

Investigating Performance of Solid Desiccant Dehumidification in an Evaporative Cooling System

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Abstract

Desiccant evaporative cooling (DEC) systems are still at early stage of utilisation and commercialise development. The primary focus of this study is to analyse the simulated performance of DEC in hot and humid climates. Four configurations of one-stage solid desiccant dehumidifiers with two-stage evaporative cooling systems have been simulated using TRNSYS simulation software for a typical test room. The psychrometric process and the coefficient of performance (COP) for each configuration is presented and configurations for the most efficient system have been identified.

Keywords One-stage desiccant dehumidifier, evaporative cooling, hot and humid climate

1.0 Introduction

Refrigeration, including air conditioning, is necessary for life and will continue to expand worldwide. Its impact on environment is huge, even if refrigeration technologies can also be part of solutions for mitigating global warming. However, reduction in CO₂ emissions and fluorinated gas emissions are challenges to be addressed on an ongoing basis. Desiccant evaporative cooling (DEC) is an alternative approach designed to overcome these issues. Both sensible and latent heat loads are catered for separately using DEC systems.

The desiccant evaporative cooling technology is a combination of the desiccant dehumidifier and evaporative cooler. Evaporative cooling systems on their own are not recommended for humid weather as the performance is not acceptable in high humidity. In practical systems, a desiccant dehumidifier is combined with the evaporative cooling system is used to reduce the moisture content in the air. Desiccant materials in either solid, liquid or combined form have been widely used for removing moisture from ambient air. Common solid desiccant materials include polymers, silica, zeolites, alumina, hydratable salts and mixtures while typical liquid desiccants are include lithium chloride (LiCl), calcium chloride (CaCl), lithium bromide (LiBr), tri-ethylene glycol (TEG) and calcium chloride–lithium chloride mixture (1). The difference in water content between the desiccant material and air makes the moisture transfer from the ambient air to the desiccant material. In the regeneration process, the desiccant material is heated up to remove moisture content from the

desiccant material. Recent research has shown significant innovation to DEC system for hot and humid weather. Desiccant systems are based on moisture dehumidification which makes them suitable for relatively moderate to high humidity regions (2). Bespoke desiccant system designs would be feasible for particular regions or conditions, but more research is required to enable such specific system designs for specific conditions.

The evaporative cooling method was introduced to utilise the natural cooling effects of evaporation to cool down buildings. There are several configurations of evaporative coolers (either direct or indirect) that are operated in ventilation or recirculation cycles (3,4). The only energy required in DEC systems is that to drive the fans and water pump and to regenerate the desiccant dehumidifier during the regeneration process. This regeneration energy can be provided by any renewable, low-grade thermal energy source such as solar or waste heat. The sensible and latent loads can be controlled separately in this system using a humidistat and thermostat for the control of wet and dry bulb temperatures respectively. This system is a suitable alternative to mechanical vapour compression systems and can be efficiently used for air-conditioning applications with fewer power requirements (5).

Sultan et al. (2) review of the solid desiccant cooling system showed that at this stage, the actual large size of the system is an obstacle in the way of its acceptance as a technological advantage over alternative vapour compression (VC) system. For example, the Energy Efficiency Ratio (EER) for split and multi-split air conditioners is 2.5 (6) even though this value is still high compared to the coefficient of performance (COP) of DEC system. Dezfouli et al.(7) have reported a COP of 1.06 for two-stage solar desiccant cooling system and Jani et al. (8) have achieved a COP of up to 2.0 for solid desiccant-vapour compression hybrid air-conditioning system, some of the researchers also reported that DEC can be more acceptable for central cooling. At the same time, there is a lack of knowledge and familiarity with desiccant technology amongst designers, developers, architects and end users. As a result, it will be difficult for DEC to expand into the market unless highly efficient desiccant systems are designed and there is a wide acceptance by the end user due to government incentives. For this to happen, there is need for more research into the design of compact and efficient DEC and its performance assessed and quantified. The design should then be validated by the appropriate regulatory body.

Daou et al. (9) found that one of the most significant advantages of desiccant cooling systems lies in the possibility of their regeneration by use of free energy derived from waste and solar power, without any emissions. Hence, the system is not only ideal for controlling temperature and humidity, but it is also has the advantage of saving energy, being cost-effective and having a low environmental impact. DEC systems that go through the regeneration process using renewable energy sources such as waste heat, natural gas, and solar energy are therefore the most popular systems (9–12). The DEC systems also have the advantage of improving the system energy performance (13) as well as reducing energy consumption (11,12,14–16).

Heidarinejad et al. (17) investigated various cooling systems in a variety of multi-climate countries. They used a basis of numerical simulation and long-term meteorological data to analyse the cooling process of various DEC systems. It was found that the DEC system is a workable system for most of the cities of Iran, Oman, Turkey, and Saudi Arabia. Feasibility studies have also been conducted by Mavroudaki et al. (11) on solar-driven desiccant cooling in various European cities representing different climatic zones on the continent and showed that primary

energy savings were achieved in all weather conditions. However, the savings were smaller in humid zones due to the high temperatures required for desiccant regeneration. Up to 80% of energy can be saved when the latent load constitutes 90% of the total cooling load; a percentage obtained through simulation results of hybrid liquid desiccant cooling and vapour compression air conditioning systems regenerated by solar energy (12).

Based on the studies and publications mentioned already, it was found that very limited research has been conducted on the significant parameters that affect solid desiccant evaporative cooling systems. There are limited studies that focus on one-stage and two-stage desiccant evaporative cooling systems without comprehensive results. This paper investigates the performance of different types of solid desiccant evaporative cooling system focusing on hot and humid climate conditions.

2.0 Basic desiccant evaporative cooling system configuration

The basic DEC system constitutes a one-stage dehumidification process with a direct evaporative cooling system. The system consists of a solid rotary desiccant wheel (SRDW), a heat recovery wheel (HRW), evaporative cooler(s) (ECs), a heater and a fan. A schematic layout of a one-stage desiccant direct evaporative cooling system in ventilation mode is shown in Figure 1. On the process side, the ambient air at state 1 is passed through the solid rotary desiccant wheel for dehumidification. Its moisture is partly adsorbed by the desiccant wheel, and the processed air temperature is increased as the heat of adsorption drifts its temperature. At state 2, a warm and dry air stream exits from the desiccant wheel. The air stream is then sensibly cooled in the air-to-air heat recovery wheel between state 2 and state 3. No latent change takes place at this stage. The cool and dry air is further cooled, and slight moisture is gained when passing through a direct evaporative cooler (EC1) from state 3 to state 4 before entering the room. In the regeneration side, after leaving the room, return air passes through another evaporative cooler (EC2) and heat recovery wheel (HRW) to recover heat from warm and dry air (state 6 to state 7). Between state 7 and state 8, it is further heated by an electric heater (EH) to a required regeneration temperature (state 8). From state 8 to state 9, regeneration air extracts moisture from the desiccant wheel (SRDW) and expels the moisture to the ambient at state 9. The psychrometric chart of this system is shown in Figure 2.

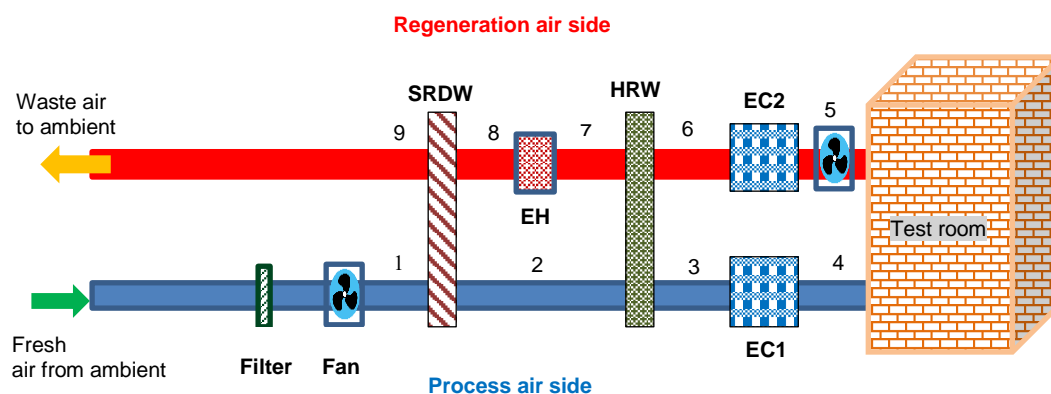


Figure 1- Schematic layout of one-stage desiccant direct evaporative cooling system.

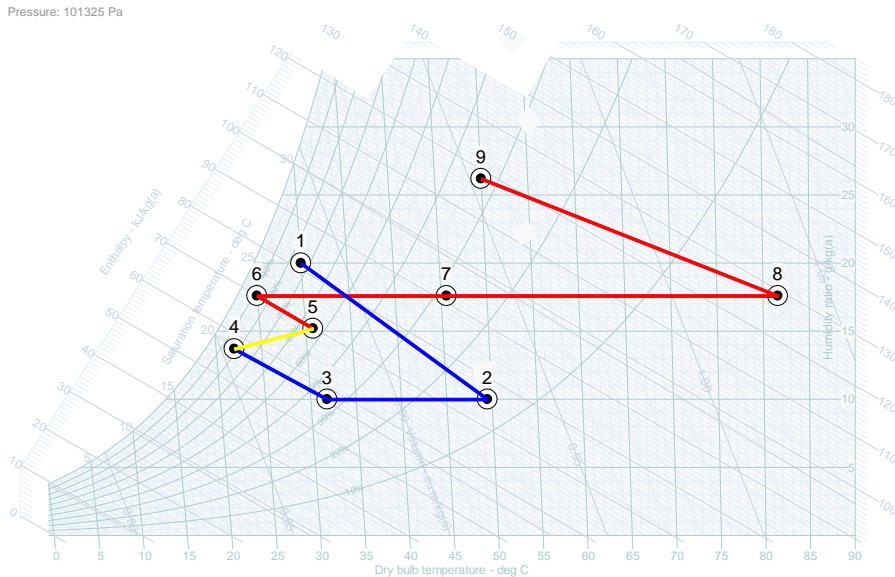


Figure 2 – Psychrometric chart of the above DEC system.

3.0 Simulation model description

As described previously, a desiccant air-conditioning system consists of several components. A brief description of each component is presented in this section with the aim of explaining the selection of specific models for each component of operation in a TRNSYS simulation studio. In the set-up of a DEC system, the input of the one component is linked to the output of the other component.

3.1 Solid rotary desiccant dehumidifier

Type1716a component models a rotary desiccant dehumidifier containing nominal silica gel whose performance is based on equations for F1-F2 potentials developed by Jurinak (18) that are rearranged to solve for the temperature in Equations 1 and 2:

$$T = \left(\frac{2865}{4.344\omega^{0.8624} - F1} \right)^{1/1.49} \tag{Equation 1}$$

$$T = 6360(F2 + 1.127\omega^{0.07969})^{1/1.49} \tag{Equation 2}$$

in which T is measured in K and ω is measured in kgH₂O/kgAir.

The F1 and F2 isopotential lines are further modified for non-idealities in the system by the use of two effectiveness values ϵ_{F1} and ϵ_{F2} (in Equations 3 and 4) as proposed by Banks as cited in (19).

$$\epsilon_{F1} = \frac{F1_D - F1_P}{F1_R - F1_P} \tag{Equation 3}$$

$$\epsilon_{F2} = \frac{F2_D - F2_P}{F2_R - F2_P} \tag{Equation 4}$$

This model takes the values of $\epsilon F1$ and $\epsilon F2$ as parameters (instead of computing them) and allows controls on a desired process air outlet humidity ratio. The model will determine the temperature of regeneration air required to obtain this ratio.

3.2 Air-to-air sensible heat exchanger

Type760 uses effectiveness - minimum capacitance approach to model an air-to-air heat exchanger for sensible energy heat transfer. The concept of air to air sensible heat exchanger is illustrated in Figure 3. The two air streams are called “fresh air” stream and the “exhaust air” stream exchange sensible heat so that to precondition outside (fresh). An amount of air equal to the outside air is rejected or exhausted from the zone and the two air streams are often separated by a membrane that allows energy to transfer from one stream to the other (19).

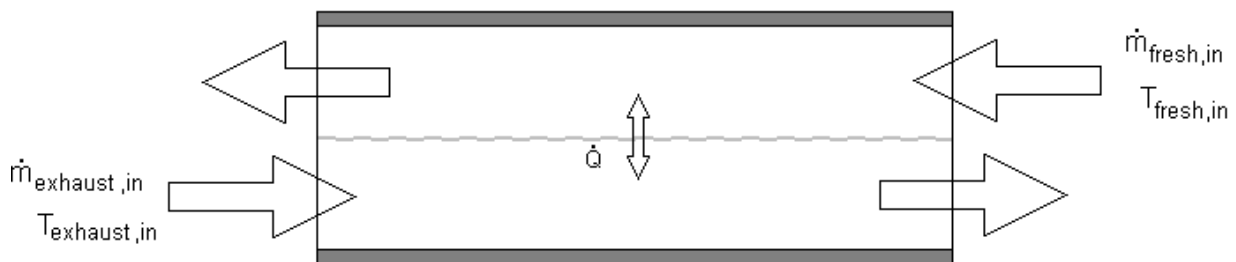


Figure 3 Air-to-Air sensible heat exchanger schematic (19)

3.3 Direct evaporative cooler

For direct evaporative cooler, Type506 models the inlet air and the saturation efficiency are the input parameters and calculates the outlet air conditions. The cooling process is assumed to be a constant wet bulb temperature process meaning that air enters and exits at the same wet bulb temperature (19). The outlet temperature is calculated using Equation 5:

$$T_{\text{AirOut}} = T_{\text{Air,in}} - \eta_{\text{saturation}} T_{\text{wbd}} \quad \text{Equation 5}$$

where T_{wbd} is the wet bulb depression and $\eta_{\text{saturation}}$ is the efficiency of the device. The wet bulb depression is defined as the difference between the inlet dry bulb air temperature and the inlet wet bulb air temperature.

3.4 Indirect evaporative cooler

In an indirect evaporative cooler, the air stream being cooled (the primary air stream) does not actually come into physical contact with the wetted medium as it does in a direct evaporative cooler. In an indirect evaporative cooler, a secondary air stream passes over a wetted surface, evaporating some of the moisture from that surface and therefore cooling the air. The primary and cooled secondary streams exchange sensible heat transfer.

The Type757 for which the primary and secondary air streams are input parameters and the device effectiveness as a function of primary stream inlet air dry bulb temperature and secondary stream inlet air wet bulb temperature has been used. The model calculates outlet air conditions and assumes that the secondary air stream process is a constant wet bulb temperature process meaning that air enters and exits at the same wet bulb temperature. The device effectiveness, which is

defined by Equation 6 below as per ASHRAE Handbook of Applications 1999 as cited in (19).

$$\varepsilon = \frac{T_{primary,in} - T_{primary,out}}{T_{primary,in} - WB_{primary,in}} \quad \text{Equation 6}$$

4.0 Basic DEC system validation

To validate the system performance, a basic one-stage DEC system in ventilation mode from Figure 1 was simulated in TRNSYS 17. The validation study was made to be compared with published simulation data by Dezfouli et al. (7). The test condition was set for a typical room with a cooling load of 3.51 kW, flow rate for process and regeneration side was 870 kg/hr and humidity ratio set point was 0.01 kg/kg. The simulation results were then plotted for outdoor design condition of 30°C DB, 0.0200 kg/kg humidity ratio and 74% relative humidity. The state points of process and regeneration air at all points for current study were then compared to Dezfouli et al. (7) as shown in Table 1. It was noted that the simulation results of the current work show a large deviation from the published data and have percentage difference between 1.1% to 22% for temperature, 0% to 13% for humidity ratio and 4% to 41% for relative humidity.

It was clearly observed that the temperature at desiccant wheel demonstrated the most significant percentage difference. This deviation is also found in the dehumidification process in SRDW at state point 2. It was identified as the dehumidifier used (Type683) by Dezfouli et al. (7) was obsolete and this type was models using design condition as input parameters and calculate the dehumidifier potential effectiveness as output. Another potential difference was the potential effectiveness of the dehumidifier values (F_1 and F_2) used by Dezfouli et al. (7) were 0.1 and 0.07 respectively. Thus, dehumidifier employed in this simulation was the Type1716a where potential effectiveness values of F_1 and F_2 are assigned by the users. Therefore, the chosen values of potential effectiveness (εF_1 and εF_2) for a moderately efficient desiccant wheel were selected as 0.05 and 0.75 respectively in this study based on Jani et al. (20).

Deviations between studies have also been identified in terms of the mass and energy balance between process side at states 1 to 2 and regeneration side at states 8 to 9 of the dehumidifier. The deviation in results may be due to changes in the humidity ratio and ambient temperature when the heat and mass are transferred from the regeneration side to the process side. It is likely the process of pre-cooling and pre-heating the regeneration air in the solid rotary desiccant wheel affects the humidity ratio and the temperature because of the leakage of air in the air passage. The state points for the system considered in this study and that of Dezfouli et al. (7) were plotted in psychrometric chart as shown in Figure 4. It was shown that the psychrometric chart of current research had similar pattern to the published literature examining basic one-stage desiccant evaporative cooling systems.

Table 1 Data comparison between current research results and published data from literature.

INPUT AND PARAMETER VALUES									
Room cooling load	3.51	kW							
Flow rate process and regeneration	870	kg/hr							
Dehumidifier humidity ratio set point	0.01	kg/kg							
Desiccant wheel potential effectiveness	F1	0.05							
	F2	0.75							
Ambient air DB temperature	30	°C							
Ambient air relative humidity	74	%							
Ambient air humidity ratio	0.0200	kg/kg							
State Point	DB Temperature °C			Humidity ratio kg/kg			Relative humidity %		
	Current research	Dezfouli et. al (7)	% difference	Current research	Dezfouli et. al (7)	% difference	Current research	Dezfouli et. al (7)	% difference
2	60.4	49.5	22.0	0.0100	0.0100	0.0	7.8	13.2	-40.7
3	35.7	31.8	12.3	0.0100	0.0100	0.0	27.3	33.9	-19.6
4	24.2	22.0	10.0	0.0137	0.0148	-7.4	77.6	82.2	-5.6
5	32.8	30.8	6.5	0.0152	0.0161	-5.6	51.1	54.1	-5.6
6	27.4	25.0	9.8	0.0176	0.0196	-10.2	84.4	87.9	-4.0
7	51.6	45.6	13.2	0.0176	0.0196	-10.2	23.0	28.0	-17.8
8	82.5	81.6	1.1	0.0176	0.0196	-10.2	5.9	5.5	7.1
9	51.2	50.0	2.4	0.0262	0.0302	-13.2	35.6	33.0	8.0

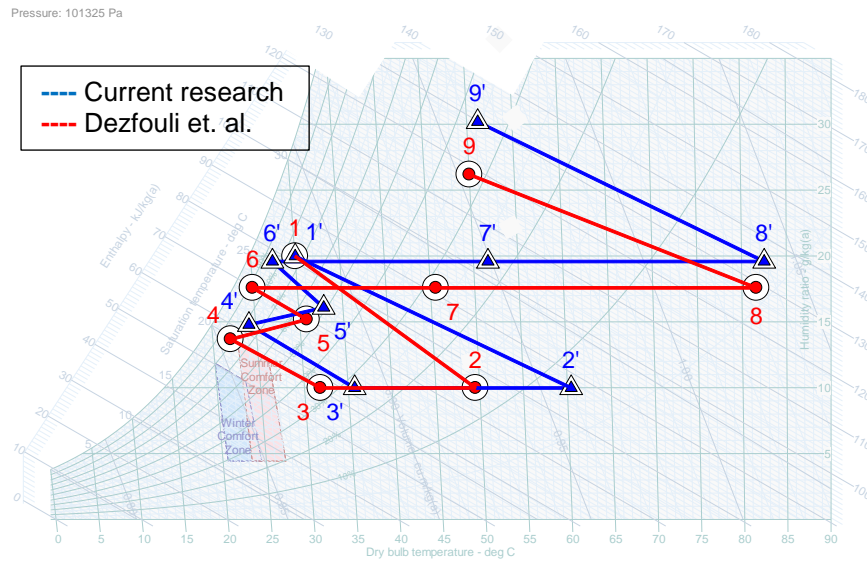


Figure 4 Psychrometric comparison between the results of the current research with the simulation results of Dezfouli et al. (7).

5.0 Simulation case study description

In this study, after the basic DEC system was validated as shown above in Section 4.0, the comparative performance of four types of desiccant evaporative cooling system were evaluated using TRNSYS 17. DEC systems consist of four types of one-stage solid desiccant dehumidifier with two-stage evaporative cooling systems of direct, indirect, direct/indirect and indirect/direct. Each DEC system configuration is explained in the next sections.

5.1 Environment set up

A test room was set up for a 1.75 kW (1.31 kW sensible and 0.44 kW latent) cooling load. The systems were simulated in the hot and humid weather of Malaysia. In this study, the outdoor conditions for Kuala Lumpur, Malaysia with reference to ASHRAE (20) has selected at 2% of annual cooling climate data which gave 33.5°C design dry-bulb (DB) temperature and relative humidity (RH) of 60%. The ideal indoor design condition according to the ASHRAE (20) is a DB temperature between 24°C to 26°C, a relative humidity of 50% to 65% and a humidity ratio of 0.0098 kg/kg.

5.2 Comparison of four configurations

Four types of configuration were selected to evaluate the performance of desiccant evaporative cooling systems. The DEC system was developed as two-stage evaporative coolers were arranged in series as shown in Figure 5. The arrangement of the desiccant wheel and heat exchanger remained as before as was shown in Figure 1.

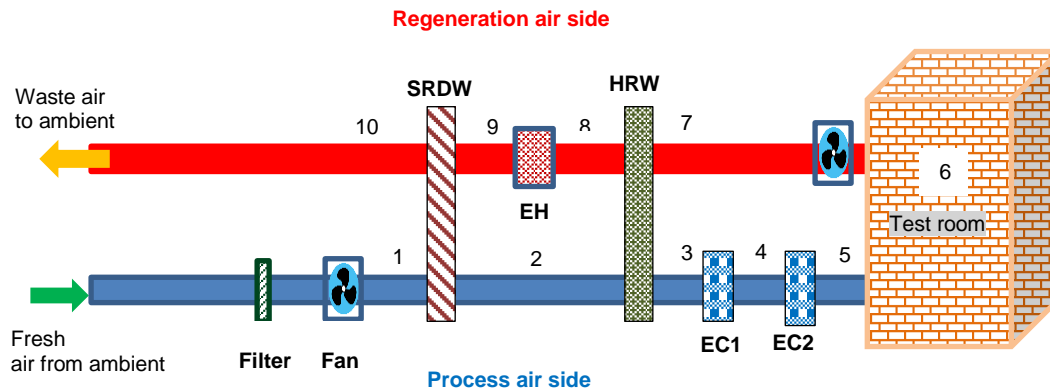


Figure 5 Desiccant evaporative cooling system arranged in series

As can be seen from Figure 6, direct/direct evaporative cooling (DDEC) used two direct evaporative coolers to cool the supply air to the room. The second configuration is the direct/indirect evaporative cooling (DIDEC) system illustrated in Figure 7. In the DIDEC, the input secondary air was collected from return air room and output secondary air was supplied to the HRW. The third configuration is an indirect/indirect evaporative cooling (IDIDEC) system as seen in Figure 8, which works by recycling return air as the input secondary air to the indirect evaporative cooler twice before supplying it to the HRW. The fourth configuration is an indirect/direct evaporative cooling (IDDEC) system (Figure 9) which uses the same concept of recycling the return air for secondary air intake in the indirect evaporative cooler. For conventional systems, a quite disseminated rule of thumb refers to a flow rate per cooling load of $193 \text{ (m}^3\text{/h)/kW}$ (400 cfm/ton) (21). However, this value is not considered suitable for desiccant systems given that through the desiccant operation cycle, the temperature of supply air to the building is significantly higher, compared to that of conventional systems. For this reason, higher values for the air flow rate were proposed for use in desiccant systems in the relevant literature. According to Panaras et al. (21) the air flow rate, the regeneration temperature, the operation cycle and the subsystems level of performance constitute the design parameters of desiccant systems. Therefore, the flow rate of both process and regeneration side are selected at 1343.52 kg/hr disregards to the draft issues of air speed in the room. Each TRNSYS component's description and assumptions are summarised in Table 2.

One-stage desiccant dehumidifier with two-stage direct.direct evaporative cooler

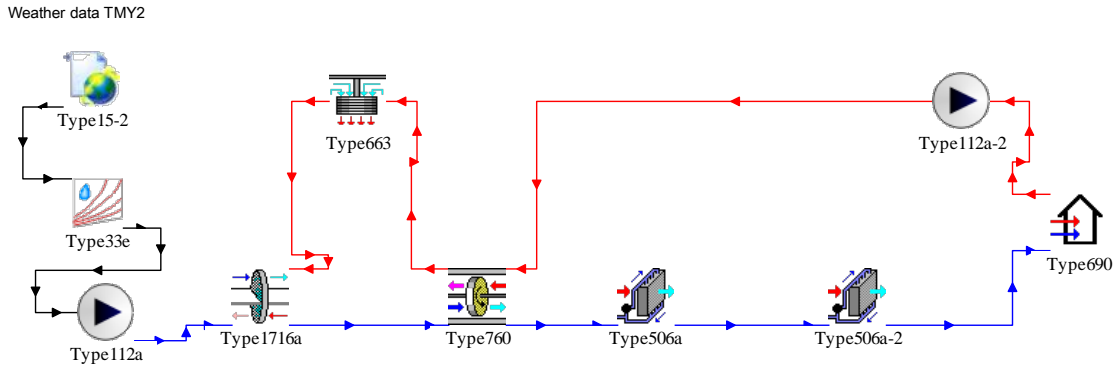


Figure 6 DDEC simulation layout

One-stage desiccant dehumidifier with two-stage direct/indirect evaporative cooler

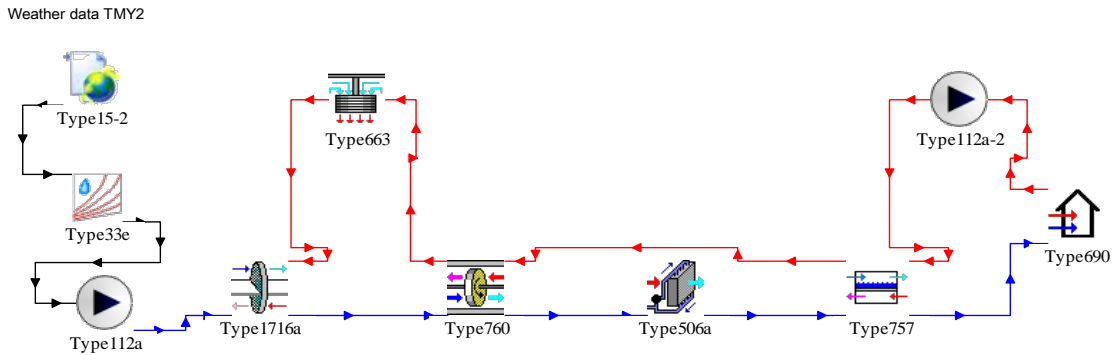


Figure 7 DIDEDEC simulation layout

One-stage desiccant dehumidifier with two-stage indirect.indirect evaporative cooler

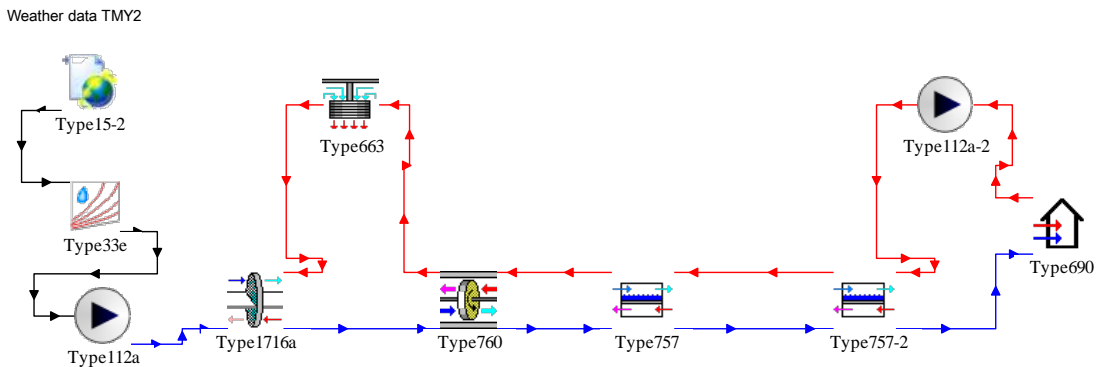


Figure 8 IDIDEDEC simulation layout

One-stage desiccant dehumidifier with indirect/direct evaporative cooler

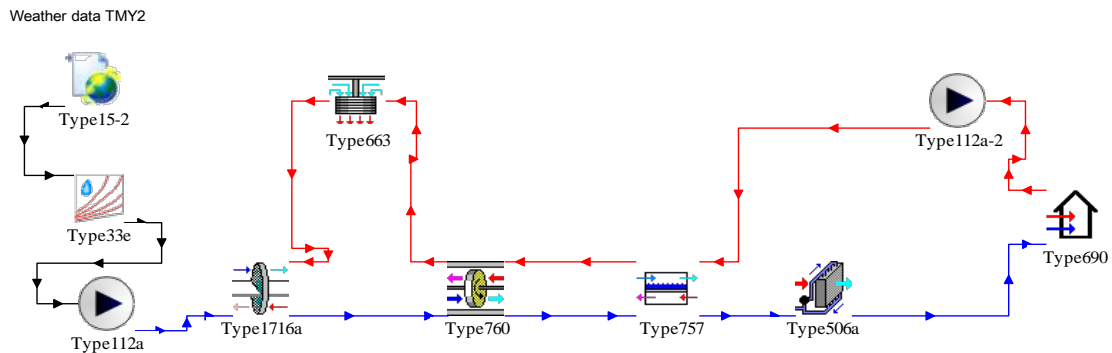


Figure 9 IDDEC simulation layout

Table 2 TRNSYS components description and assumptions

Type	Legend	Component description	Assumptions
Type15-2	-	Weather data processor	TMY2 Meteorom weather data
Type33e	-	Psychrometric	-
Type112a	Fan	Single speed fan/Blower with Humidity Effects	Flowrate 1343.52 kg/hr, rated power 1000 W, motor efficiency 0.85
Type1716a	SRDW	Solid rotary desiccant wheel	Dehumidifier effectiveness F_1 0.05, F_2 0.75, humidity ratio set point 0.005
Type760	HRW	Air-to-air heat recovery wheel	Sensible effectiveness 75.6%
Type506a	DEC	Direct evaporative cooler	Saturation efficiency 0.8
Type757	IDEC	Indirect evaporative cooler	-
Type690	-	Test room environment	1.75 kW cooling load
Type663	EH	Unit heater	Rated flowrate 1343.52 kg/hr Rated fan power 1000kg/hr Motor efficiency 0.8 Heater capacity 5kW Heater efficiency 1.0

6.0 Results and analysis of the performance of four DEC system.

To compare the performance of the system, the coefficient of performance (COP) of the system is calculated. The COP is defined as the rate of heat extracted divided by the rate of heat regeneration as cited by Daou et al. (9) where the bigger the value, the better the performance of the system. The formula can be written as follows:

$$COP = \frac{\dot{m}_p (h_1 - h_5)}{\dot{m}_r (h_9 - h_8)} \tag{Equation 7}$$

where \dot{m} is the mass flow rate of air (kg/hr), h is the enthalpy of air (kJ/kg) and the subscripts correspond to the state point in the desiccant cooling system.

6.1 Effect of ambient condition at 33.5°C DB and 60% RH

From the simulation results for the four configurations as mentioned above, ten selected state points of the systems are plotted on a psychrometric chart and shown in Figures 10. By referring to the psychrometric chart in Figure 10, in the DDEC system, the ambient air at state 1 is passed through the solid rotary desiccant wheel for dehumidification on the process side. Warm and dry air stream exits from the desiccant wheel at state 2. The air stream is then sensibly cooled in the air-to-air heat recovery wheel between state 2 and state 3. The cool and dry air is further cooled, and slight moisture is gained when passing through first direct evaporative cooler (EC1) from state 3 to state 4 and further cooled by second direct evaporative cooler (EC2) from state 4 to state 5 before entering the room. In the regeneration side, after leaving the room (state 6), return air passes through the heat recovery wheel (HRW) to recover heat from warm and dry air (state 7 to state 8). Between state 8 and state 9, it is further heated by an electric heater (EH) to a required regeneration temperature (state 9). From state 9 to state 10, regeneration air extracts moisture from the desiccant wheel (SRDW) and expels the moisture to the ambient at state 10. Other configurations having similar process except between state 3 to 7 where evaporative cooling process occurs.

Hence, the comparison of performance data for four configurations were tabulated in Table 3. As referred to in Table 3, the achieved room temperature (at state 6) of all four systems was below 26°C which met the recommended indoor comfort condition (20). However, as recommended by ASHRAE (20), only IDIDEC and IDDEC met the acceptable indoor comfort conditions of both the room DB temperature and humidity ratio of 26°C below and 0.0098 kg/kg respectively. The IDIDEC had the lowest humidity ratio in the supply and room air condition because of the humidity ratio set point in SRDW was set at 0.005 kg/kg where this too dry. This set point value can be adjusted higher in IDIDEC since no humidity imposed to the system in indirect EC. Therefore, it had the lowest regeneration temperature required compared to the other three configurations. The humidity ratio in the room of DDEC and DIDEC were above the comfort condition of 0.0098 kg/kg as stated in ASHRAE (20). It was observed that direct evaporative cooler will add moisture in the air, while indirect evaporative cooler will not. Therefore, the humidity ratio set point (HRSP) in desiccant wheel can be higher if an indirect evaporative cooler was used. The IDIDEC system has shown the highest COP value at 1.12 while the lowest was 0.59 for DIDEC system. The COP of DDEC system is 0.61 which was higher compared to experimental study of similar configuration carried out by Kodama et al. (23) with a COP of 0.540. The IDDEC system had a COP of 0.93. Based on this comparison of all four configurations, it was found that the IDIDEC and IDDEC systems have met the requirements of ASHRAE at selected input parameters.

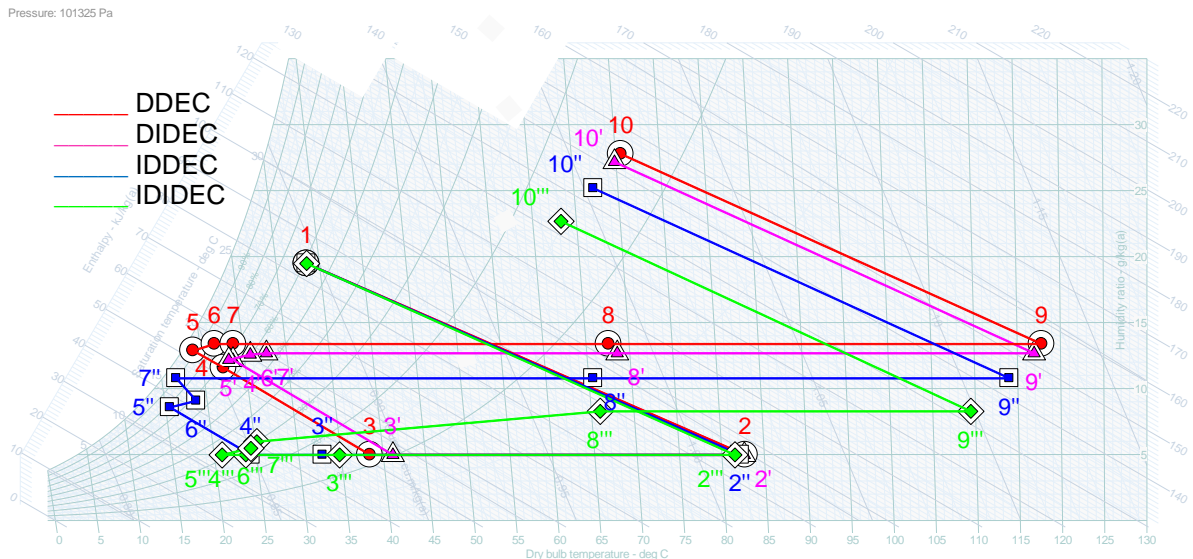


Figure 10 System psychrometric chart for a) DDEC; b) DIDE; c) IDDEC; and d) IDIDE.

6.2 Effect of ambient condition at 33.5°C DB and 95% RH

To evaluate which of the four configurations has the biggest impact in the humid weather, the ambient condition of a 33.5°C dry-bulb temperature and 95% RH were simulated alongside a consistent flow rate, humidity ratio set point and cooling load. The results are outlined in Table 4. From the results, it can be seen that DIDE, IDIDE and IDDEC had a room RH below 55%, while DDEC was above 65%. However, only IDDEC satisfied to room dry-bulb temperature below 26 °C.

7.0 Conclusions

In this study, four DEC system configurations were simulated. The model was simple and easy to implement. This study aimed to understand how one-stage dehumidification with a two-stage evaporative cooler can provide cooling in hot and humid climates especially tropical climates. The investigation of the performance of these four DEC systems has helped to identify the system that is most suited to satisfy the cooling load for hot and humid weather. Of the four, the configurations that met the requirements set out by ASHRAE (20) were the IDIDE and IDDEC system. Further research can be carried out into comparisons the effects of ambient temperature and relative humidity, flow rate, dehumidifier effectiveness and regeneration temperature and then determine the cooling capacity and COP for each system configuration.

Table 3 Performance data of four types of DEC system

Ambient condition	Dry-Bulb Temperature 33.5 (°C)			Relative Humidity 60%			Humidity Ratio 0.0195 kg/kg					
EC Types	DDEC			IDIDEC			DIDEC			IDDEC		
Description	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)
Supply Air	19.1	0.0129	93	20.9	0.0050	33	23.1	0.0122	69	15.5	0.0086	78
Room	21.7	0.0134	82	24.4	0.0055	29	25.7	0.0126	61	18.7	0.0091	67
Regeneration Inlet Air	117.8	0.0134	1	109.4	0.0083	1	116.9	0.0127	1	114.0	0.0108	1
COP	0.61			1.12			0.59			0.93		

Table 4 Supply air, room air and regeneration inlet air temperature at 95% ambient relative humidity

Ambient condition	Dry-Bulb Temperature 33.5 (°C)			Relative Humidity 95%			Humidity Ratio 0.0317 kg/kg					
EC Types	DDEC			IDIDEC			DIDEC			IDDEC		
Description	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)	DBT (°C)	HR (kg/kg)	RH (%)
Supply Air	24.7	0.0177	90	23.0	0.0050	29	30.3	0.0158	58	18.1	0.0097	75
Room	30.0	0.0182	68	29.1	0.0055	22	32.5	0.0163	53	25.8	0.0101	49
Regeneration Inlet Air	181.0	0.0182	0	170.1	0.0101	0	179.3	0.0167	0	175.3	0.0135	0

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