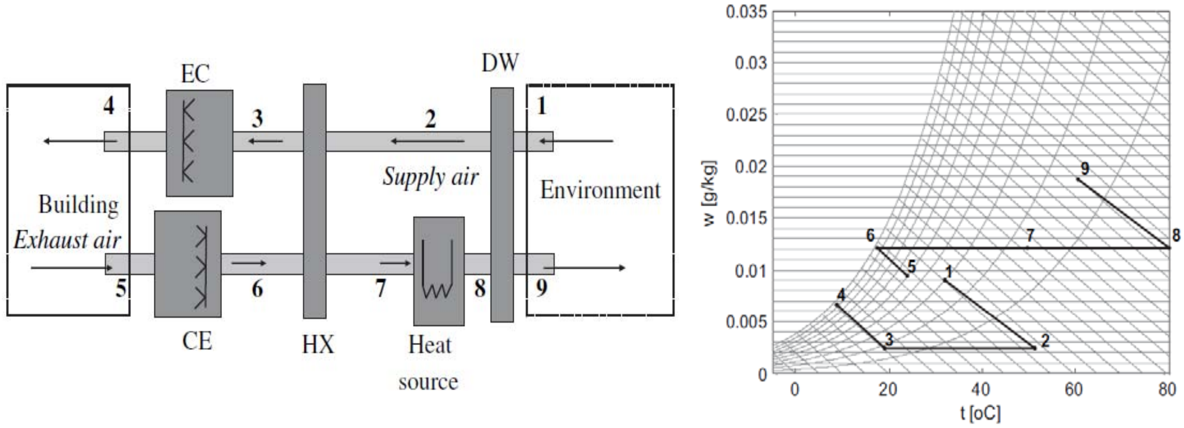

Figure 1 – A conventional desiccant air cooling cycle schematic (left) and psychrometric process (right) (Panaras et al., 2010)

A solar desiccant air cooling process works on the principle of the conventional desiccant air cooling in which the solar thermal collectors are used to drive the desiccant regeneration. Eicker et al (2010) studied a desiccant air cooling system installed in the Motaro Library in Spain. The system was coupled with both PVT air collectors and solar thermal collectors to supply regeneration temperatures between 50 ^{o}C to 70 ^{o}C. From the data monitoring, the calculated solar fraction at a constant 9,000 m^3/hr was 84%. This indicated a good potential of utilising solar energy for the desiccant air cooling process. However, the high regeneration temperature resulted in a lower thermal COP of 0.65 – 0.73 during the summer months. Furthermore, the carry over heat and adsorption heat were the two main factors limiting the dehumidification performance (Goldsworthy and White, 2012).

Theoretically, a PV/T collector can achieve higher energy efficiency than a PV module or a solar thermal collector alone (He et al., 2011). Zondag (2002) showed that the 1D steady state model provided a good estimation on the annual performance of the PV/T collector with much faster computational time to suit an initial stage analysis. Furthermore, Sproul et al (2012) investigated the analogy approach to simplify the modelling and calculation process for the solar thermal collector which could be extended to describe the PV/T collector. In addition, Bilbao (2012) pointed out that higher accuracy could be achieved in modelling the performance of a PV/T water collector when considered the sky temperature separately from the ambient air.

Pre-cooling enhances dehumidification. For the same amount of moisture removal by the desiccant wheel, the desiccant regeneration temperature requirement for the desiccant cooling cycle with pre-cooling would be lower than the cycle without pre-cooling. This is because, by pre-cooling the desiccant wheel inlet process air, the outlet dehumidified process air would have higher relative humidity. Since in the regeneration process, the required desiccant regeneration air temperature has to be heated a least to a point which its relative humidity is below the relative humidity of the outlet dehumidified process air to absorb the moisture from the desiccants, pre-cooling would effectively lower the desiccant regeneration temperature.

Ground coupled cooling of the process air stream is a good option instead of the evaporative cooler as there is no moisture addition to the process air stream. Thus, the required dehumidification load could be minimised which reduces the regeneration temperature requirement. Florides and Kalogirou (2007) have reviewed different ground coupling technologies in which vertical and horizontal ground heat exchangers were found to be the common options that could be applied in the desiccant cooling process. Furthermore, the ground water could be extracted directly for the heat exchanging process throughout the year. Eicker and Vorschulze (2009) showed the benefit of the ground coupled heat exchangers in the air conditioning operations where the ground temperature could be utilised to cool the space through both sensible air cooling and radiant floor cooling.

This study investigates the performance of the PV/T driven desiccant air cooling application coupled with ground cooling. The benefit of pre-cooling on reducing the regeneration temperature requirement under different inlet air conditions was compared to the conventional desiccant cooling process. The potential of a ground coupled desiccant air cooling process driven by PV/T water heating cycle is also studied.

2. Investigated Process Description

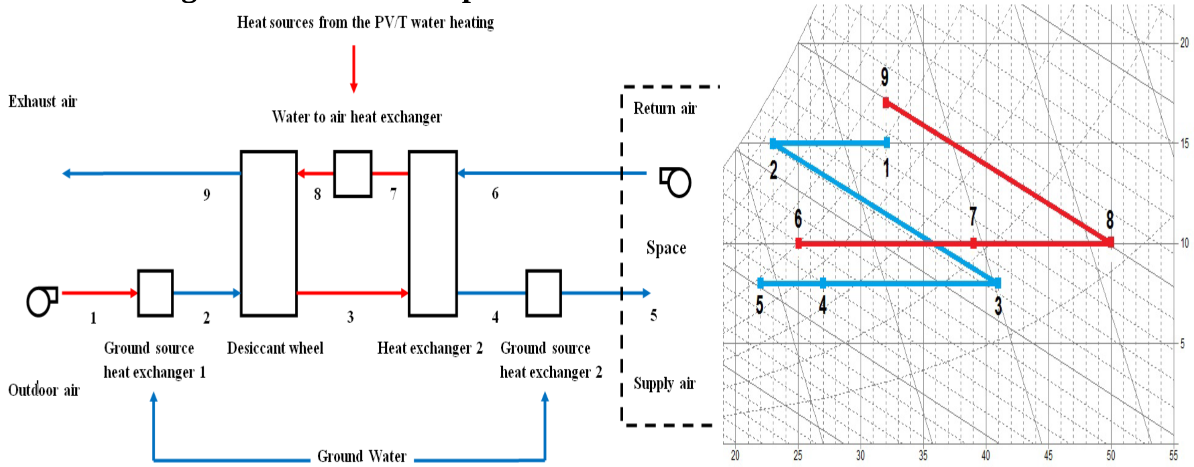

Figure 2 – Ground coupled PV/T desiccant air cooling cycle schematic (left) and psychrometric process (right)

Figure 2 shows the schematic of the proposed ground coupled PV/T water driven desiccant air cooling process. Under normal operation, the outdoor hot air is pre-cooled through the ground coupled heat exchanger 1 (1-2). The supply air passes through the desiccant wheel for dehumidification where the outlet air temperature increases (2-3). This air is then cooled down through the heat exchanger (3-4) and the ground coupled heat exchanger 2 (4-5) to be supplied to the indoor space. Mixture of the supplied air and the space air becomes the return air (5-6). The return air passes through the heat exchanger where it absorbs the heat from the supply air stream (6-7). Then it is further heated by the PV/T water heating cycle (7-8) and regenerates the desiccant (8-9). In the PV/T water heating cycle, the PV/T collector is connected to the hot water storage tank. The auxiliary heater is used as a backup system to ensure that the outlet water temperature meets the desiccant regeneration temperature requirement. In the ground cooling process, the cooled bore water is pumped up from the ground and passed through the heat exchanger to cool down the supply inlet air. This process is also shown on a psychrometric chart.

For the ground coupled heat exchanging cycle, further modelling is needed to explore the potential for winter heating. As found by Li et al (2006), the heat extraction and heat rejection to the ground should be balanced for the system to work sustainably.

3. Mathematical Model

3.1. Desiccant wheel

The desiccant wheel was modelled by the analogy approach as follows (Juranick, 1982):

$$n_{F1} = \frac{F_{1,3} - F_{1,2}}{F_{1,8} - F_{1,2}} \text{ and } n_{F2} = \frac{F_{2,3} - F_{2,2}}{F_{2,8} - F_{2,2}}$$

$$F_{1,i} = -\frac{2865}{T^{1.49}} + 4.344w^{0.8624} \text{ and } F_{2.i} = \frac{T^{1.49}}{6360} - 1.27w^{0.07969}$$

For these equations, *i* represents the air state as:
i = 2, inlet process air to the desiccant wheel
i = 3, outlet process air from the desiccant wheel
i = 8, inlet regeneration air to the desiccant wheel

The two constant dimensionless effectiveness value of n_{F1} and n_{F2} were used to characterise the performance of the silica desiccant wheel and expressed by the functions of the combined potentials of F1,i and F2,i for each air state point of the desiccant wheel. The two combined potentials of F1,i and F2,i are a function of the temperature and humidity ratio at a air state point of the desiccant wheel.

This study assumed a moderate desiccant wheel performance where n_{f1} and n_{f2} equals to 0.08 and 0.8 as suggested by Panaras et al (2010). By setting the specific humidity of the supply output air and return inlet air at 0.008 kg/kg and 0.010 kg/kg respectively, the inlet and outlet temperature from the desiccant wheel could be found through an iterative process.

3.2. PV/T water collector

For a PVT collector, the rate of energy outputs could be expressed from Duffie and Beckman (2006) and Florschuetz (1979) under the steady state as:

$$\dot{Q}_e = A_c \xi \tau G n_{e\_STC}[1 - \beta_{STC}(\tilde{T}_p - T_{STC})]$$

$$\dot{Q}_u = A_c F_R\left[\left(\alpha - n_{e\_STC}\xi\right)\tau G - U_L(T_{in} - T_{am})\right]$$

Where

$F_R = \frac{\dot{m}C_p}{A_c U_L}\left[1 - exp\left(-\frac{A_c U_L F'}{\dot{m}C_p}\right)\right]$ known as the heat removal factor and F' is the collector efficiency factor.

U_L is the overall heat loss coefficient which accounted for the heat loss through the top (radiative and convective), edge and bottom of the collector.

The fluid temperature change through the PV/T water collector could be derived from Bambrook and Sproul (2012) as:

$$\Delta T = (T_o - T_{in}) = \frac{A_c F_R}{\dot{m}C_p}[(\alpha - n_{el}\xi)\tau G - U_L(T_{in} - T_{am})]$$

3.3. *Ground coupled heat exchangers and heat exchangers*

The ground coupled heat exchangers and heat exchangers are modelled with the effectiveness approach as (Cengel, 2002):

$$\varepsilon = \frac{\dot{Q}_{c\ or\ h}}{\dot{Q}_{max}} = \frac{\dot{m}_c C_{p,c}(T_{c,o} - T_{c,in})}{(\dot{m}C_p)_{min} \times (T_{h,in} - T_{c,in})} = \frac{\dot{m}_h C_{p,h}(T_{h,in} - T_{h,o})}{(\dot{m}C_p)_{min}(T_{h,in} - T_{c,in})}$$

In this study, a heat exchanger effectiveness of 0.85 is assumed based on the experimental measurement by Panaras et al (2010). The heat exchanger is the most effective way to cool the process air stream. Therefore, this component would be designed to its maximum potential to reduce the work by other component such as the ground coupled cooling.

For the ground coupled heat exchanger, as the ground water temperature varies for a particular site, the ground water temperature will be site specific. For this initial analysis, inlet water temperature of 20°C was assumed.

3.4. Performance indicators

Two indicators were used to evaluate the performance include thermal COP and SF defined as (Mei et al., 2006):

$$COP = \frac{\dot{m}_{supply}(h_{in,supply} - h_{o,supply})}{\dot{m}_{reg}(h_{in,reg} - h_{o,reg})}$$

$$SF = \frac{\dot{Q}_u}{\dot{Q}_{regen,heat}}$$

3.5. *Simulation method*

TRNSYS software was used to simulate the performance of the PV/T desiccant air cooling process shown in Figure 2 in January under the Sydney climate data. The model was built and simulated under the steady state of the supply air and the return air conditions. Table 1 summarised important input parameters for different major components.

Table 1 - Inputs parameters for the major components in the TRNSYS simulation

Component	Type	Parameter	Value
Desiccant wheel	Type 683	Humidity ratio set point	0.008 kg/kg
Fan	Type 112a	Air flow rate	0.1 kg/s
Ground heat exchanger	Type 557a	Inlet fluid temperature	20 °C
Heat exchanger	Type 91, Type 760b	Heat exchanger effectiveness	0.85
Hot water tank	Type 4b	Tank volume	300 L
		Auxiliary heating mode	0
Pump	Type 3b	Inlet mass flow rate	0.02 kg/s
PV/T water collector	Type 50b	Collector area	2 m^2
Return air		Temperature	25 °C
		Specific humidity	0.01 kg/kg

4. Result and Discussion

4.1. Ground coupled desiccant cooling process and conventional desiccant cooling process comparison

4.1.1. Effect of inlet air temperature on the regeneration temperature required

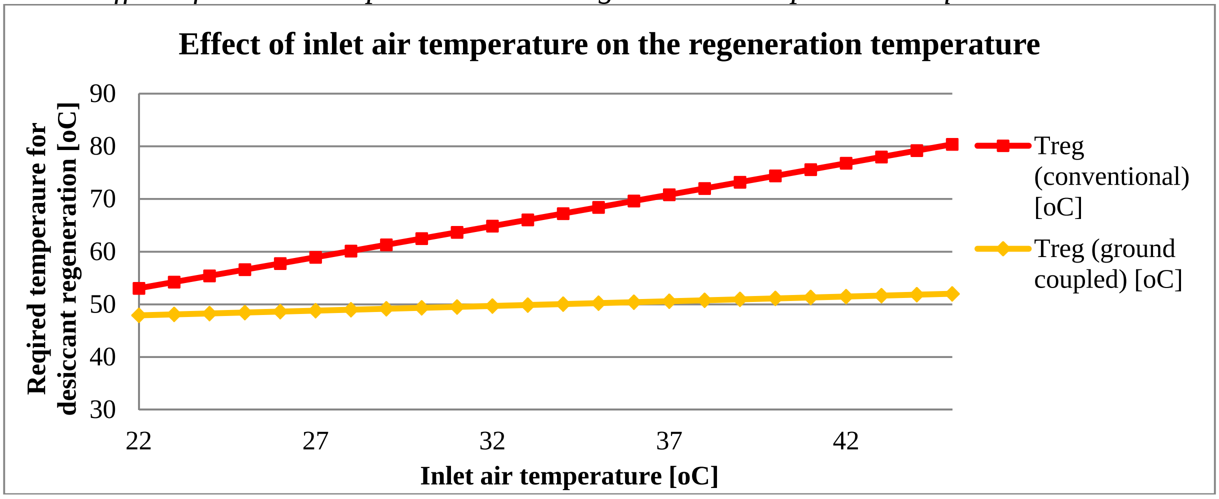

Figure 3 – Effect of inlet air temperature on the regeneration temperature (ground coupled vs. conventional)

By varying the inlet air temperature from 22 °C to 45 °C while keeping the inlet specific humidity at 0.015 kg/kg and other parameters constant, Figure 3 shows that the increase in the inlet air temperature increased the regeneration temperature. The comparison between the ground coupled and conventional process revealed that the regeneration temperature required for the ground coupled process was less affected by the inlet ambient air temperature. This highlighted the importance of pre-treating the inlet air in the process.

4.1.2. Effect of inlet air specific humidity on the regeneration temperature required

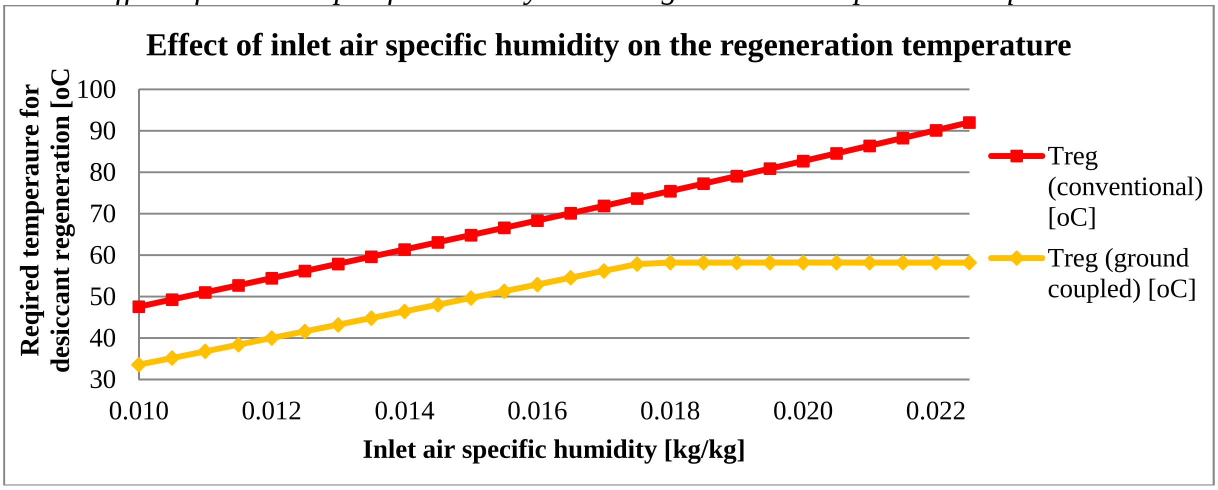

Figure 4 – Effect of inlet air specific humidity on the regeneration temperature (ground coupled vs. conventional)

From figure 4, the increase of the inlet air specific humidity at the constant inlet air temperature of 32°C resulted in higher required regeneration temperature as the dehumidification load became larger. Comparison between the ground coupled and conventional process indicated that the ground coupled process would limit the regeneration temperature at a certain inlet air specific humidity. When the inlet specific humidity increased beyond certain inlet specific humidity i.e. 0.017 kg/kg, the required regeneration temperature became constant. This was due to the pre-cooling process where the air with high specific humidity was cooled below its dew point and the moisture would be condensed out.

4.2. PV/T collector operation

The PV/T collector was used to drive and offset the thermal energy input to the desiccant regeneration. It is ideal to operate the PV/T at lower temperature to increase its electrical energy output. However, as shown from the previous Figure 3 and Figure 4, the required regeneration temperature was above 50 °C. Therefore, it is important to operate the PV/T collector to suit the desiccant regeneration process.

4.2.1. Effect of mass flow rate on PV/T outputs

Controlled inlet water mass flow rate is an important parameter to optimise the outputs from the PV/T collector to couple with the desiccant regeneration process. It could be seen from Figure 5 that the increase of the inlet mass flow rate enhances both the electrical and thermal outputs. However, the outlet fluid temperature was reduced. Therefore, the flow rate should be kept low to regenerate the desiccant regeneration while high flow rate would be preferred for high electrical and thermal outputs. This result agreed with Bambrook and Sproul's finding (2011) who also highlighted the importance to optimise the mass flow rate with consideration of the fan power consumption. In this study, a mass flow rate of 0.02 kg/s was used as near 40 °C outlet fluid temperature could be achieved. Optimisation of the flow rate requires further research in the future with consideration of the pump power and variable speed to suit the operation of the desiccant wheel.

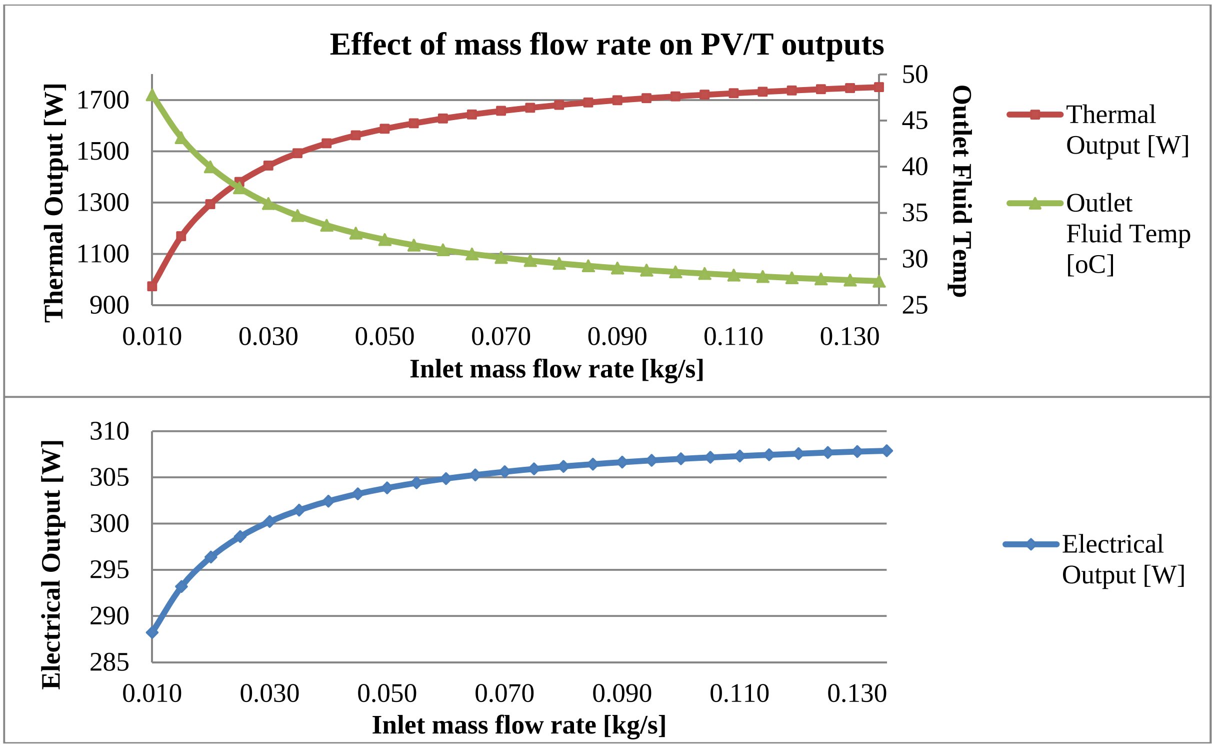

Figure 5 – Effect of mass flow rate on PV/T outputs

4.3. Simulated performance under Sydney climate in January

Figures 6-8 show the simulation results of the daily average supply air temperature, thermal COP and solar fraction of the ground coupled PV/T desiccant cooling process with parameters summarised in Table 1. The daily operation schedule was set between 9am to 5pm for the simulated month of January.

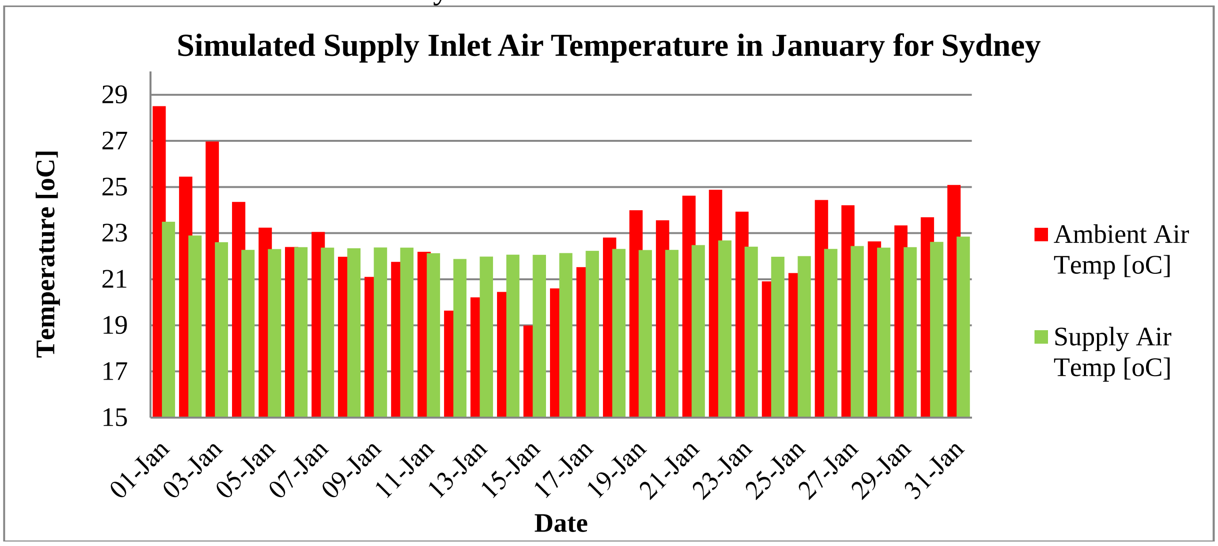

Figure 6 – Simulated daily average supply air temperature outlet

Figure 6 shows that the ground coupled desiccant cooling process could supply air temperature between 21°C and 23°C with daily average ambient air temperatures between 19°C to 29°C during January. This also indicates that the outlet temperature was limited by the inlet ground water temperature of 20°C for this simulation study. An additional cooling method would be needed to further reduce the indoor temperature. However, the supply air from the desiccant cooling would be effective to remove the indoor moisture. In addition, Figure 6 shows that in some days in January, the daily average temperature is below the supply air temperature. Therefore, appropriate control strategies must be implemented, such as an economy mode, or bypassing the ambient air directly to the indoor area, when the ambient air condition is within the human comfort.

Figure 7 and 8 show that the daily average thermal COP and the solar fraction were greatly affected by the climate conditions. In addition, the PV/T output was insufficient to drive the desiccant regeneration process as the outlet temperature was below the minimum regeneration temperature. Therefore, an auxiliary heating element would be needed in the ground coupled PV/T cooling process. Further research in the operating optimisations and control strategies would be needed for the ground coupled PV/T desiccant cooling process.

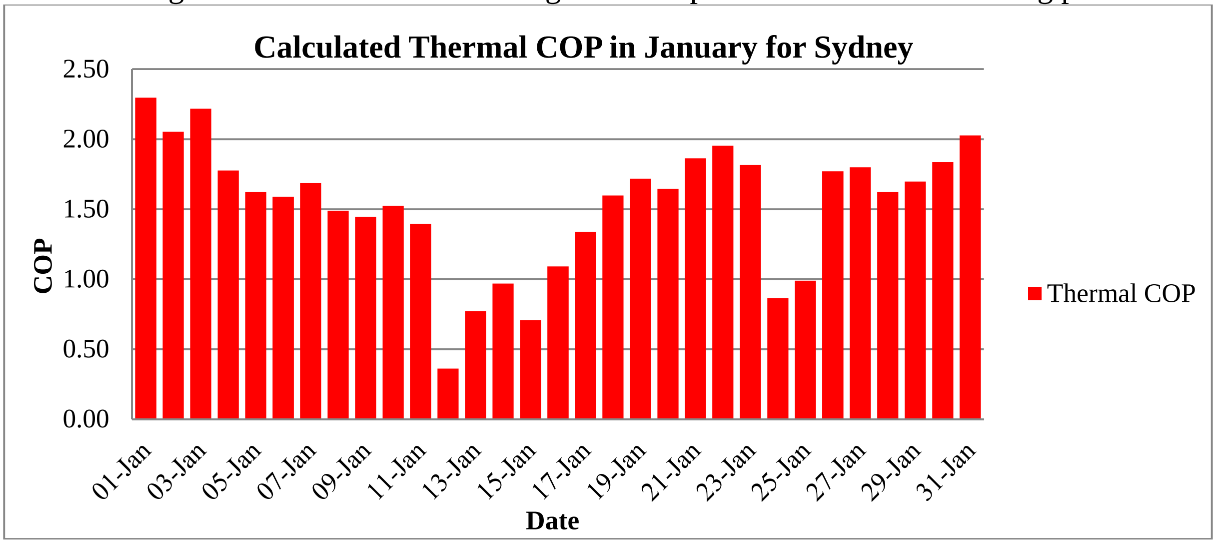

Figure 7 – Simulated daily average thermal COP

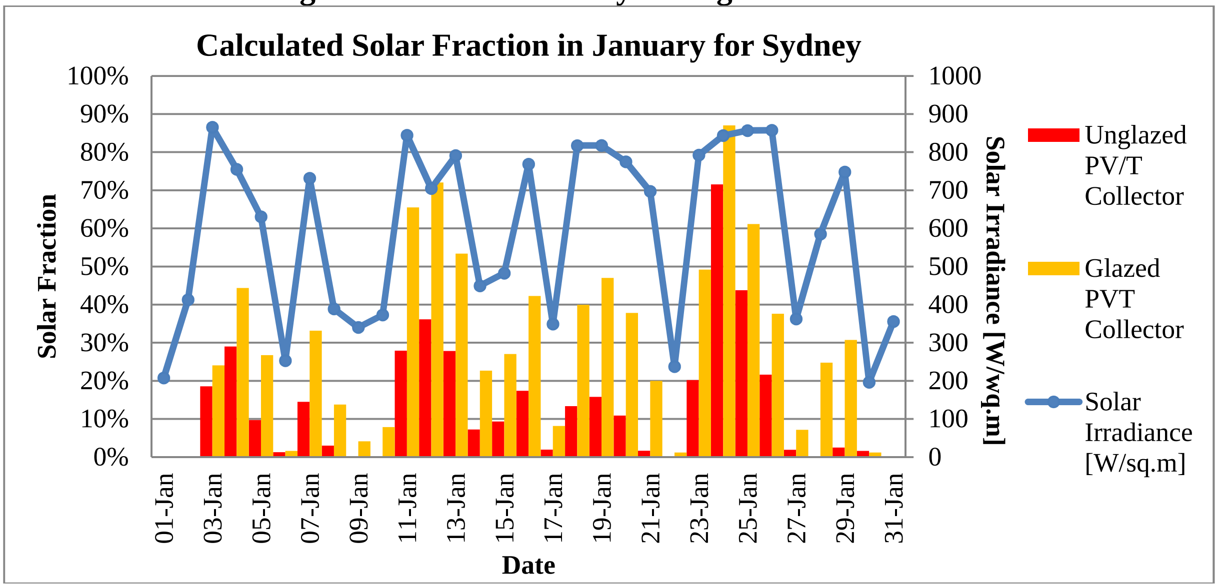

Figure 8 – Simulated daily average solar fraction

5. Conclusion

From the initial stage investigation, simulation results showed the ground couple PV/T desiccant air cooling process is could supply air with temperature below 25°C with specific humidity set at 0.008 kg/kg under the Sydney climate in the month of January. The thermal COP and the solar fraction varied greatly with the dynamic climate conditions.

In comparison to the conventional desiccant cooling process, the required regeneration temperature for the ground coupled desiccant cooling process with pre-cooling was less affected by the inlet air conditions. The pre-cooling limited the dehumidification load by condensing the moisture out from the air when the inlet air specific humidity is high.

Operation modelling of the PV/T collector showed low mass flow rate was preferred for higher outlet fluid temperature to suit the desiccant regeneration.

More detailed analysis of the ground coupled PV/T desiccant cooling process would be needed with improvements on the modelling and simulation to more accurately estimate the performance of this process. Furthermore, control and optimisation strategies would need to be focused to operate the cooling process to suit different climate conditions.

References

Bambrook, S.M. and Sproul, A.B, 2012, ‘Maximising the energy output of a PVT air system’, *Solar Energy*, 86, p1857–1871.

Bilbao, J.B, 2012, ‘Modelling and design considerations for residential applications in different climates and demand conditions’, Ph.D. Thesis, UNSW Australia.

Cengel, Y.A, 2002, ‘Heat transfer – a practical approach’, 2nd Ed., McGraw-Hill. Boston.

Duffie, J.A. and Beckman, W.A, 2006. ‘Solar engineering of thermal processes’, 3rd Ed., Wiley, New York, USA.

Eicker, U. and Vorschulze, C, 2009, ‘Potential of geothermal heat exchangers for office building climatisation’. *Renewable Energy*, 34, p1126–1133.

Eicker, U., Schneider, D., Schumacher, J., Ge, T. and Dai, Y.J, 2010, ‘Operational experiences with solar air collector driven desiccant cooling systems’, *Applied Energy*, 87, p3735-3747.

Florides, G. and Kalogirou, S, 2007, ‘Ground heat exchangers - A review of systems, models and applications’, *Renewable Energy*, 32, 2461–2478.

Florschuetz, L.W, 1979, 'Extension of the Hottel-Whillier model to the analysis of combined photovoltaic/thermal flat plate collectors', *Solar Energy*, 22, p361-366.

Goldsworthy, M. and White. S.D, 2012, 'Limiting performance mechanisms in desiccant wheel dehumidification', *Applied Thermal Engineering*, 44, p21-28.

He, W., Zhang, Y., and Ji, J, 2011, ‘Comparative experiment study on photovoltaic and thermal solar system under natural circulation of water’. *Applied Thermal Engineering*, 31, p3369-3376

Jurinak J.J, 1982, ‘Open cycle solid desiccant cooling – component models and system simulations’, Ph.D. Thesis, University of Wisconsin-Madison.

La, D., Dai, Y.J., Li, Y., Wang, R.Z. and Ge, T.S, 2010, ‘Technical development of rotary desiccant dehumidification and air conditioning: a review’, *Renewable and Sustainable Energy Reviews*,14, p130–147.

Li, X., Chen, Z. and Zhao, J, 2006, ‘Simulation and experiment on the thermal performance of U-vertical ground coupled heat exchanger’, *Applied Thermal Engineering*, 26, p1564–1571

Mei, L., Infield, D., Eicker, U., Loveday, D. amd Fux, V, 2006, ‘Cooling potential of ventilated pv façade and solar air heaters combined with a desiccant cooling machine’. *Renewable Energy*, 31, p1265-1278.

Panaras, G., Mathioulakis, E., Belessiotis, V. and Kyriakis, N, 2010, 'Experimental validation of a simplified approach for a desiccant wheel model', *Energy and Buildings*, 42, p1719–1725.

Panaras, G., Mathioulakis, E., Belessiotis, V. and Kyriakis, N, 2010, ‘Theoretical and experimental investigation of the performance of a desiccant air-conditioning system’, *Renewable Energy*, 35, p1368–1375.

Sproul, A.B., Jose.I.B. and Bambrook, S.M, 2012, ‘A novel thermal circuit analysis of solar thermal collectors’, AuSES 2012 Annual Conference.

Zondag, H.A., De Vries, D.W., Van Helden, W.G.J., Van Zolingen R.J.C. and Van Steenhoven, A.A, 2002, 'The thermal and electrical yield of a PV/thermal collector', *Solar Energy*, 72, p113–128