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Energy and Comfort Evaluation of Fresh Air-Based Hybrid Cooling System in Hot and Humid Climates

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Abstract: Maintaining mechanical ventilation has been identified as a potential strategy for reducing the risk of virus infections. However, in hot and humid climatic conditions, delivering fresh air to a building comes at an energy cost and could impact occupant comfort due to the persistent need for simultaneous cooling and dehumidification. In this paper, the performance of a novel hybrid air conditioning system that handles fresh air is studied. In this system, dehumidification is accomplished by a solid desiccant dehumidifier coupled with a cooling coil integrated with the cooling tower of an existing chiller system. Using the data available from an operational desiccant cooling system, a system-level model has been developed and validated to study the potential application of the system in hot and humid climates. The study found that such a system is effective in delivering sensible cooling in all types of climates; thanks to the two-stage cooling in cooling coil and chilled water coils, respectively. However, the system is effective in delivering thermal comfort in regions where the climate has a relatively moderate ambient humidity. For the tropical cities of Darwin, Kuala Lumpur and Bangkok, the system can provide comfortable temperatures, but faces challenges in keeping the humidity within the comfort zone. The system electrical coefficient of performance (COP) is higher than that of refrigerative systems. This system also has the benefit over the refrigerative system of the supply air, which is entirely fresh ambient air and is expected to improve the indoor environmental quality largely.

Keywords: air conditioning; dehumidification; thermal comfort; desiccant; ventilation; solar cooling

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1. Introduction

A recent report released by the World Health Organization (WHO) has identified various factors for improving indoor conditions in the context of COVID-19 [1]. This report identifies mechanical and natural ventilation as key strategies for maintaining good indoor ventilation in buildings.

However, mechanical ventilation systems consume energy to pre-treat the air depending upon the climatic conditions. In hot and humid regions, outside air needs to be dehumidified and cooled to maintain comfort conditions, adding to the energy use of the HVAC systems [2].

Irrespective of the ventilation specific energy use of HVAC systems, the need for air conditioning in the developing world—which happens to be the regions characterized by hot and humid climates—has been identified as one of the key drivers of increased electricity demand [3]. This is particularly true for the conventional refrigerative systems known for their high electricity demand and consequent environmental impacts. This situation has prompted researchers worldwide to look for viable alternatives. One such alternative is the use of thermally driven air conditioning systems, which consume less electrical energy and can rely on locally available heat sources such as waste heat and solar energy. Solid desiccant evaporative air conditioning presents such an opportunity as well as technical and economic challenges [4–14]. Researchers [15,16] have presented various possible system configurations deemed appropriate for various climatic conditions. The thermodynamic equations of the state of the system for the humid climate applications were discussed in [17].

A solid desiccant evaporative air conditioning system (SDEAC) provides thermal comfort by employing the dehumidifying capability of a solid desiccant system and the cooling capability of an evaporative cooling system. Further, these systems are open cycle systems using fresh air and hence suitable for managing ventilation requirements in buildings. Such a system can potentially reduce the reliance on electrical energy for thermal comfort provision. To be able to compete technically and economically with the conventional refrigerative systems, SDEAC must be properly designed to achieve the following: (1) to minimize the parasitic energy consumption, (2) to minimize the use of regeneration heat, and (3) to be on par with the conventional refrigerative systems in delivering the thermal comfort. To achieve the above goals, several system configurations have been proposed [15]. Despite the voluminous literature available on desiccant air conditioning systems [11,16,18], studies addressing the thermal comfort aspects of system performance are limited.

Previous studies [11,19] have found that a single-stage SDEAC system with several configurations is not able to adequately satisfy the thermal comfort requirements of the conditioned space. In particular, the values of the room humidity ratio are outside of the comfort zone for some of the time. A single-stage SDEAC system with an enthalpy exchanger proposed by Henning [15] has also been found to be unable to satisfactorily bring the room comfort conditions into the comfort zone, including in a city with a less humid climate such as Brisbane and Townsville [20].

The standard desiccant cooling cycle mentioned in [15] was recently studied to investigate the system performance for Brisbane's climate zone [19] with a better electrical COP compared to that of the refrigerative system, but with thermal comfort conditions of the conditioned spaces not fully satisfied. Two recent studies conducted on a two-stage desiccant system were also carried out [21,22]. In these studies, the psychrometric process of the two-stage SDEAC with various configurations was presented; however, the actual air conditions in the rooms are missing. Instead, the authors studied the conditions of air as the air undergoes conditioning from one component to another until it is supplied to the room. While this gives the real conditions of air as it enters the room, what is most important is the resulting conditions of air as it interacts with sensible and latent heat gains of the occupied space. As such, system modelling involving sensible and latent heat gains requires modelling air-conditioned space where the plots of air temperature vs. humidity ratio can be obtained. Using this approach, the resulting conditions of air can be directly checked against the standard thermal comfort zones, such as developed by ASHRAE [23]. Many of the proposed approaches use a system configuration that employs indirect evaporative cooling systems. Despite their benefits, these systems are still not economically viable, and hybrid air conditioning systems that employ active dehumidification with conventional cooling systems will have market attractiveness as long as thermal comfort considerations are achieved.

Previous investigations on hybrid cooling systems involving desiccant systems include heat pump-assisted desiccant cooling [6,24], ground assisted desiccant cooling [25], desiccant based ejector cooling [26] and PCM integrated desiccant cooling [27]. However, the proposed system is different from systems in the literature as this system combines a solid desiccant dehumidifier with a conventional cooling coil integrated with the cooling tower of an existing chiller system and this is first of its kind in the available literature.

In this context, the contributions of our study are:

- Development and performance evaluation of a solid desiccant driven simple hybrid air conditioning system in humid climates. In this system, air is first dehumidified using a desiccant wheel, and then the air is cooled by utilizing cooling water from cooling towers used for operational chillers. Further sensible cooling is achieved by a chiller.

- Low-temperature heat is used for regeneration. Thus, this system can operate with solar heat sources such as flat plate and evacuated collectors and low-temperature waste heat sources.
- The evaluation of the thermal comfort that can be delivered by this hybrid system designed for hot and humid climatic conditions, showing good potential.

This paper evaluates the dehumidification capability of the proposed hybrid cooling system exposed to the climatic data of Brisbane, Townsville, Darwin and Kuala Lumpur representing cities (regions) with sub-tropical/tropical climatic conditions.

To place the paper's main message into a proper context, the readers should be reminded that conventional refrigerative systems installed for air-conditioned buildings are not always able to deliver the thermal comfort required for a conditioned space without the need for extra energy. In hot and humid climates where a system is sized to satisfy the design load, it may suffer either (a) energy inefficiency to maintain the room design conditions or (b) thermal discomfort due to very high humidity levels at part load conditions. Situation (a) occurs in a constant air volume (CAV) system where reheating is part of the system to maintain comfort at part load. Situation (b) is encountered in a variable air volume (VAV) system where at part load—outdoor air temperature is well below the design temperature, but the humidity ratio is basically constant. In such a case, the control system responds by reducing the chilled water flow, which in turn causes the system dehumidifying capacity to reduce. This results in the space humidity levels rising beyond the comfort boundary [28–30].

2. The Proposed Configuration

The proposed solid desiccant-based hybrid cooling system is shown in Figure 1. Dehumidification is carried out on the supply side solely by a desiccant wheel. The supply air temperature after passing through the desiccant wheel increases due to heat adsorption. In order to reduce this, a cooling coil (CC) supplied with water from a cooling tower is used. This removes the bulk of the sensible heat gained after the dehumidification. An optional sensible heat recovery (SHRW) can provide temperature reduction as well. This configuration is similar to that proposed by previous researchers [31,32]. However, unlike La et al. [31]., this study considers only a single-stage dehumidification system to reduce the parasitic electricity consumption and does not use an evaporative cooler (that has the potential to increase the humidity of air stream) as proposed by Hands et al. [32]. Monitoring data from the operational system in Australia has been used for validation of the model developed for this study (see Section 4).

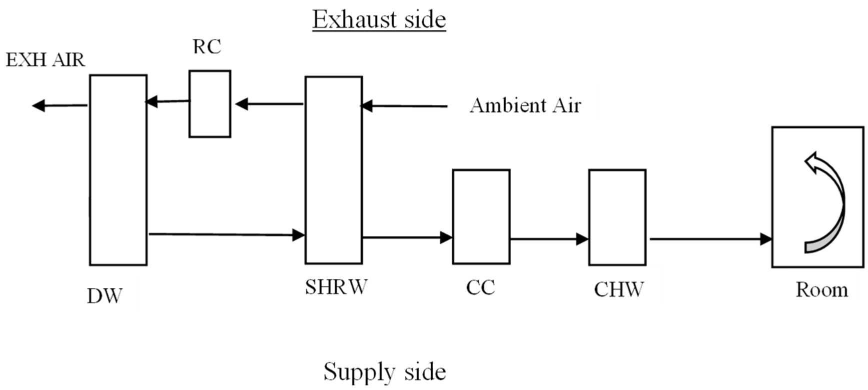


Figure 1. A proposed solid desiccant-based hybrid cooling system.

Final cooling is accomplished by the chilled water coil placed prior to the supply side of the building. On the exhaust side, the ambient air enters the regeneration (heating) coil

(RC 1) and is heated up. It then passes through the optional energy recovery wheel and is heated. The regeneration heat source can be low-temperature waste heat or solar heat delivered by flat plate or evacuated collectors.

As opposed to the familiar desiccant cooling system along with evaporative or indirect evaporative cooling [7], the system does not use the direct evaporative cooler to cool down the air; hence there is no moisture added to the supply air stream during the system operation.

Figure 2 shows the typical psychrometric conditions of the proposed cooling system. As shown, on the supply side (blue lines), air enters the desiccant wheel (DW) at 30.4 °C and a humidity ratio of 18.3 g/kg—point 1. In the desiccant wheel, the air is heated but is also dehumidified and leaves the DW at [50.7 °C, 11.3 g/kg]—point 2. In the sensible heat recovery wheel (SHRW), the air is further cooled sensibly to 42.2 °C—point 3. The process in the cooling coil (CC) is also sensible, and air leaves the coil at 25.3 °C at the same humidity ratio—point 4. Further cooling (and possible dehumidification) is attained as the air enters the chilled water coil (CHW), which bring the temperature down to 17.6 °C—point 5. The dehumidification can occur in the CHW if the air is cooled below its dew point temperature.

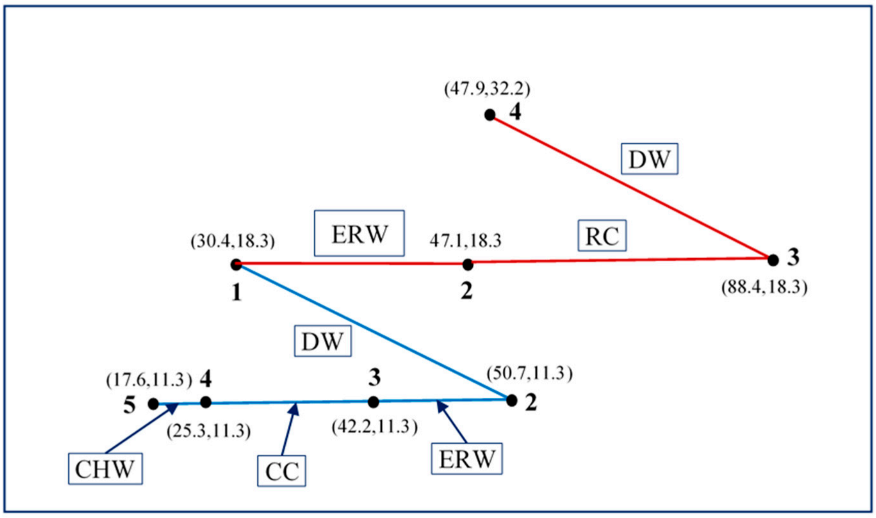


Figure 2. A typical psychrometric process of the proposed cooling system.

On the exhaust side—red lines—the fresh air enters SHRW and is heated to 47.1 °C. The significant rise in the air temperature occurs in the regeneration coil, where air leaves at 88.4 °C with a constant humidity ratio. The air temperature is then reduced as soon as it removes the moisture from the desiccant.

3. Methodology and Formulation

This study evaluates the thermal comfort and energy performance of the proposed hybrid system. To achieve this, the thermal comfort conditions in a conditioned space in the form of plots of space temperature vs. humidity ratio is compared with the ASHRAE comfort zone [22]. The fact that there is no perfect system able to satisfy all the conditions all year round—taking into account extreme weather conditions—it is decided to set the 75% satisfaction rate for the comfort conditions during the cooling season as an acceptable system for a given climate region. Both room temperature and humidity ratio mentioned above represent the state of the condition of the air at a particular time (hour) during the conditioning period. In this study, an entire block of four office spaces (offices 1–4) in a building—Figure 3, scheduled for conditioning from 09.00–17.00, is modelled as being conditioned by the proposed system. Each office has dimensions of (L × W × H) of 4 m × 4.4 m × 2.75 m. Such an office block can, in reality, represent a private surgery building in—say—a shopping center. All offices are assumed to operate during the same period with the identical room temperature setting and room occupancy. As such, it can be modelled as one single zone in the model. The system is turned on during the scheduled period to maintain the room temperature at 25 °C (+0.5 °C, −1.5 °C). It is also assumed that during the system operation, the air volume flow rate of supply air to the room is constant for both systems and may vary for different cities. The chosen flow rate will depend on the satisfaction of the two performance indicators discussed above. Once the flow rate is established, the thermal and electrical COPs of both systems are recorded. These COPs will then be compared.

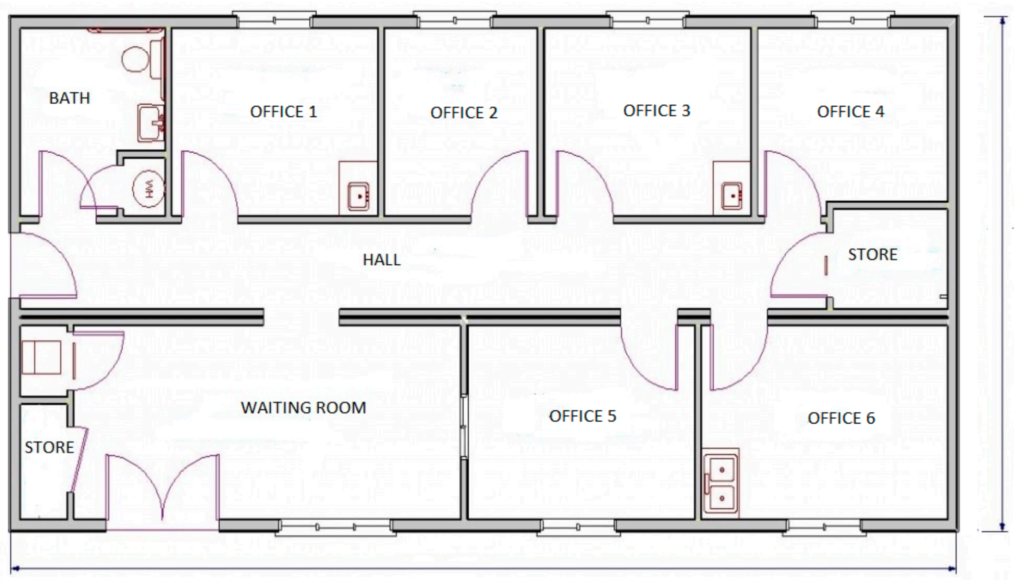


Figure 3. A floorplan of the office spaces being modelled.

The system operation is modelled in TRNSYS software. In this study, TRNSYS's TYPE56—multi-zone building model—was used, as opposed to TYPE690 used in [11]. As such, the room sensible and heating loads are not directly input into the model; rather, the sensible and latent heat gains are entered. These inputs, detailed in [19], consists of (1) a number of people occupying the space during the scheduled space conditioning period, their level of activities and the appliances and lighting. The following are the assumptions made.

- The weather data used in the model for each of the cities in each of the countries are accurate.
- The building used in the TRNSYS model is a multi-zone building.
- The building is constructed using bricks and external and internal walls are thermally insulated. Windows are single glazed.
- The results produced from the Trnsys model is for the selected building.

The values of some of the quantities are calculated within the relevant TRNSYS components, while others are formulated based on these values. The following quantities are formulated to enable the calculation of electrical and the thermal system of the system.

Supply and exhaust fan power, PF [W], can be expressed as follows.

$$P_F = \Delta p \times q/\eta_F \quad (1)$$

where:

Δp = total pressure increase in the fan (Pa, N/m^2);

q = air volume flow delivered by the fan (m^3/s);

η_F = fan efficiency.

The desiccant wheel power input to drive the wheel is estimated to be 0.12 kW, which is the value used in the validation data.

The electrical power input to pump the fluid through cooling coils CC1 and CC2 and the chilled water coils CHW can be calculated as follows.

$$P_P = Q \times dP\ /(3600 \times \eta_P \times \eta_m) \tag{2}$$

where:

Q = volume flow rate, m^3/h;
dP = Differential pressure across the pump, kPa;
ηP = pump efficiency;
ηm = motor efficiency.
Electrical power consumed by the system:

$$EC = P_{SA} + P_{EA} + P_{DW} + P_{SHRW} + P_{pumps} \tag{3}$$

The electrical COP is defined as the electrical energy consumed by the system (EC) for a given conditioning (cooling and dehumidifying) load (L).

$$COPEL = L/EC \tag{4}$$

The electrical energy consumed by the system consists of the electrical energy consumed by each component as previously derived for the whole operating period of the system.

Cooling load (L) is estimated as the enthalpy change in the supply air as it moves from state 1 to 5, as shown in Figure 2.

Electrical COP is used to compare the cooling system eligibility as an alternative to the conventional refrigerative air conditioning systems.

The thermal COP is defined as the ratio of the conditioning load in kW (L) to the thermal energy required for regeneration in kW (HR):

$$COP_{TH} = L/HR \tag{5}$$

Both COP values are presented on an annual basis of cooling.

The summary of the main inputs to the TRNSYS model is shown in Table 1. Any change applied to particular cities being studied is mentioned in Section 5 (Results and Discussions).

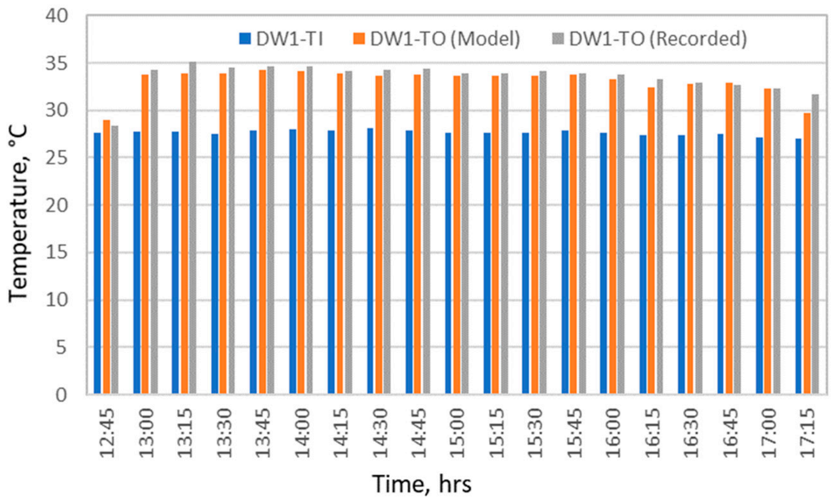
Table 1. Main inputs to the TRNSYS models.

Room temperature setpoint	25 °C (+0.5, −1.5)
Conditioned spaces	The whole office block (Offices 1–4)
Supply air mass flowrate	2640 kg/h
Office space conditioning schedule	Living: 09.00–17.00
Infiltration	0.5 ACH (air change per hour)
Regeneration temperature set point	90 °C (Brisbane, Townsville, Darwin, Kuala Lumpur, Bangkok)
Desiccant wheel	Power consumption 0.2 kW
Energy recovery wheel	Power consumption 0.1 kW
Fan power	See Equation (1)
Pump power	See Equation (2)—with an assumption that a total head of 5 m is available to tap the cooling from cooling and chilled water coil. Pump efficiency = 0.7
Airflow rate at the regeneration side	1320 kg/h
Cooling/chilled water flow rate	1500 kg/h
Weather data used	Brisbane, Townsville, Darwin, Bangkok, Kuala Lumpur

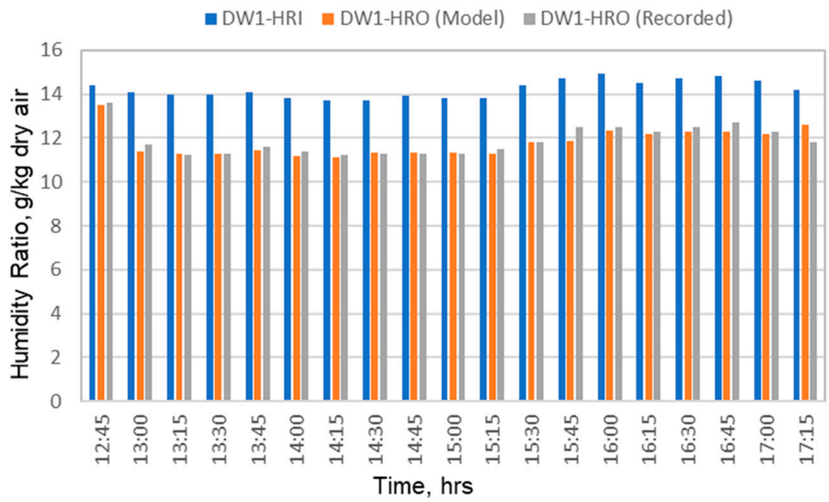
4. Model Validation

__TRNSYS is a well-known modelling framework used in the thermal system modelling community for over two decades. Hence, components, such as cooling and chilled water coils are standard heat (and mass) transfer components whose characteristics have been well known, are available in TRNSYS. However, desiccant wheel models are to a large extent dependent upon parameters such as the type of material used in the wheel and the regeneration temperature and face velocity. Hence, data from an operational two-stage desiccant system have been used to validate the desiccant wheel model used in our study. The data is taken from a large scale solar trigeneration system fitted with a solar desiccant cooling system installed at the Hamilton campus of the New South Wales Technical and Further Education (TAFE) commission in Newcastle, Australia. This system is described in the article by Hands et al. [32].

Figures 4 and 5 show the temperature and humidity ratio of supply air comparisons between the test data and our model for two typical days when the operating system functioned in the dehumidification mode. As seen, the results of both the model and the recorded values are in close agreement for both days. The average percentage error is determined to be less than 4%.

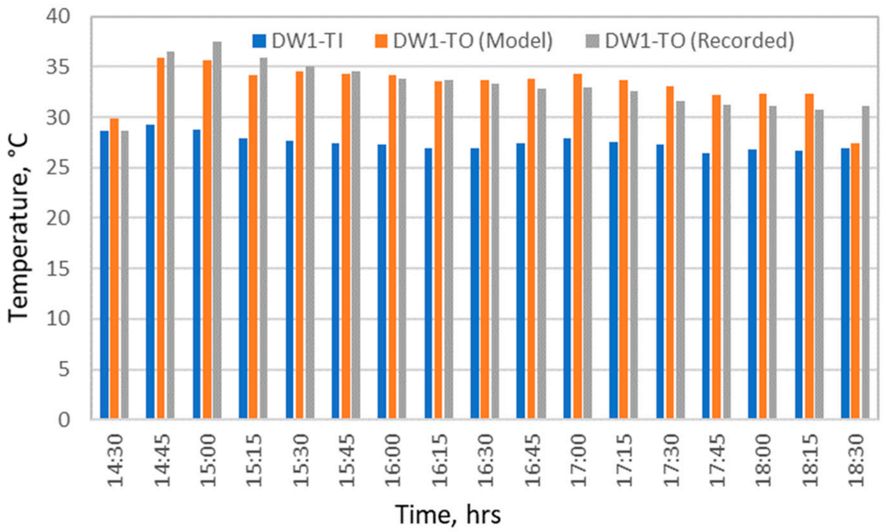


(**a**)



(**b**)

Figure 4. (**a**) Modelled and recorded values of (**a**) temperature across desiccant wheel (DW1) for day I, and (**b**) modelled and recorded values of humidity ratio across desiccant wheel (DW1) for day I.



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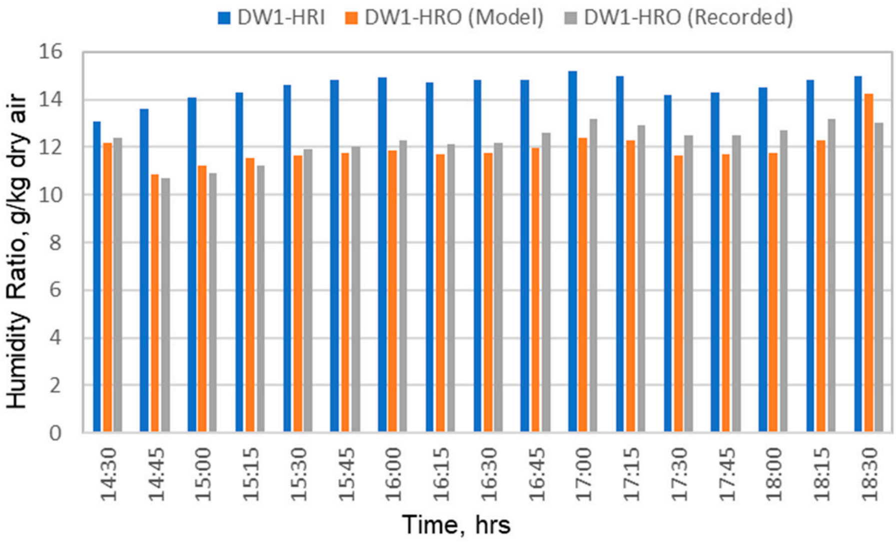

(**b**)

Figure 5. (**a**) Modelled and recorded values of temperature across DW1 for day II, and (**b**) modelled and recorded values of humidity ratio across DW1 for day II.

5. Results and Discussion

This section presents details of the system's operation in various cities with hot and humid conditions, viz Brisbane, Townsville, and Darwin (Australia) and Kuala Lumpur (Malaysia). A detailed study has been carried out to study the effect of regeneration temperature and the presence of an SHRW on the system performance.

The results of the study are presented as two sets of parameters. The first part is an assessment of the thermal comfort delivery capability of the system and the second part is the system energy performance indicator, expressed as electrical and thermal COP. Ideally, a system should be able to satisfy both requirements; however, in some circumstances, the satisfaction of either performance indicator may suffice. Results have been explained in detail for one location (i.e., Brisbane), and a summary of the results for other locations have been provided.

Figure 6 shows the plot of temperature vs. humidity for the ambient condition for Brisbane. Figure 7 shows the plots of the room temperature vs. humidity for the Brisbane climate for a system where a sensible heat energy recovery wheel (SHRW) was included. As shown, the system satisfies the room temperature requirements during its entire operation.

On the humidity requirement, the system is able to provide an acceptable humidity ratio for 80.2% of its operating hours. This is more than a 200% improvement compared to the use of an untreated air conditioning system.

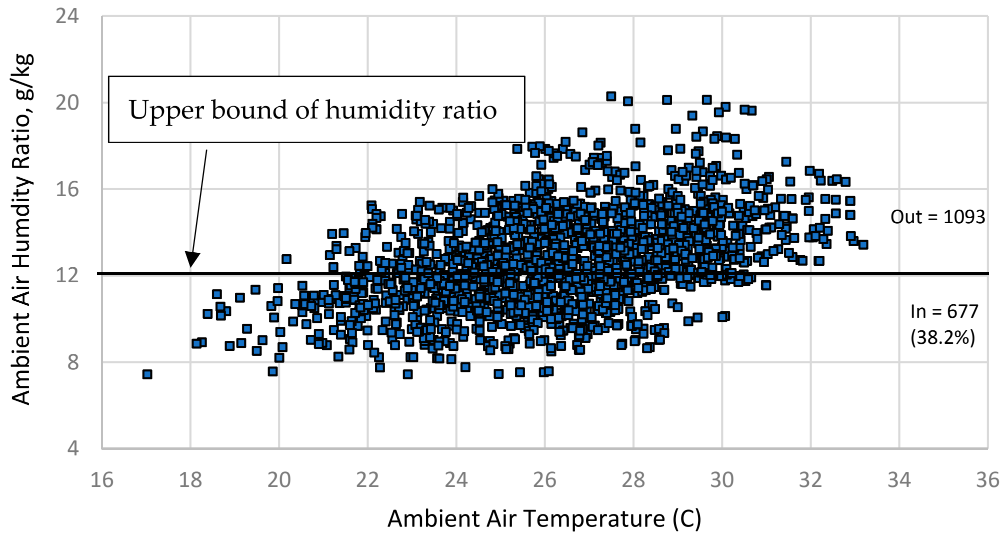

Figure 6. Plots of room temperature vs. humidity for Brisbane (Ambient condition).

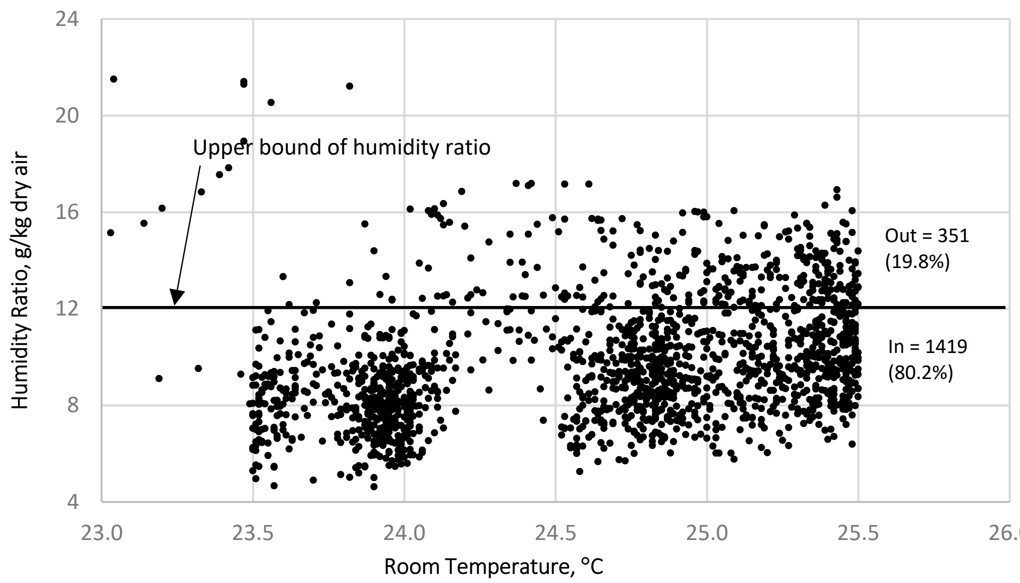

Figure 7. Plots of room temperature vs. humidity for Brisbane (system configuration with SHRW, regeneration temperature 90 °C).

For approximately four to five months of the year, the system does not operate as the temperature in the building does not exceed the upper-temperature limit of 25.5 °C, therefore the system will not commence operating to lower the temperature and humidity in the building. This may then also affect the humidity ratio of the air in the building, causing a higher level of thermal discomfort for the occupants in the building. The average room temperature was 24.6 °C, whilst the average room humidity ratio was 9.9 g/kg, which is well below the ASHRAE's maximum limit of 12 g/kg.

Figure 8 shows the summary of hourly humidity, temperature conditions inside the building for various selected cities such as, Kuala Lumpur, Townsville and Darwin. Each of these figures shows the percentage number of hours during which the humidity level in the building is in the comfort range.

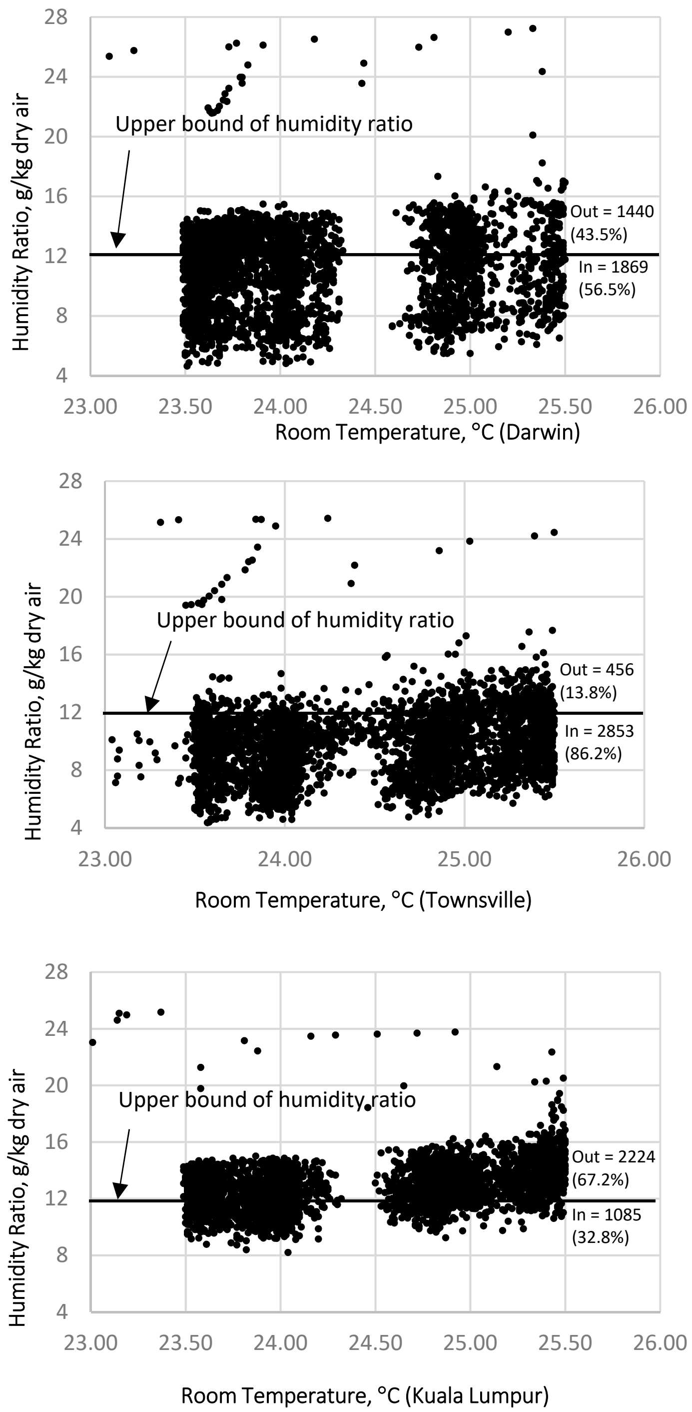

Figure 8. Summary of hourly humidity, temperature conditions for various cities.

Table 2 shows the summary of the electrical coefficient of performance (COP) values of the hybrid system calculated for whole year's operation in various cities. The COP of the system range from 6.86 to 8.37 with highest COP recorded for Kuala Lumpur and lowest for Darwin which is the hottest city among the chosen cities with high humidity. At this context it should be noted that these values are much higher than conventional air conditioners for which the COP is between 3–7.

Table 3 shows the summary of thermal comfort indicators of the system operating in various cities. For all the cities, the system can provide comfortable temperature inside the selected building all the time. Additionally, the system can provide comfortable humidity for most of the time during the year. However, the system can provide comfortable humidity in buildings in cities with moderately humid climates such as Brisbane and Townsville than cities with highly humid climates.

Table 2. Summary of energy performance indicators of the system operating in various cities.

City	COP_{Th}	COP_{EL}
Brisbane	0.5	7.75
Townsville	0.49	7.53
Darwin	0.46	6.86
Kuala Lumpur	0.53	8.37

Table 3. Summary of thermal comfort indicators of the system operating in various cities.

City	TCS [1]—Temp	TCS [2]—Hum
	(%)	(%)
Brisbane	100	80.2
Townsville	100	86.2
Darwin	100	56.5
Kuala Lumpur	100	32.8

[1] Thermal comfort satisfaction based on room temperature. [2] Thermal comfort satisfaction based on room humidity ratio.

6. Conclusions

A novel, two-stage solid desiccant-based fresh air handling hybrid cooling system configuration has been studied for its operational benefits for various tropical climates. A low temperature (less than 100 °C) heat source has been used for regenerating the system. The dehumidifier model has been validated with test data. The study has evaluated the suitability of this air conditioning configuration for humid climatic conditions viz. Darwin, Townsville, Kula Lumpur, and Brisbane.

The system operating in all cities satisfies the space temperature requirements for the whole operation. This system also performs well in terms of the humidity level requirement in Brisbane and Townsville, where the ambient humidity is relatively mild. However, for the cities with more challenging humidity levels, the system could not satisfy the humidity requirements all year round.

The annual electrical COP of the system was determined as 7.75, 7.53, 6.86 and 8.37, respectively, for the selected building under the climatic conditions of Brisbane, Townsville, Darwin and Kuala Lumpur, respectively, which is far better compared to the conventional air conditioning system. Thus, the system delivers electrical energy saving benefits for all the locations compared to a compressor-based refrigerant cooling system. This system also has the benefit over the refrigerative system as the supply air, which is entirely fresh ambient air and is expected to improve the indoor environmental quality largely. Due to its fresh air handling capability combined with energy-saving potential, these systems can be considered for managing indoor conditions in hot and humid climates.

Author Contributions: R.N. initiated the project and developed the initial TRNSYS model. S.S. provided data from CSIRO for validation, general inputs and comments for the manuscript. R.P. developed the TRNSYS model and provided inputs and analysis based on the model outcomes. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

CC	Cooling coil
CHW	Chilled water coil
DW	Desiccant wheel
EXH AIR	Exhaust air
ERW	Energy recovery wheel
RC	Regeneration coil
SHRW	Sensible heat recovery wheel
COP	Coefficient of Performance
COP_{EL}	Electrical Coefficient of Performance
COP_{TH}	Thermal Coefficient of Performance
L	Load
HR	Thermal energy required for regeneration
Δp	Total Pressure increase (Pa, N/m^2)
P_F	Supply & exhaust fan power (W)
q	Air volume flow (m^3/s)
η_F	Fan efficiency
P_P	Pump power
Q	Volume flow rate (m^3/h)
η_P	Pump efficiency
η_m	Motor efficiency
EC	Electrical power consumed
P_{SA}	Power consumed in the supply air system (W)
P_{EA}	Power consumed in the exhaust air system (W)
P_{SHRW}	Power consumed by the sensible heat recovery wheel (W)
P_{pumps}	Total power consumed by the pumps

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