Techno-economic Analysis of a Solar Desiccant-Evaporative Cooling System with Different Collector Types for Australian Office Buildings

Yunlong Ma[1], Lisa Guan[1] and Richard Brown[1]

[1] *Science and Engineering Faculty, Queensland University of Technology (QUT), 2 George Street, Brisbane, 4000, Australia*

E-mail: yunlong.ma@student.qut.edu.au

Abstract

By using building simulation program EnergyPlus, this paper evaluates the feasibility of adopting the solar desiccant-evaporative cooling (SDEC) system for a typical commercial building in all eight Australian capital cities, including Adelaide, Brisbane, Canberra, Darwin, Hobart, Melbourne, Perth and Sydney. Three different types of solar collector subsystem configurations for the proposed SDEC system are investigated, which are solar thermal (ST) collector, solar photovoltaic (PV) panel, and photovoltaic-thermal (PVT) collector. The technical and economic performances are analysed, including solar fraction (SF), system coefficient of performance (COP), annual site energy consumption, and life cycle cost (LCC). The simulation results show that for the proposed three SDEC systems, Darwin could achieve the highest yearly averaged SF and COP. The yearly SF in Darwin is 1.06 for the SDEC-PV system, 0.9 for SDEC-ST system, and 1.48 for SDEC-PVT system. The yearly system COP is 4.53 for SDEC-ST, 5.7 for SDEC-PV and 4.7 for SDEC-PVT respectively. It is also found that the SDEC-PVT system has the best system performance for Adelaide, Brisbane, Darwin, Melbourne, Perth and Sydney, from both technical and economic points of view. For other two capital cities, the SDEC-PV system is the best option due to the lowest LCC.

1. Introduction & Literature Review

Australia is currently facing the challenge of dramatic peak electricity demand due to large residential and commercial heating, ventilating, and air conditioning (HVAC) penetrations. Research indicates that the Australian building industry consumes 40% of the nation's total electricity energy and is responsible for 27% of national greenhouse gas (GHG) emissions (Baniyounes et al., 2013). Commercial buildings, in particular, account for approximately 61% total building electricity energy depletion and 8~10% total building GHG emissions (Daly et al., 2014). In addition, the HVAC system installed in buildings is the largest energy consumption contributor, which represents 68%, followed by 19% for lighting and 13% for others. Overall, commercial HVAC systems contribute about 30% of the total energy demands in commercial buildings (Baniyounes et al., 2013). Therefore, developing innovative HVAC technology towards sustainability is vitally important for Australia to decrease the nation's electricity consumption and GHG emissions.

Since Australia has a rich solar energy resource with the highest average solar radiation per square meter in the world (Geoscience Australia and ABARE, 2010), solar air conditioning technology is highly desirable as its availability coincides with the cooling demand. Thus, the peak electricity demand due to wide use of air conditioning in summer can be reduced as it matches with the peak solar irradiance (Angrisani et al., 2014). Recently, solar desiccant-based air conditioning technology has been widely developed in Australia as it is considered to be an attractive alternative for conventional vapour compression cooling system because of energy efficient and environmentally friendly characteristics. Baniyounes et al. (2012) conducted the study of a solar desiccant cooling for an institutional building in central Australian subtropical climate using TRNSYS. They demonstrated that the system would achieve an annual COP of 0.7, 22% solar fraction, 4.4 tonnes of GHG emissions reduction, and 22 years payback by installing 10m^2 solar collectors and 0.4 m^3 storage tank. Goldsworthy and White (2011) conducted the optimisation of a solar desiccant cooling system with indirect evaporative cooler in Australia. They found that for 70^{o}C regeneration temperature, a supply/regeneration flow ratio of 0.67 and an indirect evaporative cooler secondary/primary flow ratio of 0.3 gives the best performance with the COP above 20. White et al. (2009) assessed the performance of a solar desiccant cooling system without thermal backup for an office space in Melbourne, Sydney, and Darwin. They concluded that increasing the indirect evaporative effectiveness, reducing the desiccant wheel regeneration temperature, and increasing collector areas will apparently result in improved desiccant cooling cycle system performance in Melbourne and Sydney, but not evident in Darwin.

However, from the literature review and the best of authors' knowledge, there are no existing investigations that evaluate three different types of solar collectors (solar thermal, solar PV and PVT) for the same desiccant cooling system in terms of energy and economic performances. Furthermore, there are also no baseline case studies for the feasibility of solar desiccant cooling for the representative office building under all Australian climates. Therefore, a feasibility study of the solar desiccant-evaporative cooling system with three solar collector types for a typical office building in all eight Australian capital cities will be conducted in this research, by means of a techno-economic analysis.

2. Methodology

In order to assess the proposed SDEC-ST, SDEC-PV, and SDEC-PVT system performance, a year round simulation is conducted for each city using computer simulation software EnergyPlus. The technical and economic indicators will be analysed, including solar fraction, system coefficient of performance, annual site energy usage, and life cycle cost. To obtain hourly values of the cooling load and run EnergyPlus for a year round simulation, an Australian Representative Meteorological Year (RMY) climate data file for each city is required. The Australian RMY files are typical weather data developed for the Australian Greenhouse Office for use in complying with Building Code of Australia. It is supplemented by solar radiation estimated on an hourly basis from earth-sun geometry and hourly weather elements (U.S. Department of Energy, 2015).

2.1. Building model description

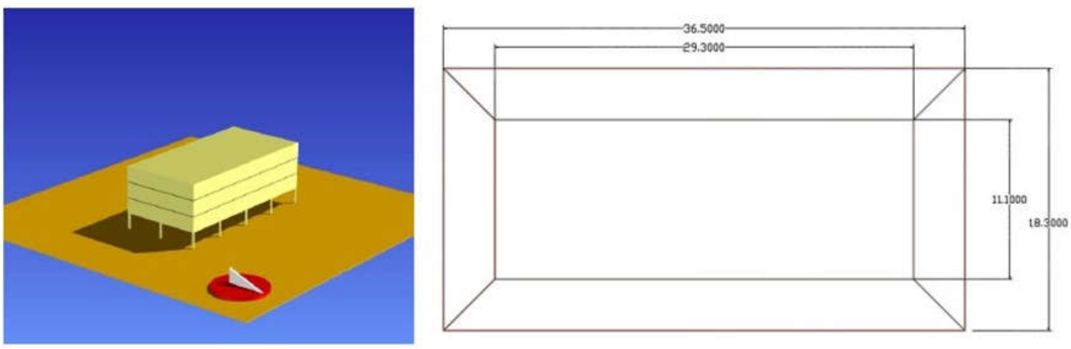

Figure 1. Building model geometry and zone division

As recommended by the Australian Building Codes Board (ABCB) to be a representative medium sized commercial building in the central business district of Australian capital cities (ACADS-BSG, 2002), the building to be modelled is a 5-zone, rectangular, three-storey office building with a basement car park, which is demonstrated in Figure 1 above. The building physical properties and internal load assumptions are summarized in the following tables. The selection of different schedules are based on Donnelly (2004).

Table 1. Building physical properties

Building feature	Value
Number of storey	3
Footprint dimensions	36.5 m×18.3 m
Gross floor area	2003.85 m^2
Aspect ratio	2:1
Floor-to-ceiling height	2.7 m
Plenum wall height	0.9 m
Car park height	3 m
Building total height	13.8 m
Orientation	long axis East-West
Number of zones/floor	5
Infiltration	1 ACH, no infiltration during HVAC operation
Outside air rate	10 L/s per person
Lighting schedule	91.5 h/week
Equipment schedule	97.45 h/week
Occupancy schedule	53.75 h/week
HVAC schedule	60 h/week
HVAC setpoints	24°C, 50% RH for cooling & 20°C for heating, 1°C tolerance

Table 2. Building envelope and internal load assumptions

Parameters	Value
Roof	Metal deck, air gap, foil, roof space, R2.0 batts, 13 mm acoustic tiles (U=0.277 W/m^2K)
Floor	175mm Concrete Slab with carpet (U=1.32 W/m^2K)
Exterior wall	200 mm heavy weight concrete, R1.5 batts, 10 mm plasterboard (U=0.554 W/m^2K)
Window	Single 6 mm Clear Glass, WWR=0.4 (U=5.89 W/m^2K)
Lighting power density	15 W/m^2
Equipment load density	15 W/m^2
Occupant density	10 m^2/person

2.2. *System configuration and simulation input parameters*

The basic SDEC system combines the desiccant process with evaporative cooling together, which is mainly comprised of: (1) the solar subsystem which consists of solar collectors, storage tank, and backup heater; (2) the desiccant subsystem, which includes a desiccant wheel (DW), desiccant material (generally silica gel), a sensible air-to-air heat exchanger (HX), and a regeneration heating coil; and (3) the evaporative cooler (EC). The schematic diagram of a typical SDEC system is demonstrated in Figure 2 below.

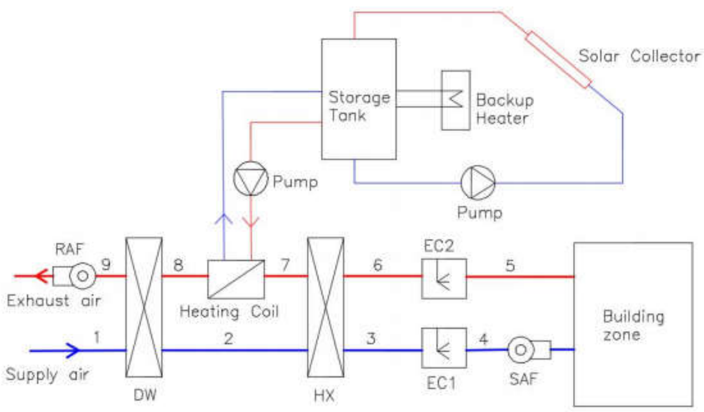

Figure 2. Schematic diagram of the SDEC system

To reduce energy consumption, control strategies are applied to the system so that the desiccant system is operating only when the outdoor air dry bulb temperature is above 18°C and humidity ratio is higher than 0.008 kg/kg. It is controlled by a sensor that provides an on/off signal to solar subsystem pumps. When the outside air conditions are under the control setpoints, the solar collector pump and regenerative hot water pump will be off to disable the solar subsystem so that the desiccant wheel and regeneration air heater will be turned off. This would significantly avoid unnecessary backup energy consumption.

For the SDEC-PV system, the system configuration is different with the other two systems. First, it does not have a solar hot water loop. Thus, it does not require the storage tank, backup heater, and hot water pumps. In addition, it uses an electric heating coil to provide regeneration instead of the hot water heating coil. It should also be noted that the PV cells only produce direct current (DC) electricity while the building utilities and HVAC system equipment will consume alternating current (AC), therefore a DC-AC inverter is needed for the SDEC-PV and SDEC-PVT systems. Moreover, the two systems are connected with city electricity grid, which will either receive surplus electricity from or provide auxiliary electricity to the systems. The simulation parameters are summarised in the following table.

Table 3. Simulation parameters

Parameters	Value	Parameters	Value
General		*Desiccant wheel*	
Supply temperature setpoint (°C)	18	DW nominal air flow rate (m^3/s)	19
Solar collector type	Flat Plate	DW nominal electric power (W)	100
Fluid type	Water	DW nominal air face velocity (m/s)	4
Collector area (m^2)	760	Minimum regeneration temperature (°C)	50
Collector tilt	25°	*Heat exchanger*	
Collector loop flow rate (kg/s)	3	HX type	Flat Plate
Hot water loop flow rate (kg/s)	2.4	HX nominal air flow rate (m^3/s)	19
Regenerative hot water design temperature (°C)	75	Ratio of supply to secondary hA values	1
Backup heater fuel type	Electricity	Nominal electric power (W)	0
Backup heater efficiency	1	Nominal supply air inlet temperature (°C)	54
Backup Heater overall loss coefficient (W/K)	0	Nominal supply air outlet temperature (°C)	24
Backup heater capacity (kW)	100	Nominal secondary air inlet temperature (°C)	20
Storage tank volume (m^3)	7.2	*Direct evaporative cooler*	
ST collector		Coil maximum efficiency	0.9
Collector optical efficiency c_0	0.753	Recirculating water pump power (W)	50
Collector heat loss coefficient c_1 ($W/m^2 \cdot K$)	-5.2917	*Regeneration heating coil*	
Collector heat loss coefficient c_2 ($W/m^2 \cdot K^2$)	0.00638	Regeneration heating coil capacity (kW)	330
PV panels		Rated inlet water temperature (°C)	75
Cell efficiency	0.2	Rated inlet air temperature (°C)	35
Inverter efficiency	0.985	Rated outlet water temperature (°C)	45
PVT panels		Rated outlet air temperature (°C)	50
Thermal conversion efficiency	0.3	Rated ratio for air and water convection	0.5
Front surface emittance	0.84	*Supply & regeneration air fan*	
Cell efficiency	0.2	Fan delta pressure (Pa)	500
Inverter efficiency	0.985	Fan total efficiency	0.7

2.3. *Main system components modelling*

2.3.1. *Desiccant wheel*

Desiccant wheel is the key component in the SDEC system which deals with both sensible and latent heat transfer between the process and regeneration air streams. In EnergyPlus this model is a balanced flow desiccant heat exchanger which assumes the same air volume flow rate and face velocity through the regeneration and process sides. Its performance is specified through an empirical performance data that predicts the regeneration air stream outlet temperature and humidity ratio based on the entering regeneration and process air stream conditions and face velocity. The regeneration outlet air temperature and humidity ratio governing equations are expressed as following (U.S. Department of Energy, 2013):

$$RTO = B_1 + B_2 * RWI + B_3 * RTI + B_4 * \left(\frac{RWI}{RTI}\right) + B_5 * PWI + B_6 * PTI + B_7 * \left(\frac{PWI}{PTI}\right) + B_8 * RFV \quad (1)$$

$$RWO = C_1 + C_2 * RWI + C_3 * RTI + C_4 * \left(\frac{RWI}{RTI}\right) + C_5 * PWI + C_6 * PTI + C_7 * \left(\frac{PWI}{PTI}\right) + C_8 * RFV \quad (2)$$

where RTO is regeneration outlet air dry bulb temperature in ^{o}C; RWI is regeneration inlet air humidity ratio in kg/kg; RTI is regeneration inlet air dry bulb temperature in ^{o}C; PWI is process inlet air humidity ratio in kg/kg; PTI is process inlet air dry bulb temperature in ^{o}C; RFV is regeneration (and process) face velocity in m/s; B_n is temperature equation coefficient; RWO is regeneration outlet air humidity ratio in kg/kg; and C_n is humidity ratio equation coefficient. The coefficients of B_n and C_n are shown in Table 4 below from manufacture data (Desiccant Rotors International Pvt. Ltd). A humidity ratio setpoint of 0.005 kg/kg is applied on the desiccant wheel process air outlet node for the setpoint of desiccant wheel (Dezfouli et al., 2014). The RWI is set to 0.011 kg/kg based on return air conditions and the RTI is set to 50^{o}C.

Table 4. Coefficients for desiccant wheel temperature and humidity ratio equations

B1	B2	B3	B4	B5	B6	B7	B8
-27.18302	-184.967	1.00051	11603.3	-50.755	-.0168467	58.2213	.598863
C1	**C2**	**C3**	**C4**	**C5**	**C6**	**C7**	**C8**
.01213878	1.09689	-.0000263341	-6.33885	.00938196	.0000521186	.0670354	-.000160823

2.3.2. *Heat exchanger*

The air-to-air flat plate heat exchanger is modelled as follows (Panaras et al., 2011):

$$\varepsilon_{HX} = \frac{t_2 - t_3}{t_2 - t_6} \quad (3)$$

$$t_2 - t_3 = t_7 - t_6 \quad (4)$$

where ε_{HX} is heat exchanger effectiveness; t_2 is heat exchanger process air inlet dry bulb temperature in ^{o}C; t_3 is heat exchanger process air outlet dry bulb temperature in ^{o}C; t_6 is heat exchanger regeneration air inlet dry bulb temperature in ^{o}C; and t_7 is heat exchanger regeneration air outlet dry bulb temperature in ^{o}C. The sensible heat exchanger is considered to present air streams of equal flow rate and no heat losses to environment.

2.3.3. *Evaporative cooler*

The direct evaporative cooler is modelled using the following relation (U.S. Department of Energy, 2013). It assumes a constant effectiveness model and the wet bulb temperature remains constant between the inlet and outlet of the cooler.

$$T_{db,out} = T_{db,in} - \varepsilon\left(T_{db,in} - T_{wb,in}\right) \quad (5)$$

where $T_{db,out}$ is the dry bulb temperature of the air leaving the cooler in ^{o}C; $T_{db,in}$ is the dry bulb temperature of the air entering the cooler in ^{o}C; $T_{wb,in}$ is the wet bulb temperature of the air entering the cooler in ^{o}C; and ε is the cooler effectiveness.

2.4. Building model and system validation

Experimental data is not available for the building model and system validation as this is an archetypal building and no existing SDEC systems are in operation with this building model. Therefore, the monthly indoor comfort condition is used for the validation. Figure 3 shows the simulation results of the monthly indoor temperature for the SDEC-ST system in all 8 studied locations. It is clear that the building indoor temperature can maintain the designed conditions of 24^{o}C in summer and 20^{o}C in winter, indicating the building model and HVAC system are constructed correctly. It should point out that for some hourly time steps in Darwin, the average building indoor temperature is higher than 24^{o}C. This is because the outdoor air humidity ratio is so high that the desiccant wheel could not dehumidify the outdoor air to its setpoint condition in these hours. But its monthly averaged temperature can meet the indoor design condition.

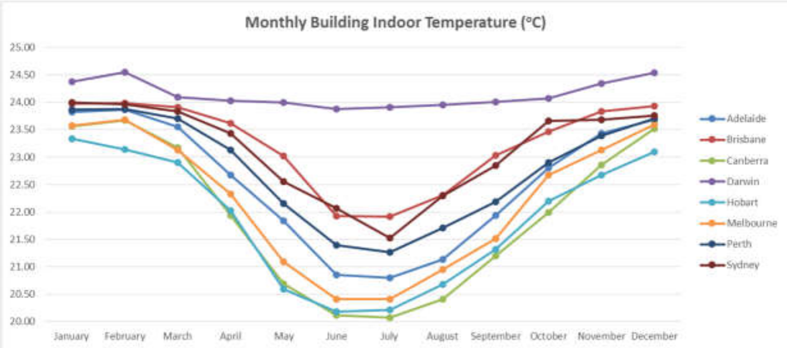

Figure 3. Monthly building indoor temperature for SDEC-ST system

3. Results and Discussions

3.1. Solar fraction

Solar fraction is expressed as the ratio of solar energy contribution to the total energy input for driving the solar cooling system, which can be defined as the following equation:

$$SF = \frac{Q_{Solar}}{Q_{Reg}} = \frac{Q_{Solar}}{Q_{Solar} + Q_{Aux}} \tag{6}$$

where Q_{Solar} is the solar thermal gain for solar thermal panels, or electricity output for PV panels, or both solar thermal and electric gains for PVT collectors in kW; Q_{reg} is the heat input for driving the desiccant wheel in kW, including solar thermal gain or solar electric input for regeneration, and the backup energy if necessary; Q_{Aux} is the backup energy input in kW.

Figure 4 demonstrates the annual averaged SF of the three proposed SDEC systems for all 8 Australian capital cities. It indicates that generally the SDEC-PV system has the lowest annual SF due to low electricity conversion efficiency. While the SF of SDEC-PVT is the highest because of both electricity and thermal production. In addition, for the comparison among different locations, Darwin has the highest SF value of about 0.9 for SDEC-ST, 1.06 for SDEC-PV, and 1.48 for SDEC-PVT. The reason for the higher SF of SDEC-PV in Darwin is that the solar radiation is so abundant that the PV panels could produce more electricity than the cooling system consumes. Brisbane has the second largest annual averaged SF values of 0.79 for SDEC-ST, 0.53 for SDEC-PV, and 1.1 for SDEC-PVT. While Hobart has the lowest annual SF values of only around 0.51 for SDEC-ST, 0.2 for SDEC-PV, and 0.69 for SDEC-PVT.

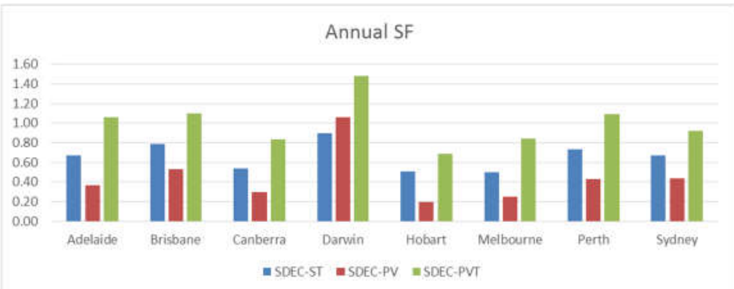

Figure 4. Annual solar fraction

3.2. Coefficient of performance

COP is defined as the ratio of the system cooling capacity to the total HVAC system energy input. The higher the COP is, the more efficient the system is. For the SDEC system, the system COP can be defined as:

$$COP_{des} = \frac{Q_C}{Q_{reg}} = \frac{m_s \times (h_o - h_s)}{Q_{reg}} \quad (7)$$

where Q_C is the cooling effect in kW; m_s is supply air mass flow rate in kg/s; h_o is the enthalpy of outside air in kJ/kg; and h_s is the enthalpy of supply air in kJ/kg. The COP is counted only when the desiccant wheel is in operation for each time step.

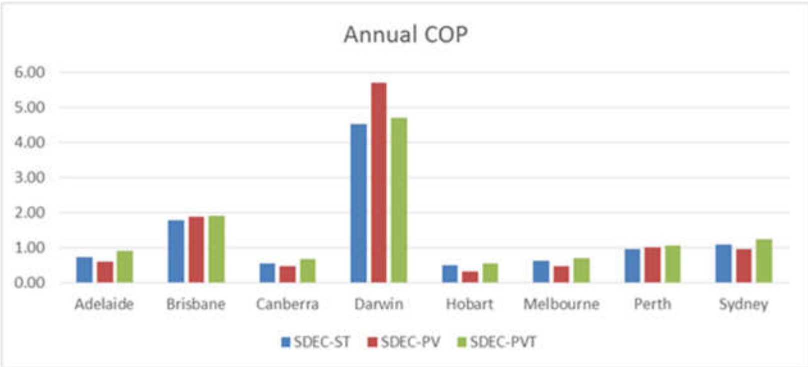

Figure 5. Annual system COP

Figure 5 shows the annual system COP results. It is obvious that Darwin has dramatically higher annual COP than others, reaching 5.7 for SDEC-PV, 4.7 for SDEC-PVT and 4.53 for SDEC-ST. While Hobart has the lowest annual COP for all three SDEC systems. In addition, the SDEC-PVT system has the highest yearly COP value among the three solar collector subsystem configurations on the whole, followed by SDEC-ST and SDEC-PV.

3.3. Annual site energy consumption

Annual site energy is the total electric energy consumed by the whole building including the HVAC system, lighting and equipment. The annual site energy consumption is defined as:

$$E_{des} = E_{air} + E_{hyd} + E_{Aux} + E_l + E_{equip} \quad (8)$$

where E_{air} is the conventional system air side electricity energy consumption in GJ; E_{hyd} is the electric energy consumption of hydronic equipment in GJ; E_l is the interior lighting consumption in GJ; E_{equip} is the equipment energy consumption in GJ; and E_{Aux} is the electric energy consumption by the auxiliary heater in GJ.

It is assumed that the produced PV electricity is not only used by the HVAC components but also can be used by lighting and equipment for the building. Thus, the net site annual electricity consumption could be described in Figure 6 below.

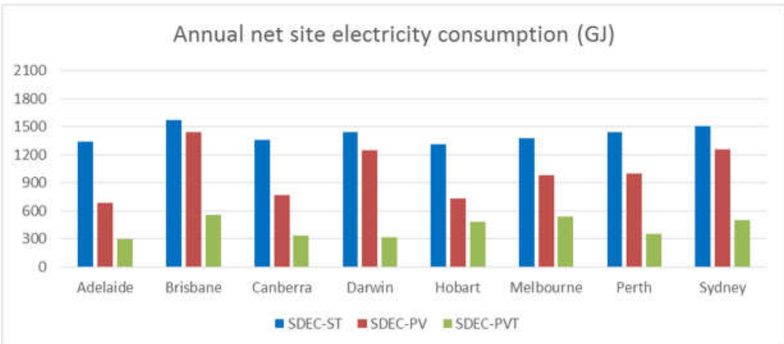

Figure 6. Annual net site electricity consumption

It shows that the SDEC-PVT system is the most energy efficient configuration followed by SDEC-PV and SDEC-ST. Additionally, Brisbane consumes the most energy for all three SDEC systems, reaching 1570 GJ for SDEC-ST, 1444 GJ for SDEC-PV, and 556 GJ for SDEC-PVT.

Adelaide consumes the least annual electricity energy for SDEC-PV and SDEC-PVT, only 682 GJ and 298 GJ respectively. While for SDEC-ST system, Hobart is the lowest energy consumer with 1314 GJ annually.

3.4. Life cycle cost

Solar air conditioning systems are characterised by high initial cost (IC) and low operating cost (OC). The LCC is the summation of the IC and OC through the lifespan of the air conditioning system. The present value method is used to calculate LCC by introducing a present worth factor (PWF), which is used to compare the future cost of a renewable energy system to today's cost taken into account an obligation recurs each year at i inflation rate and d discount rate over N years of lifespan as expressed in Eq. (10) (Duffie & Beckman, 2013).

$$PWF(N,i,d) = \sum_{j=1}^{N} \frac{(1+i)^{j-1}}{(1+d)^j} = \begin{cases} \frac{1}{d-i}\left[1-\left(\frac{1+i}{1+d}\right)^N\right], if\ i \neq d \\ \frac{N}{i+1}, if\ i = d \end{cases} \quad (10)$$

where PWF implies the present worth factor; i is the inflation rate; d is the discount rate; and N is the life cycle of the system.

Then, the LCC is calculated using Eq. (11) (Abdel-Salam et al., 2014).

$$LCC = IC + PWF \times OC \quad (11)$$

The system initial investment costs strongly depend on the equipment capacity. The fuel cost and rated PV output are included in Table 6. The equipment costs are summarised in Table 7.

Table 6. Fuel cost and rated PV output for each city

	Adelaide	Brisbane	Canberra	Darwin	Hobart	Melbourne	Perth	Sydney
PV output (kW)	181	170	185	160	185	172	177	163
Fuel cost (c/kWh)	13.38[a]	10.48[b]	5.95[c]	16.3[d]	7.5[e]	19.8[f]	14.13[g]	13.05[h]

a. http://businesstech.co.za/news/general/41218/south-africas-electricity-price-shock. Nationwide value.
b. Barclay D., Chester C., Cooper J., and Liddy A. 2015. Regulated retail electricity prices for 2015-16. Final determination. Queensland Competition Authority. Brisbane.
c. Retail electricity price recalibration 2015-16. Final decision. Independent Competition and Regulatory Commission.
d. Northern Territory of Australia. Electricity Reform Act. Electricity Pricing Order, 2005.
e. AURORA ENERGY PTY LTD, Interim Price-Regulated Retail Service Price Determination, 29 July 2013.
f. Essential Services Commission 2013, Retailer Margins in Victoria's Electricity Market-Discussion Paper, May.
g. www.finance.wa.gov.au
h. Energy Australia. Business Customer Price List. Regulated Retail Tariffs.

Table 7. Costs of different components and economic parameters

Component	References	Price/unit
Pumps	Australian Construction Handbook, 2011	\$2040*2
Desiccant dehumidifier	Desiccant Rotors International Pvt. Ltd	\$46000
Heat exchanger	Guangzhou Jiema Heat Exchange Equipment Co., Ltd	\$10000
Evaporative cooler	Australian Construction Handbook, 2011	\$3300*2
Solar thermal collector	Alizadeh, 2008	\$300/m^2
PV panels	IEA, 2014	\$1.7/W
PVT panels	Mittelman et al., 2007	\$2.7/W on average
Fans	Australian Construction Handbook, 2011	\$8600*2
Air distribution units	Australian Construction Handbook, 2011	\$3300*15
O&M cost	Henning, 2007; Salasovich & Mosey, 2011	0.17% of total investment cost for PV system; 1% for solar thermal collector system
Lifespan of system	Baniyounes et al., 2012	25 years
Inflation rate	Donnelly, 2004	2.5%
Discount rate	Baniyounes et al., 2012	8%

Therefore, the LCC of the proposed SDEC systems for all eight Australian locations can be obtained in Table 8.

Table 8. Life cycle cost of the proposed SDEC systems

Million $	Adelaide	Brisbane	Canberra	Darwin	Hobart	Melbourne	Perth	Sydney
SDEC-ST	1.056	1.006	0.698	1.268	0.763	1.403	1.151	1.118
SDEC-PV	0.779	0.977	0.614	1.151	0.648	1.137	0.953	1.010
SDEC-PVT	0.774	0.808	0.711	0.759	0.773	0.996	0.802	0.817

The economic performance results show that generally the SDEC-PVT system has the lowest LCC and the SDEC-ST has the highest LCC among the three proposed SDEC systems except in Canberra and Hobart. For Canberra and Hobart, the SDEC-PV system has the best economic performance while the SDEC-PVT system is the worst. This is due to high PVT module costs and low energy savings achievements. It is also obvious that comparing the SDEC-PVT with SDEC-ST, Darwin has the largest potential for the application of the SDEC-PVT system because of the biggest life cycle cost savings.

4. Conclusions

In this paper, the feasibility of the solar desiccant-evaporative cooling for a typical office building has been assessed under a variety of Australian climates from energy and economic respects. Three different types of solar subsystem configurations have been investigated, namely SDEC-ST, SDEC-PV, and SDEC-PVT. The simulation results indicate that the SDEC-PVT system has the best system performance overall because of the highest annual SF and COP, as well as the lowest yearly net energy consumption and life cycle cost. However, from economic aspect, the SDEC-PVT system is more advantageous for Adelaide, Brisbane, Darwin, Melbourne, Perth and Sydney since the least LCC. While for other two cities, the SDEC-PV system is more economically suitable.

It is noted that the technical and economic performance results are strongly dependent on the current assumptions about the system components parameters and economic factors such as equipment costs and fuel costs. However, under the current conditions, the simulation results imply that the SDEC system is technically and economically feasible for Australian commercial buildings. Further studies will be focused on the sensitivity analysis for the system optimisation in order to further improve the simulation results accuracy and reliability.

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