







## Article

# Dynamic Evaluation of Desiccant Dehumidification Evaporative Cooling Options for Greenhouse Air-Conditioning Application in Multan (Pakistan)

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**Citation:** Ashraf, H.; Sultan, M.; Shamshiri, R.R.; Abbas, F.; Farooq, M.; Sajjad, U.; Md-Tahir, H.; Mahmood, M.H.; Ahmad, F.; Taseer, Y.R.; et al. Dynamic Evaluation of Desiccant Dehumidification Evaporative Cooling Options for Greenhouse Air-Conditioning Application in Multan (Pakistan). *Energies* **2021**, *14*, 1097. <https://doi.org/10.3390/en14041097>

Academic Editors: Jan Danielewicz and Krzysztof Rajski  
Received: 27 December 2020  
Accepted: 14 February 2021  
Published: 19 February 2021

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**Abstract:** This study provides insights into the feasibility of a desiccant dehumidification-based Maisotsenko cycle evaporative cooling (M-DAC) system for greenhouse air-conditioning application. Conventional cooling techniques include direct evaporative cooling, refrigeration systems, and passive/active ventilation, which are commonly used in Pakistan; however, they are either not feasible due to their energy cost, or they cannot efficiently provide an optimum microclimate depending on the regions, the growing seasons, and the crop being cultivated. The M-DAC system was therefore proposed and evaluated as an alternative solution for air conditioning to achieve optimum levels of vapor pressure deficit (VPD) for greenhouse crop production. The objective of this study was to investigate the thermodynamic performance of the proposed system from the viewpoints of the temperature gradient, relative humidity level, VPD, and dehumidification gradient. Results showed that the standalone desiccant air-conditioning (DAC) system created maximum dehumidification gradient (i.e., 16.8 g/kg) and maximum temperature gradient (i.e., 8.4 °C) at 24.3 g/kg and 38.6 °C ambient air conditions, respectively. The DAC coupled with a heat exchanger (DAC+HX) created a temperature gradient nearly equal to ambient air conditions, which is not in the optimal range for greenhouse growing conditions. Analysis of the M-DAC system showed that a maximum air temperature gradient, i.e., 21.9 °C at 39.2 °C ambient air condition, can be achieved, and is considered optimal for most greenhouse crops. Results were validated with two microclimate models (OptDeg and Cft) by taking into account the optimality of VPD at different growth stages of tomato plants. This study suggests that the M-DAC system is a feasible method to be considered as an efficient solution for greenhouse air-conditioning under the climate conditions of Multan (Pakistan).

**Keywords:** desiccant dehumidification; evaporative cooling; Maisotsenko cycle; greenhouse air-conditioning; Pakistan

## 1. Introduction

Closed-field production of crops and vegetables by means of greenhouses with different covering materials depends on the efficiency of the climate control system to provide optimum growth conditions with a reasonable energy cost. The extensive solar radiation during the daytime causes a greenhouse effect, resulting in high air temperature and relative humidity that can significantly exceed the optimum conditions. This can lead to a significant loss of yield and fruit quality in the absence of a proper ventilation and dehumidification system. Additionally, the plants and vegetables are more prone to pest/fungus/disease attack in higher relative humidity conditions than normal, which may also affect the production level of the greenhouse. To reduce the impact of suboptimal microclimate conditions on production, passive and active ventilation, in addition to direct evaporative cooling systems by means of pad-and-fans or swamp coolers, are conventionally used by greenhouse growers in Pakistan. The problem with these solutions is that they do not allow control over the relative humidity level of the greenhouse environment, hence leaving the plants and vegetables exposed to pests, fungus, and disease attacks. Therefore, a more appropriate air-conditioning (AC) solution that is cost-efficient and can be implemented in large scale commercial production is required. In this regard, desiccant dehumidification-based indirect evaporative cooling air-conditioning systems are gaining research attention that needs to be evaluated for AC application in different closed-field plant production environments.

The focus of this study is the city of Multan, Pakistan, which lies in a warm desert climate where direct evaporative cooling methods are used in combination with passive and active ventilation methods inside agricultural greenhouses. For the purpose of this research, a desiccant dehumidification-based Maisotsenko cycle evaporative cooling (M-DAC) system is proposed and evaluated as an alternative solution for controlling humidity levels to achieve the optimum vapor pressure deficit (VPD). The objective was to provide a systematic thermodynamic analysis of the proposed M-DAC system from the viewpoints of temperature gradient, relative humidity levels, optimum VPD, and dehumidification gradient. A review of the relevant published literature is summarized in Section 2 to identify the most recent achievements and the gaps. Modeling of the M-DAC and dynamic evaluation by means of the optimality degree and comfort ratio models are described in Section 3. The proposed solution is validated and discussed in Section 4 based on the thermodynamic analysis and the microclimate models. Section 5 concludes the study and highlights the potential aspects for further research and development.

## 2. Background and Literature Review

Evaporative cooling (EC) systems can potentially be classified into direct evaporative (DEC), indirect evaporative (IEC), and Maisotsenko cycle evaporative (MEC) systems. The MEC system is widely studied by the research community for heating, ventilation, and air-conditioning (HVAC) system applications. Pandelidis and Anisimov [1] numerically studied and compared eight different configurations of a standalone Maisotsenko-cycle evaporative heat and mass exchanger. Caliskan et al. [2] studied energy and exergy analyses of the standalone MEC system for building air-conditioning. Zhan et al. [3] studied counter flow standalone MEC system for building air-conditioning application. Caliskan et al. [4] compared traditional vapor compression air-conditioning systems with the MEC system based on energy and exergy analyses. Chua et al. [5] reviewed environment-friendly air-conditioning options for building applications. Pandelidis et al. [6] numerically simulated the MEC heat exchanger for different air-conditioning applications. Rogdakis et al. [7] numerically simulated and experimentally validated the MEC system for building air-conditioning application for the climatic conditions of Greece. Cui et al. [8] studied the MEC system for precooling of ambient air as an energy-saving technique for hot and humid climatic conditions. Riangvilaikul and Kumar [9] numerically investigated the MEC

system for different inlet conditions. Sultan [10] studied the MEC system for greenhouse air-conditioning and ventilation applications. Anisimov et al. [11] theoretically and experimentally analyzed the standalone MEC system for air-conditioning and found the MEC system to be an efficient way of indirect evaporative cooling. In addition, results from the study concluded that inlet air velocity impacted the performance of the system. Zhao et al. [12] numerically investigated the performance of the MEC system for climatic conditions of the United Kingdom, which resulted in wet-bulb effectiveness of 1.3. Zube and Gillan [13] experimentally studied a commercial type MEC air-conditioning system and evaluated heat mass transfer parameters inside a heat exchanger for the first time. Weerts [14] reported the power-saving potential of commercially available standalone MEC system compared to the vapor compression air-conditioning (VCAC) system. Maisotsenko and Treyger [15] also reported the energy-saving potential of the standalone MEC system compared to standalone VCAC system. Similarly, [16–18] also studied the standalone MEC system in-depth and concluded that the MEC system can potentially be an environmentally friendly air-conditioning option compared to the VCAC system.

The standalone desiccant air-conditioning (DAC) system has been extensively studied in the literature for various applications. Generally, the DAC system better suits regions with higher relative humidity in the ambient air. The DAC system was studied for the possible application of vehicular air-conditioning [19]. Additionally, the authors used a continuous input and output modulator (i.e., proportional-derivative (PD) controller) to increase the efficiency of the system and concluded that the DAC system coupled with the PD controller was energy conservative compared to conventional AC systems in vehicles [19]. Moreover, the DAC system was successfully installed in a wet market in Hong Kong. The authors numerically simulated the energy loads of a hypothetical wet-market using EnergyPlus for the climatic conditions of Hong Kong and compared results with the actual wet market [20]. The authors concluded that the DAC could potentially be used as a replacement for conventional air-conditioning systems in the wet markets. A solar-assisted open cycle DAC system was experimentally tested for grain storage in Melbourne, Australia [21]. The authors concluded that a 5.85 m<sup>3</sup> solar collector area was sufficient for cooling of up to 200 tonnes of grain. The coefficient of performance (COP) was a function of surrounding conditions and air mass flow rate. A maximum COP of 86.2 was observed under humid conditions for an air mass flow rate of 0.019 kg/s [21]. The standalone DAC system was studied for a greenhouse air temperature and humidity control system [22]. Moreover, the authors studied the greenhouse air-conditioning from a vapor pressure deficit (VPD) point of view. The standalone DAC system with dual desiccant wheels was studied for residual waste heat recovery in marine ships [23]. The authors concluded that the DAC system was 33.4% more power-efficient compared to traditional air-conditioning systems for marine ships. An EC-coupled DAC system was studied in two modes (i.e., ventilation and recirculation modes) [24]. The authors concluded that conventional ventilation mode is more efficient when humidity ratio is below ~10.9 g/kg, whereas the recirculation mode of the air-conditioning system is more efficient for areas where humidity ratio is above ~10.9 g/kg, consuming a higher amount of input energy. Enteria et al. [25] studied the analyses of the first and second law of thermodynamics for a solar thermal electric desiccant air-conditioning system [25]. A simulated solar-operated evaporative cooling-assisted solid desiccant air-conditioning system was developed in a tropical region of Malaysia [26]. The authors used the TRNSYS simulation environment to simulate the desiccant air-conditioning system for climatic conditions of Malaysia. The authors concluded that two-stage ventilation mode desiccant air-conditioning system is better for tropical conditions such as Malaysia, with a temperature gradient of 17.6 °C at 30 °C ambient temperature [26]. Another study investigated a dual-wheel cooling system [27]. Results from the study indicate that higher regeneration temperature (RT) produces higher total COP at both air mass flow rates (i.e., 100 and 200 m<sup>3</sup>/h) [27]. Moreover, the performance and energy-saving potential of a two-stage solar-driven rotary wheel DAC system was compared with a conventional vapor compression air-conditioning (VCAC) system

for building air-conditioning [28]. Building energy performance simulation (BEPS) was conducted for Berlin and Shanghai. The DAC system was found to be energy-saving and more efficient for both of the cities compared to VCAC system. However, in the case of Shanghai, the RT of the DAC system was 30 °C higher than that of Berlin [28]. Figure 1 shows the schematic diagram and working principle of a conventional vapor compression air-conditioning system, and solar thermal liquid desiccant dehumidification-based air-conditioning system. Moreover, the authors studied desiccant dehumidification systems and evaporative cooling systems in detail for various applications, including greenhouse air-conditioning [29–32], building air-conditioning [33], livestock air-conditioning [34], and agricultural product storage [35–44].

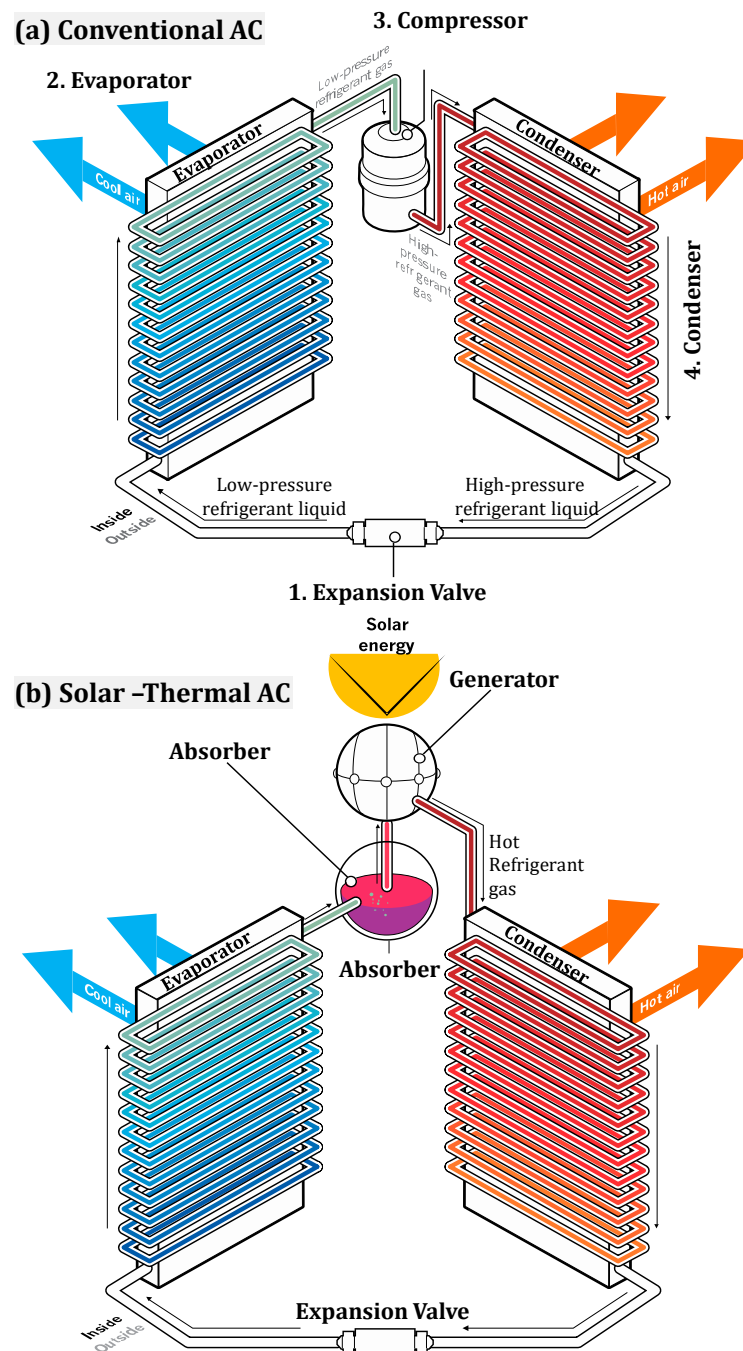
Song and Sobhani [45] studied solar-assisted desiccant air-conditioning system coupled with a Maisotsenko evaporative cooling system and phase change material for building air-conditioning application. The authors concluded that the maximum total COP of 0.404 was achieved using a solar-operated desiccant air-conditioning system coupled with a Maisotsenko evaporative cooler using a phase change material for air-conditioning of an official building in Iran. In another study, a Maisotsenko cycle-assisted desiccant air-conditioning system was studied for the climatic conditions of Japan [46]. Pandelidis et al. [47] numerically studied the Maisotsenko cycle-assisted desiccant air-conditioning system with two modifications, i.e., bypass and bypass with rotary sensible heat exchanger. The authors concluded that all the configurations successfully managed the sensible load of the environment while being highly dependent on the humidity ratio of ambient air. The addition of a rotary sensible heat exchanger in system configuration 3 increased the cooling capacity up to ~2.35 kW at 40 °C ambient temperature generating ~17.25 °C supply air temperature at the same ambient conditions [47]. Table 1 shows the summary of different desiccant-based Maisotsenko cycle evaporative cooling systems for air-conditioning applications. From Table 1 and the reviewed literature, it is evident that there is a gap in the literature regarding OptDeg and Cft plant thermal comfort indices for the proposed M-DAC system. Moreover, this study aims to use a low and easily achievable regeneration temperature (i.e., 50 °C) coupled with different comfort levels of plant growth stages, which is lacking in previous studies.

**Table 1.** Summary of different desiccant-based Maisotsenko cycle evaporative cooling systems for air-conditioning applications.

System	Findings	Regeneration Temperature	Ref.
Desiccant based indirect evaporative cooling	For 70 °C $T_{\text{regen}}$ , supply flow ratio 0.67, indirect EC flow ratio 0.3 results in maximum performance i.e., COP higher than 20	70 °C	[48]
Maisotsenko cycle desiccant evaporative cooling	For outside conditions of 25 °C, lowest temperature gradient was 14.9 °C	70 °C	[1]
Standalone MEC	For Greek climate conditions, MEC system achieved maximum temperature gradient of 20.4 °C in Athens	–	[7]
Maisotsenko cycle desiccant evaporative cooling	Different desiccant materials studied for greenhouse air-conditioning, only limitation was integration of OptDeg and Cft models for different growth stages of crops	Different regeneration temperatures	[10]
Standalone DAC system	Polymer-based sorbents PS-I and PS-II are studied at regeneration temperatures 50 and 80 °C for greenhouse air-conditioning	Different regeneration temperatures	[30]
Standalone DAC system	Activated carbon powder (ACP) and activated carbon fiber (ACF) were studied for greenhouse air-conditioning at regeneration temperatures 41 to 75 °C, only limitation was the detailed analysis of crop growth and plant comfort indices	Different regeneration temperatures	[49]

The study area (i.e., Multan) lies in a warm desert climate of the Köppen–Geiger climatic classification (Figure 2). Therefore, this justifies the need for an air-conditioning system for greenhouses. Usually, pad-type direct evaporative cooling or natural/forced

ventilation systems are used for greenhouse air-conditioning. These systems are not feasible throughout the year and fail to achieve the required optimum temperature and relative humidity conditions inside a greenhouse. Moreover, the DEC system fails to handle humidity inside the greenhouse environment, which is crucial to the vapor pressure deficit. The desiccant-based hybrid evaporative cooling systems have not been explored for greenhouse air-conditioning applications in Pakistan.



**Figure 1.** Schematic diagram and working principle of (a) conventional vapor compression air-conditioning system, and (b) solar thermal liquid desiccant dehumidification-based air-conditioning system, reproduced from [50].

Figure 3 shows the ambient climatic conditions of Multan (Pakistan) and the greenhouse optimum temperature and relative humidity zone. Figure 3 was made using Lady-

bug Rhino v6 for the climatic conditions of Multan (Pakistan). A typical meteorological EnergyPlus weather file was used in Ladybug Rhino to represent the hourly weather conditions. In Figure 3, the light fill area shows the average hourly variation, whereas the lines show the average daily variation in climatic parameters. From the viewpoint of the literature reviewed above, pad-type direct evaporative cooling systems are only suitable for controlling the temperature, and fail to achieve the optimum humidity level inside the greenhouse. To overcome this issue, an indirect evaporative cooling system could be used, which provides a sensibly cooled product air with no latent transfer from the wet to dry channels of the IEC system. However, due to the low wet bulb effectiveness of the system, it cannot be considered a suitable option for achieving the required temperature level inside the greenhouse. Additionally, higher levels of humidity inside the greenhouse can potentially result in pest, fungus, and disease attacks on the plants and vegetables. Therefore, in this study, a desiccant dehumidification-based Maisotsenko cycle evaporative cooling air-conditioning (M-DAC) system was thermodynamically analyzed for the climatic conditions of Multan (Pakistan). The performance of the system was analyzed for optimum greenhouse air-conditioning conditions from the viewpoints of temperature gradient, relative humidity level, and wet bulb effectiveness of the systems.

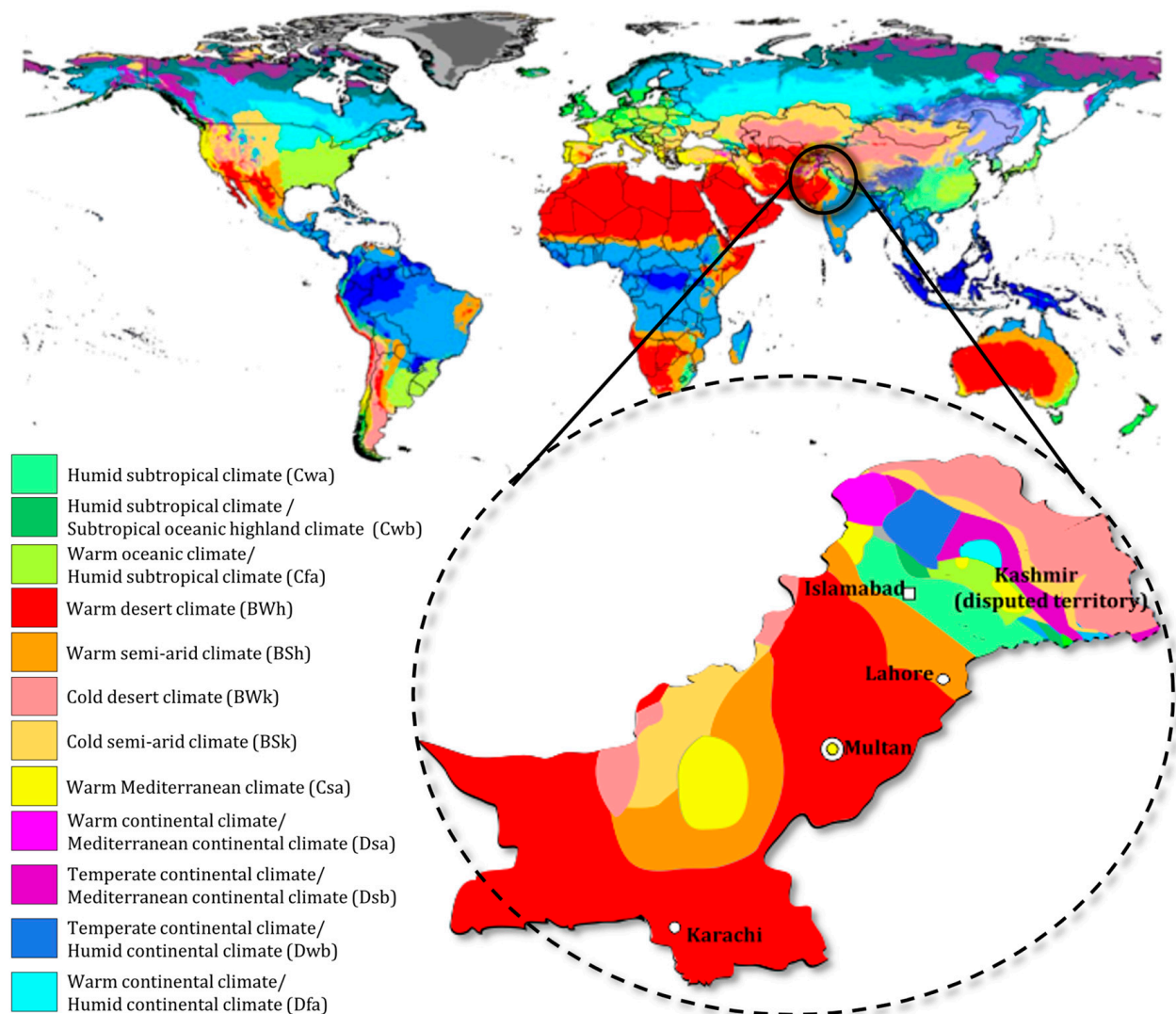


Figure 2. Köppen–Geiger climate classification of Pakistan, reproduced from [51].

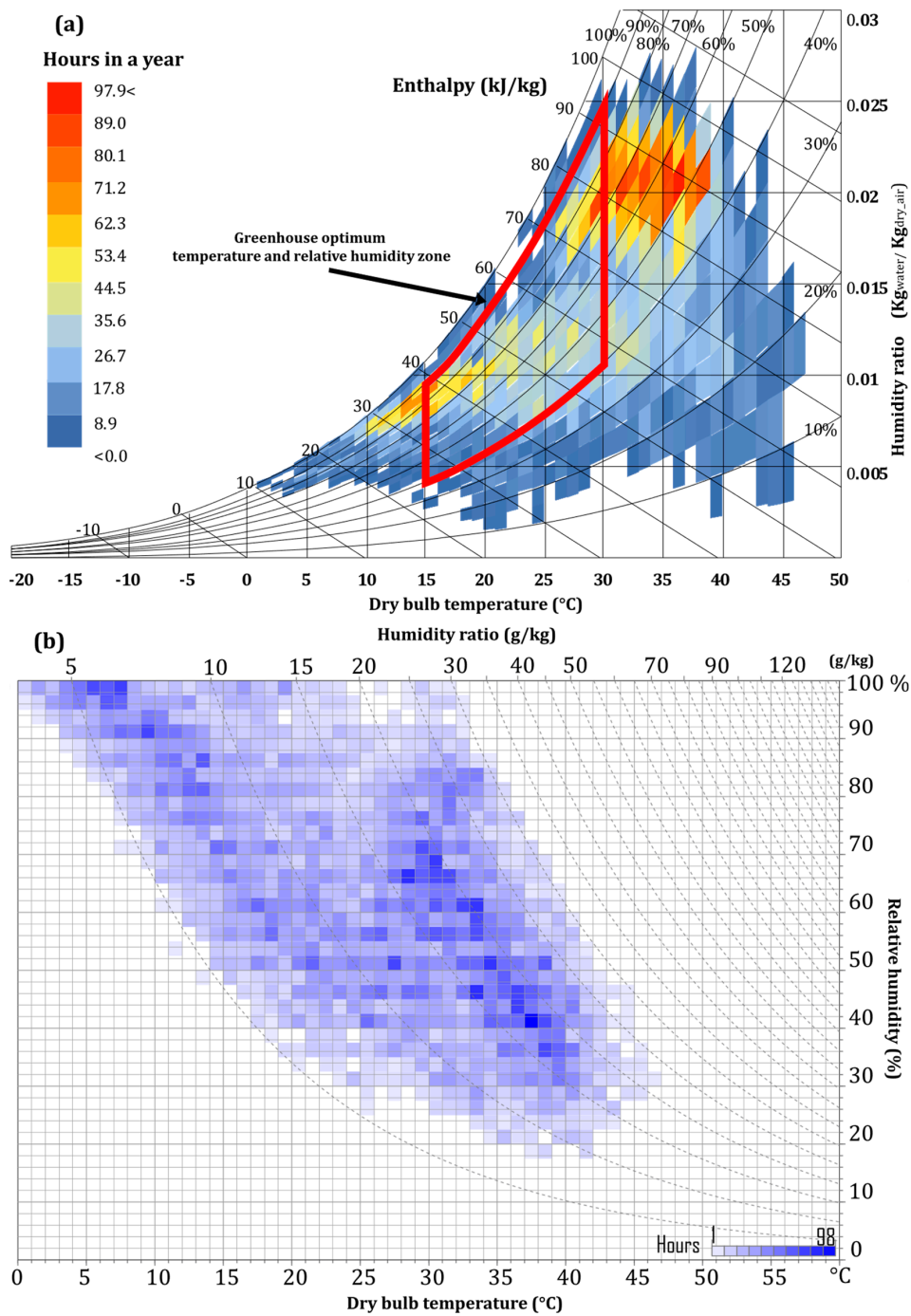


Figure 3. Ambient climatic conditions of Multan (Pakistan) on (a) psychrometric chart, and (b) relative humidity chart.

### 3. Materials and Methods

#### 3.1. Modelling M-DAC System

A general working principle of the M-DAC system is first shown in Figure 4, followed by a schematic representation in Figure 5 that demonstrates the application of the proposed system for greenhouse air-conditioning for a possible higher yield. Taking into account the climatic conditions of Multan (Pakistan), the performance of a silica gel-based desiccant

wheel for M-DAC system was investigated using Equations (1)–(4), as described in the literature [52,53].

$$F_{1,ip} = \frac{A_1}{(T_{ip} + 273.15)^{1.49}} + B_1 \left( \frac{w_{ip}}{1000} \right)^{C_1} \tag{1}$$

$$F_{2,ip} = \frac{(T_{ip} + 273.15)^{1.49}}{A_2} - B_2 \left( \frac{w_{ip}}{1000} \right)^{C_2} \tag{2}$$

$$\eta F_1 = \frac{F_{1,2} - F_{1,1}}{F_{1,8} - F_{1,1}} \tag{3}$$

$$\eta F_2 = \frac{F_{2,2} - F_{2,1}}{F_{2,8} - F_{2,1}} \tag{4}$$

where  $F_{1,ip}$  and  $F_{2,ip}$  represent the combined potential as a function of the humidity ratio in kg/kg and temperature in °C of the solid desiccant-based air-conditioning system.  $\eta F_1$  and  $\eta F_2$  represent the efficiencies of the system correlating to the combined potentials.  $\eta F_1$  and  $\eta F_2$  have typical values of 0.05 and 0.95 for a high-efficiency silica gel-based solid desiccant wheel air-conditioning system. The coefficients A, B, and C are presented in Table 2.

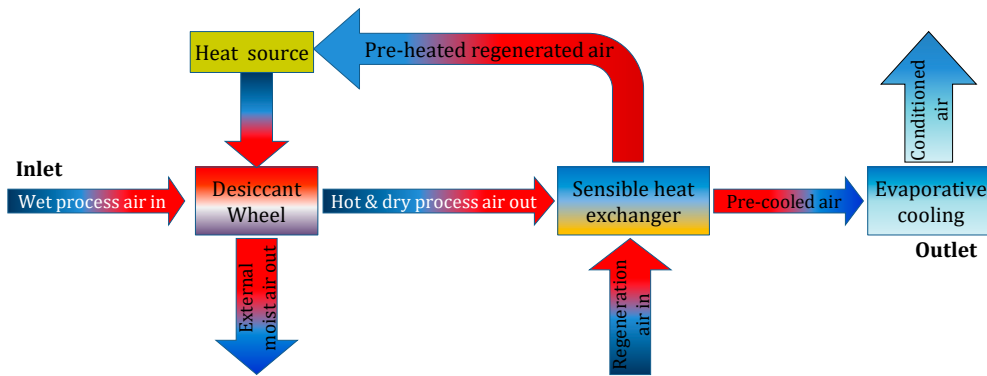


Figure 4. Working principle of a desiccant dehumidification-based evaporative cooling air-conditioning system.

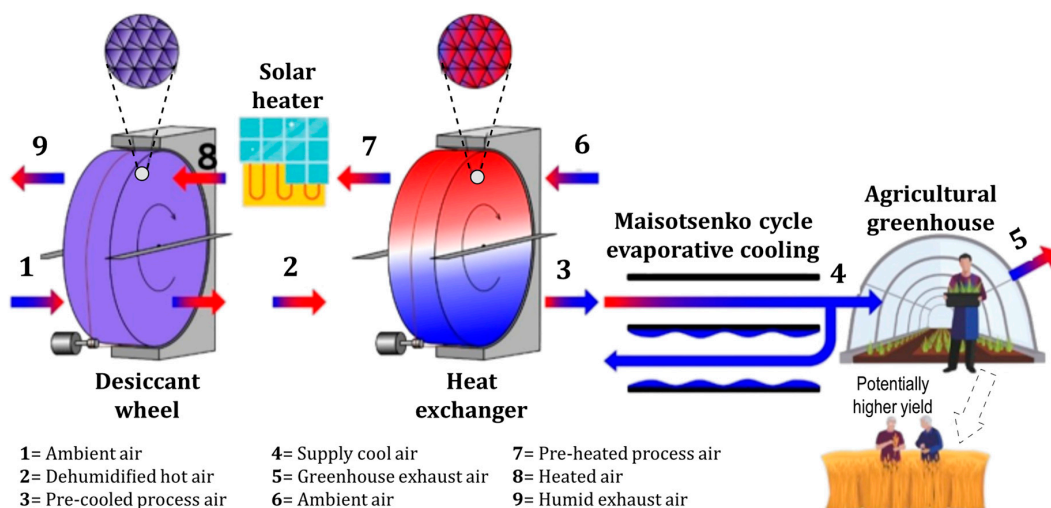


Figure 5. Schematic diagram and working principle of the proposed desiccant dehumidification Maisotsenko cycle evaporative cooling system.

In comparison with the conventional vapor compression air-conditioning systems, the desiccant dehumidification-assisted evaporative cooling air-conditioning system consumes less energy due to the absence of compressor work and is able to operate at lower regenera-



tion temperatures (i.e., 50–90 °C), which are easily attainable using solar thermal, biogas, and waste heat sources. The Maisotsenko cycle evaporative cooling system was modeled using the model of [32]. Equation (5) shows the performance model of the Maisotsenko cycle evaporative cooling system.

**Table 2.** Coefficients used in the Jurinak model (Equations (1)–(4)).

Coefficient	Value
A <sub>1</sub>	−2865
B <sub>1</sub>	4.344
C <sub>1</sub>	0.8624
A <sub>2</sub>	6360
B <sub>2</sub>	1.127
C <sub>2</sub>	0.07969

$$T_{out} = 6.70 + 0.2630(T_{in}) + 0.5298(w_{in}) \quad (5)$$

where  $T_{in}$  represents the inlet temperature of the process air in °C and  $w_{in}$  represents the inlet humidity ratio of the process air in kg/kg. Equation (5) models the temperature of product air produced from the Maisotsenko cycle evaporative cooling system. The model was developed by [32] using experimental data of a developed Maisotsenko cycle based evaporative cooling system [11].

The sensible heat exchanger (HX) used in this study was modeled using the heat exchanger standard equation (Equation (6)) provided in the literature by The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) [54].

$$T_3 = T_2 - \varepsilon_{HX}(T_2 - T_1) \quad (6)$$

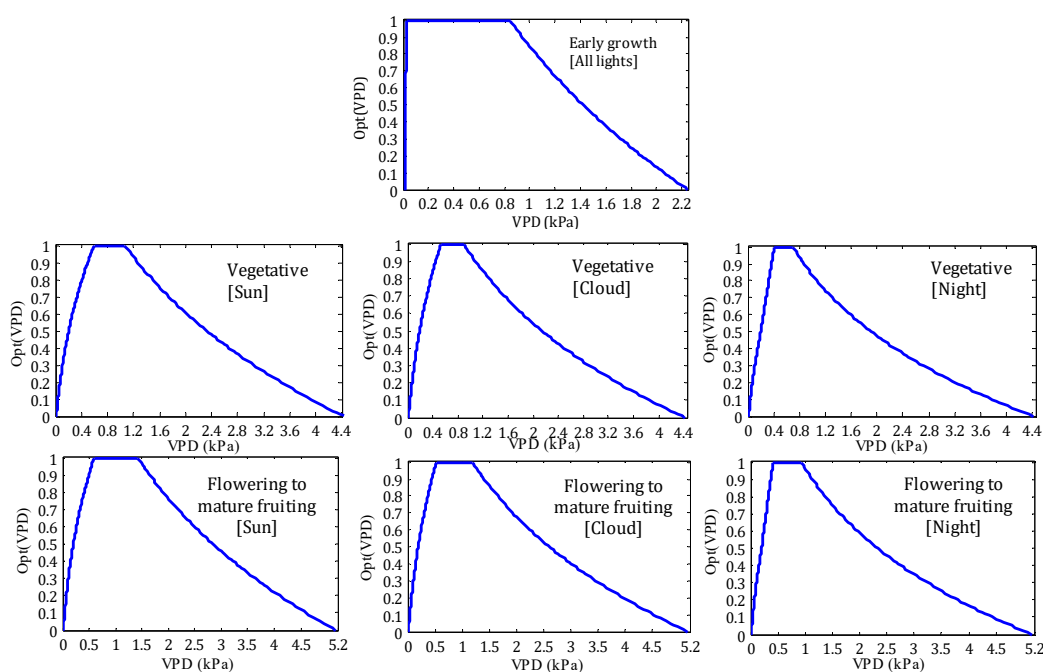
where  $T_3$  represents the outlet temperature of the sensibly cooled process air resulting from the heat exchanger,  $T_2$  represents the outlet temperature of heated and dehumidified air resulted from the desiccant air-conditioning system,  $\varepsilon_{HX}$  represents the sensible heat exchange efficiency of the heat exchanger (assumed as 0.9), and  $T_1$  represents the dry bulb temperature of the ambient air inlet into the desiccant air-conditioning system. Figure 5 shows the working principle and schematic of the sensible heat exchanger. Regeneration of the desiccant is done usually through regeneration air at higher than ambient temperature. To heat the regeneration air, the air passes through the heat exchanger which sensibly heats the air to a certain limit. For further heating of the regeneration air before entering the desiccant, a heating unit/source is used. This heating source could be a heating electric coil, solar heater, or biogas-operated thermal heat source. Waste heat from a condenser or any other source in industrial uses can also be used to regenerate the desiccant material and absorb its moisture. For this study, a regeneration temperature of 50 °C was used. This relatively lower temperature is achievable through a solar thermal system, biogas-operated thermal heat source, or waste heat. A flat plate solar collector and heat exchanger were used for this study to provide the required regeneration temperature (i.e., 50 °C). Figure 5 shows the working principle and schematic of a solar thermal heating source.

### 3.2. Optimality Degree and Comfort Ratio Model

Dynamic assessment of the proposed M-DAC system for greenhouse crop production application was carried out by means of two microclimate models, namely, OptDeg [55] and Cft-ratio [56] that take into account VPD measurements as inputs. The output of the first model, optimality degree of VPD, denoted by  $Opt(VPD)_{GS, (Light)} = \alpha : VPD \rightarrow [0, 1]$  is a quantitative value between 0 and 1 that represents how close a VPD reading inside the greenhouse is to the optimum value ( $0 \leq \alpha \leq 1$ ) at a specific growth stage (GS) and light condition. Graphical representation of the membership functions for defining  $Opt(VPD)$  under different conditions is provided in Figure 6. In this model, a VPD measurement

in the greenhouse is first calculated from Equation (7) at the time  $t_{m,n}$  (where  $m$  and  $n$  refer to a specific minute and date of the growing stage), and is then mapped to a value between 0 and 1 that quantifies the optimality degree of the microclimate for greenhouse production. In other words, an optimality-degree equal to 1 refers to a high potential yield with marketable value and high-quality fruit.

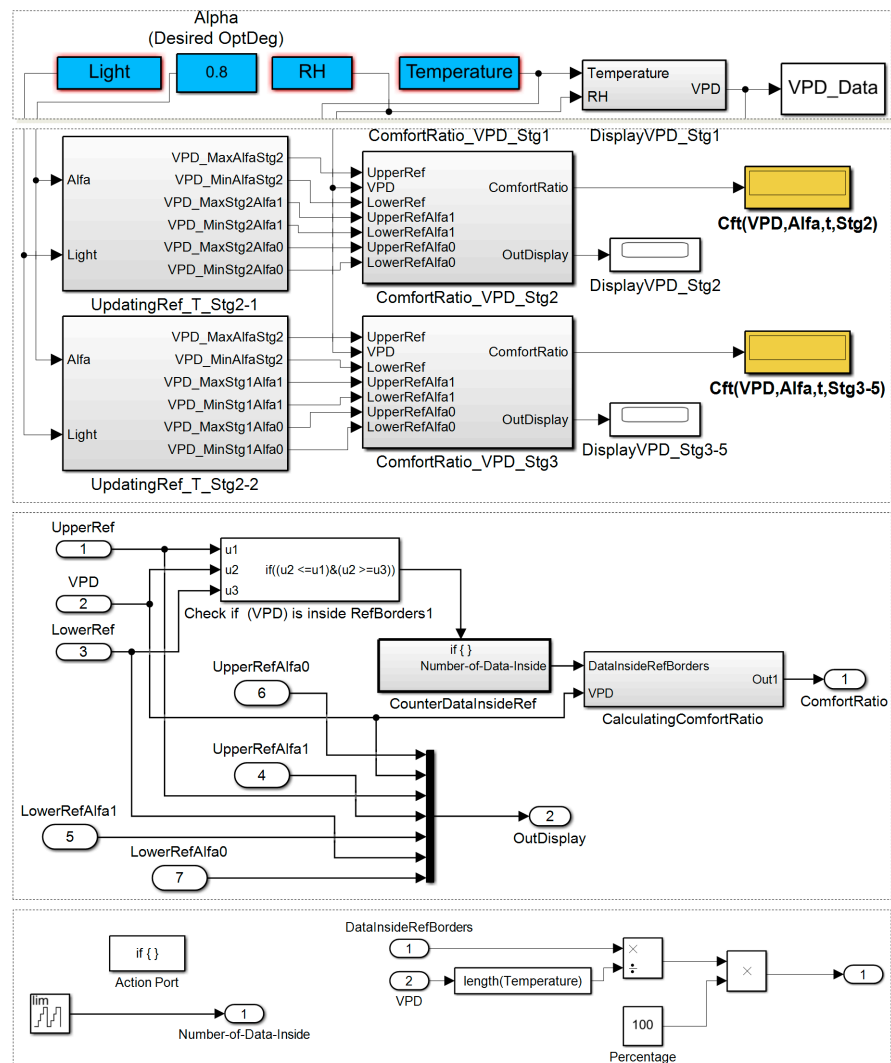
$$\begin{aligned}
 VPD_{t_{m,n}} &= f(T_{t_{m,n}}, RH_{t_{m,n}}, \tau_{t_{m,n}}) \\
 &= \frac{C_g 10^{(7.5\tau_{t_{m,n}})/(237.3+\tau_{t_{m,n}})}}{1000} \\
 &= \frac{C_g 10^{(7.5T_{t_{m,n}})/(237.3+T_{t_{m,n}})} RH_{t_{m,n}}}{1000 \cdot 100}
 \end{aligned} \tag{7}$$



**Figure 6.** Membership functions for defining optimality degrees of vapor pressure deficit at different light condition and growth stages, reproduced from [40].

It should be noted that there are various functions for calculating VPD that take into accounts different assumptions and constants; however, that presented in Equation (7) is widely accepted and used by meteorologists and commercial greenhouse growers [34]. The first part of Equation (7) represents the vapor pressure of the saturated environment (i.e., the leaf of the plant) in kPa,  $C_g$  denotes a coefficient that is equal to 610.7, and  $\tau$  represents the surface temperature of the leaf, which is usually measured by means of an infrared gun for such experiments. For the sake of simplicity, this parameter was assumed to be equal to the temperature of the air inside the greenhouse at both ambient and evaporative assisted desiccant air-conditioning system supply air temperature. In fact, assuming leaf temperature equal to the surrounding air temperature is normal practice of greenhouse growers. The second part of Equation (7) represents the vapor pressure of the air inside the greenhouse in kPa,  $T$  represents the temperature of the air inside the greenhouse in  $^{\circ}\text{C}$ , and  $RH$  represents the relative humidity of the air inside the greenhouse in (%). Because VPD is the function of temperature of the leaf surface, air temperature, and relative humidity, any slight variation in one of the parameters from the optimal values results in restricted growth of the plants, and can have significant consequences on reducing the overall yield and quality.

The OptDeg model determines the variation of a single VPD measurement from the optimal references in an instant time. Although this model can provide an overview of the performance of the M-DAC system in sample time, a different model called comfort ratio, denoted by Cft-ratio, that incorporates a sample time frame, such as 24 h was used. The comfort ratio of VPD, denoted by  $Cft(VPD, t, \alpha_s)_{GS} = \beta$ , represents the percent of vapor pressure deficit at a specific time  $t$  and growth stage, which lies between the reference conditions of vapor pressure deficit related to  $\alpha_s$ . Ideal microclimate conditions inside a greenhouse could potentially be defined as  $Cft(VPD, t, 1) = 1$ . Self-adjusted optimality degree is denoted by  $\alpha_s$  to define the reference conditions required for performance evaluation of the proposed M-DAC system or control for microclimate evaluation. The set of Simulink blocks that were implemented to determine the Cft-ratio of the VPD values for different  $\alpha_s$  generated from the M-DAC system are shown in Figure 7. For this research, we selected three levels of  $\alpha_s$  as 0, 0.5, and 1, representing failure, marginal, and ideal production. It is worth mentioning that OptDeg and Cft-ratio models were derived with a prime focus on improved fruit quality and yield. The derivations at the back of the models (which can be found in published literature) for greenhouse tomato production were condensed for the sake of simplicity in this study.

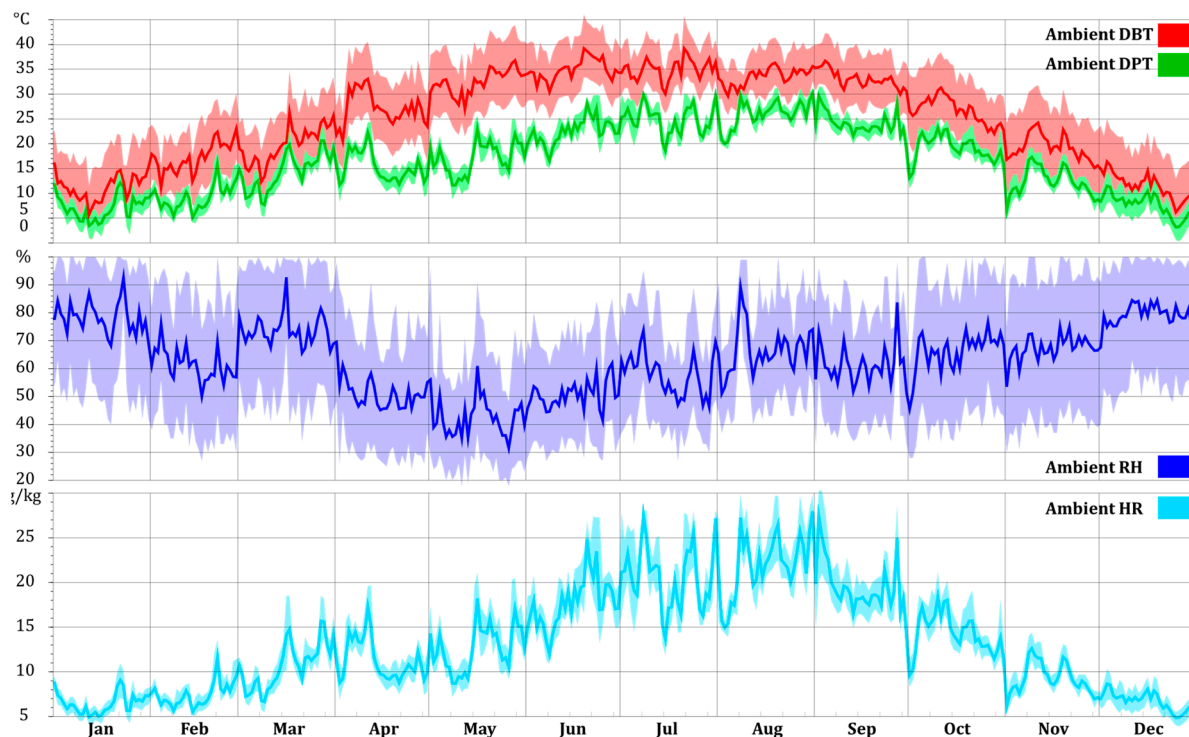


**Figure 7.** Implementation of the Cft-ratio model using Simulink blocks for validating the performance of the desiccant dehumidification Maisotsenko cycle evaporative cooling (M-DAC) system based on comfort ratios of vapor pressure deficit (VPD) for greenhouse production, reproduced from [56].

## 4. Results and Discussion

### 4.1. Validation Based on Thermodynamic Analysis

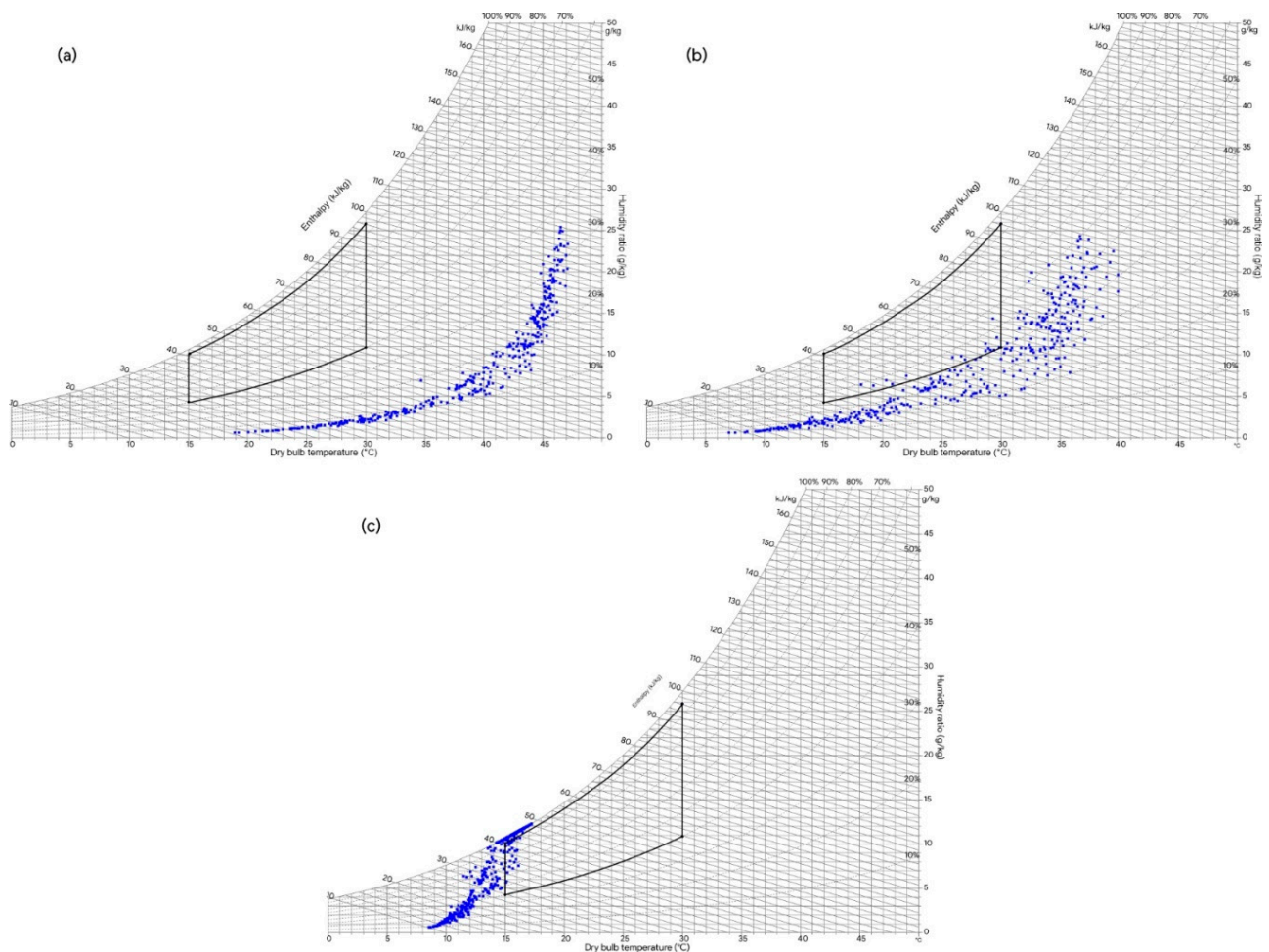
Figure 8 shows the annual profile of the climatic ambient air conditions of Multan (Pakistan). The daily, hourly, and monthly variation of dry-bulb temperature shows that cooling inside the greenhouse environment is mostly required in summer months (i.e., May to August). However, the applicability of standalone pad-type direct evaporative cooling systems is limited during the monsoon (heavy rainfall season, i.e., July to August, shown in Figure 8, humidity ratio (HR)) which results in excess relative humidity in the air. Consequently, a desiccant dehumidification-based Maisotsenko cycle evaporative cooling system was thermodynamically analyzed for the climatic conditions of Multan. Accordingly, results from the study are presented in Figures 8–13, which show the annual representation of the performance of the proposed system and the profile of the vapor pressure deficit inside the greenhouse.



**Figure 8.** Profile of annual climatic ambient air conditions of Multan (Pakistan).

The standalone DAC system created a maximum temperature gradient, i.e.,  $8.4\text{ }^{\circ}\text{C}$  more at ambient conditions of  $38.6\text{ }^{\circ}\text{C}$  with a humidity ratio of  $21.6\text{ g/kg}$  at ambient conditions of  $25\text{ g/kg}$ , which lies well outside the required greenhouse optimum temperature and humidity conditions (Figure 9a). It is worth mentioning that the temperature gradient is defined as the difference between the dry bulb temperature of the ambient air and the dry bulb temperature of the product air at the system outlet node. The studied standalone DAC system created unsuitable thermal and humidity conditions inside the greenhouse throughout the year due to a relatively higher temperature and very dehumidified air (Figure 9a). According to Figure 9b, the standalone DAC system coupled with a sensible heat exchanger (DAC+HX) (working efficiency assumed to be 0.9) created a maximum temperature gradient, i.e.,  $0.77\text{ }^{\circ}\text{C}$  more at ambient conditions of  $39.2\text{ }^{\circ}\text{C}$  with a humidity ratio of  $16.8\text{ g/kg}$  at ambient conditions of  $24.3\text{ g/kg}$ , which lies slightly outside the required greenhouse optimum temperature and humidity conditions (Figure 9b), and is very close to ambient air conditions. The studied DAC+HX system created very close thermal and humidity conditions to ambient conditions inside the greenhouse throughout the year.

Figure 9c shows the psychrometric profile of temperature and relative humidity of the desiccant dehumidification-based Maisotsenko cycle evaporative cooling (M-DAC) system for the climatic conditions of Multan (Pakistan). According to Figure 9c, most of the hourly points of the M-DAC system lie inside the required thermal and humidity conditions of the greenhouse environment. The M-DAC system created a maximum temperature gradient i.e., 21.98 °C. According to Figure 9c, the M-DAC system coupled with a sensible heat exchanger (working efficiency assumed to be 0.9) created a maximum temperature gradient i.e., 21.9 °C more at ambient conditions of 39.2 °C with a humidity ratio of 16.8 g/kg at ambient conditions of 19.6 g/kg, which lies well inside the required thermal and humidity conditions inside the greenhouse environment (Figure 9c).



**Figure 9.** Profile of annual temperature and relative humidity of (a) standalone desiccant air-conditioning (DAC) system, (b) desiccant air-conditioning coupled with sensible heat exchanger (DAC+HX) system, and (c) desiccant dehumidification Maisotsenko cycle evaporative cooling (M-DAC) system, for greenhouse air-conditioning.

Figure 10 shows the vapor pressure deficit (VPD) profile and growth stages of plants inside a greenhouse (for the temperature of the leaf equal to the temperature of the air inside the greenhouse environment). According to Figure 10, the VPD of  $<0.4 > 1.6$  kPa is considered an extremely dangerous zone for over/under transpiration. The VPD of 0.4–0.8 kPa is considered a slightly low transpiration zone (propagation/early vegetative growth of the plants). The VPD of 0.8–1.2 kPa is considered a healthy transpiration zone (late vegetation/early flowering stage of the plants). The VPD of 1.2–1.6 kPa is considered a high transpiration zone (mid/late flowering stage of the plants).

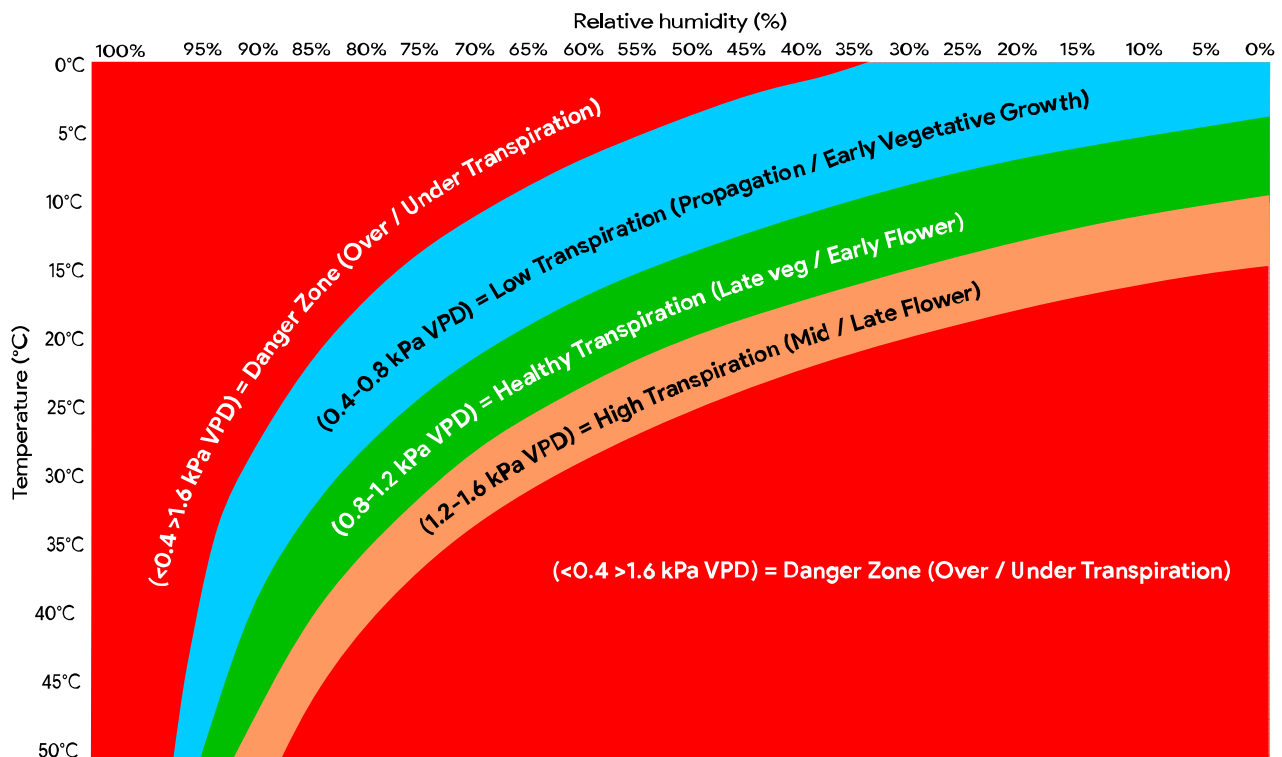
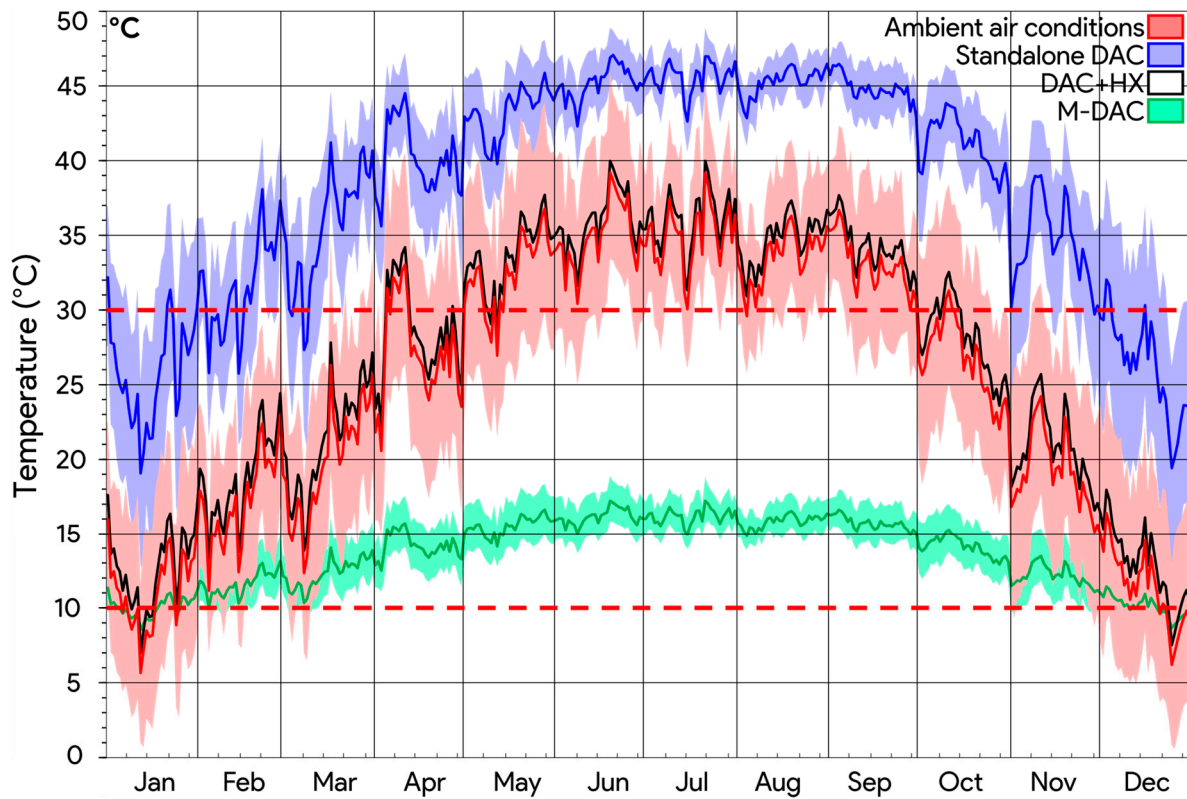


Figure 10. Vapor pressure deficit (VPD) profile and growth stages of plants inside a greenhouse.

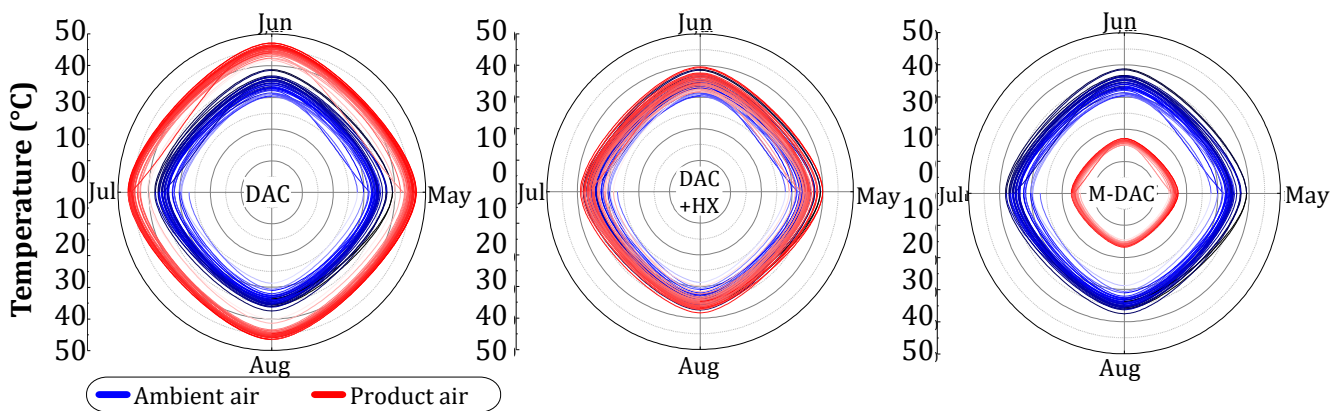
Figure 11 shows the temperature profile of the studied systems against the ambient air conditions of Multan (Pakistan) for greenhouse air-conditioning application. The ambient air conditions lie outside the required temperature conditions (marked by the dotted red line, Figure 11) from April to September (i.e., typical summer months) of Multan. Air-conditioning is required inside the greenhouse environment during the summer months. Only M-DAC system created a uniform temperature gradient which lies well inside the required temperature conditions (marked in dotted red line, Figure 11) of the greenhouse environment. Figure 12 shows the temperature gradient of the DAC, DAC+HX, and M-DAC system in the summer months (i.e., May to August) for the greenhouse air-conditioning for the climatic conditions of Multan (Pakistan). The DAC system created higher than ambient temperature conditions throughout the summer months, which is unsuitable for greenhouse air-conditioning requirements (Figure 12). The DAC + HX system created slightly higher/very close to ambient temperature conditions throughout the summer months, which is also not suitable for greenhouse air-conditioning requirements (Figure 12). The M-DAC system created a temperature gradient which is highly suitable for the required temperature conditions throughout the summer months for greenhouse air-conditioning application (Figure 12).

Figure 13 shows the dehumidification profile of the desiccant dehumidification unit of the proposed system for the summer months (i.e., May to August) for the climatic conditions of Multan (Pakistan). The desiccant dehumidification unit created a maximum dehumidification gradient of the ambient air, i.e., 6.61 g/kg at ambient conditions of 14.14 g/kg. It is worth mentioning that the dehumidification gradient is defined as the difference between the ambient air humidity ratio and the humidity ratio of the product air exiting the outlet of the proposed system. Compared to current proposed passive cooling techniques for greenhouse air-conditioning (i.e., pad-and-fan type systems for greenhouse air-conditioning systems), the proposed M-DAC system has the ability to manipulate the moisture inside the greenhouse environment, which is a key factor in defining the vapor pressure deficit inside the greenhouse. In turn, this defines the growth rate and ultimately results in better yield and fruit quality. Moreover, the M-DAC system can easily achieve

the required temperature conditions inside a greenhouse compared to passive cooling techniques, i.e., pad-and-fan type AC systems.



**Figure 11.** Annual profile of thermodynamic performance of the proposed M-DAC system for greenhouse air-conditioning for the climatic conditions of Multan (Pakistan).

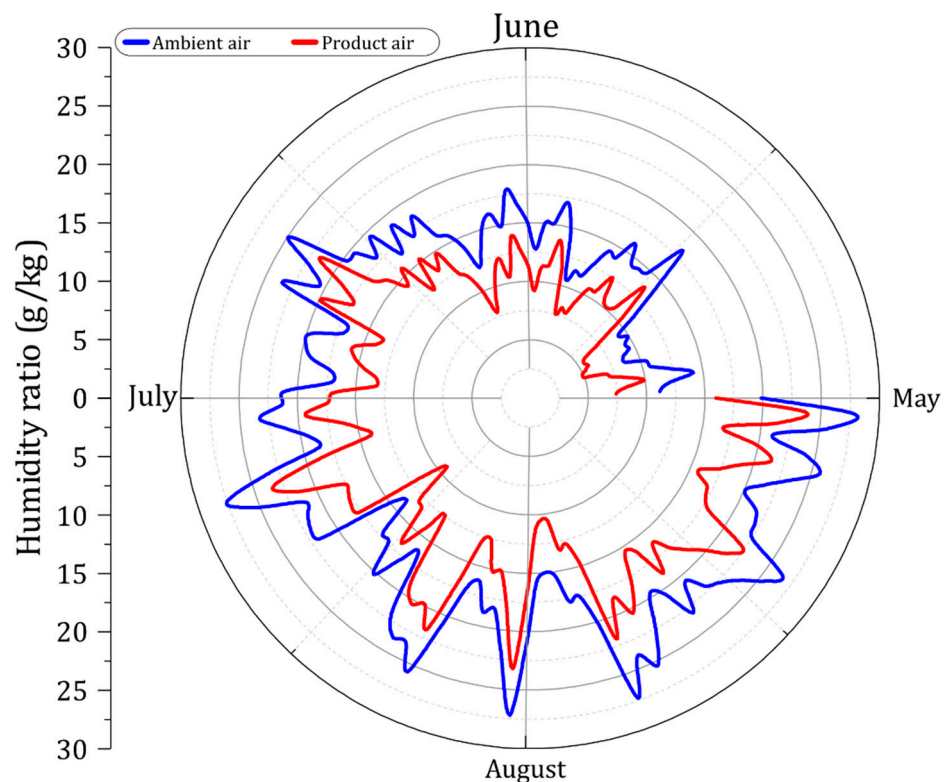


**Figure 12.** Performance profile of the proposed M-DAC system's temperature gradient in summer months of Multan (Pakistan) for greenhouse air-conditioning application.

#### 4.2. Validation Based on Optimality Degree and Comfort Ratio

Validation of the performance of the proposed M-DC system with respect to the required optimal influential parameters of the greenhouse microclimate (i.e., temperature of the air, inside RH, and vapor pressure deficit) is shown in the plots of Figure 14 for 24 h. Results clearly show that the output of the M-DAC system was very close to the requirement microclimate data generated by the OptDeg model for the vegetative to mature fruiting growth stages. Moreover, the VPD plot implies that during the mid-day hours

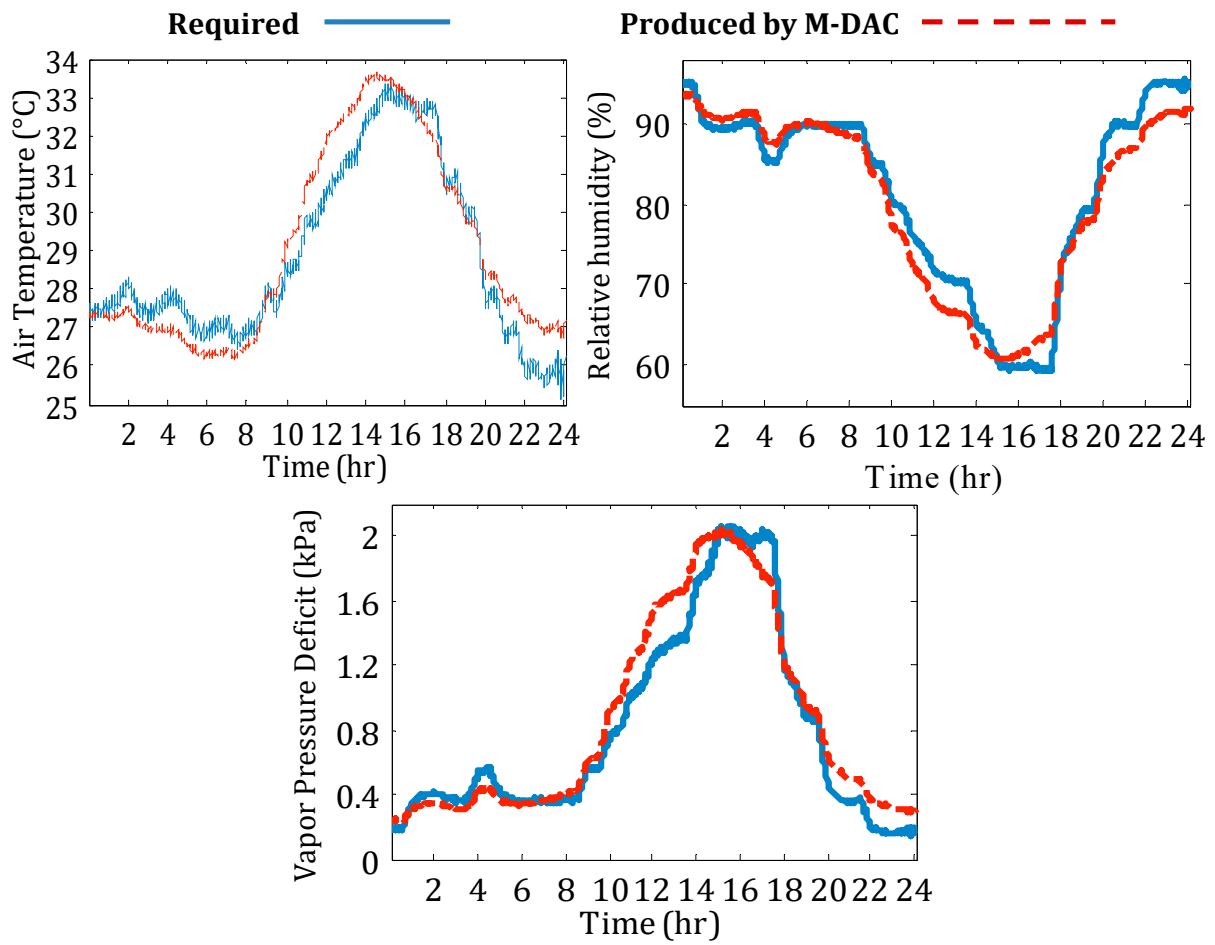
(i.e., 13:00–17:00) when the air temperature was at the highest peak and relative humidity had the lowest values, the VPD increased to 2 kPa, which is consistent with the output of the M-DAC system. Results of the VPD data generated from the OptDeg model for actual greenhouse tomato cultivation in 130 days versus the simulated VPD data generated from the M-DAC system are plotted in Figure 15. The first derivative of the VPD data sets, plotted in Figure 15a, shows that the optimum VPD values from the OptDeg model and the VPD values from the M-DAC system are not significantly different. This finding was also verified using the one-way analysis of variance (ANOVA) test for each of the five growth stages. Figure 15b shows the deviation of the VPD data of the M-DAC system from 100% optimal condition ( $\alpha = 1$ ), and compares that with actual VPD data of a commercial greenhouse that operated with ventilation and a pad-and-fan evaporative cooling system.



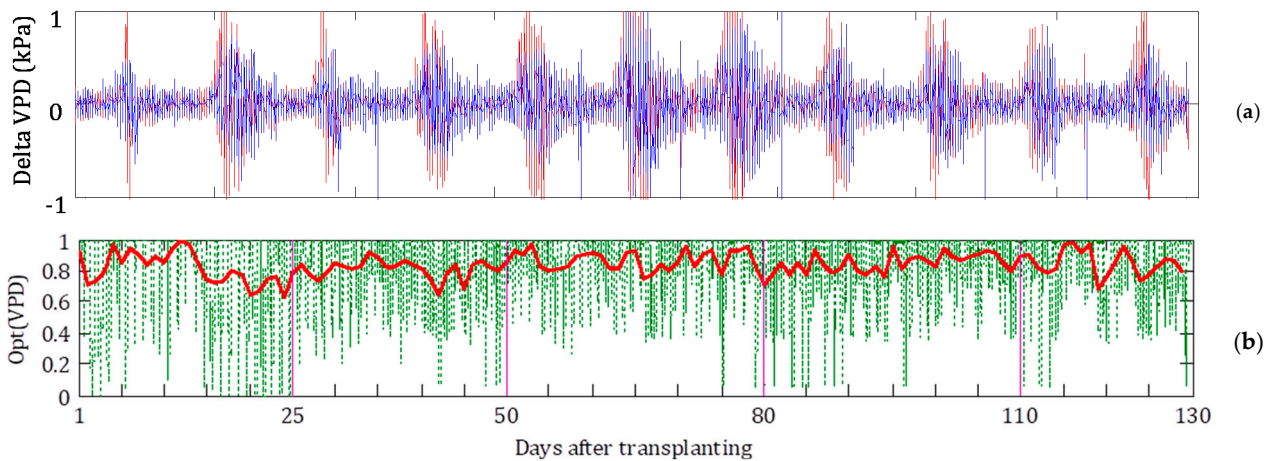
**Figure 13.** Performance profile of the proposed M-DAC system's humidity ratio in summer months of Multan (Pakistan) for greenhouse air-conditioning application.

A more in-depth result for comparing the performance of the M-DAC system and evaporative pad-and-fan system in a random day of cultivation under fruit formation growth stage is provided by means of the Cft-ratio model in the plots of Figure 16. In Figure 16, the reference conditions related to the optimal, marginal, and failure vapor pressure deficit (i.e.,  $\alpha_s = 1, 0.5$ , and 0, respectively) are presented with a green solid line, blue solid line, and red solid line, respectively. The Cft-ratio Simulink model then determined the percentage of VPD data of each case, shown in Figure 16 that were inside each reference border. From the graphical presentation, it is evident that the VPD data from the M-DAC system resulted in a higher comfort ratio than those from the evaporative pad-and-fan system. In detail, VPD values resulting from the M-DAC system never crossed the marginal reference border, whereas for the conventional pad-and-fan case, during the hours of 13:30–14:30, VPD exceeded this border and increased to 3 kPa, which is considered a failure value for greenhouse production. Sample results of the comfort ratio model for comparing the performance of the evaporative system and M-DAC system in providing optimum vapor pressure deficit for greenhouse production are given in Table 3 for two different growth stages of the tomato plant.

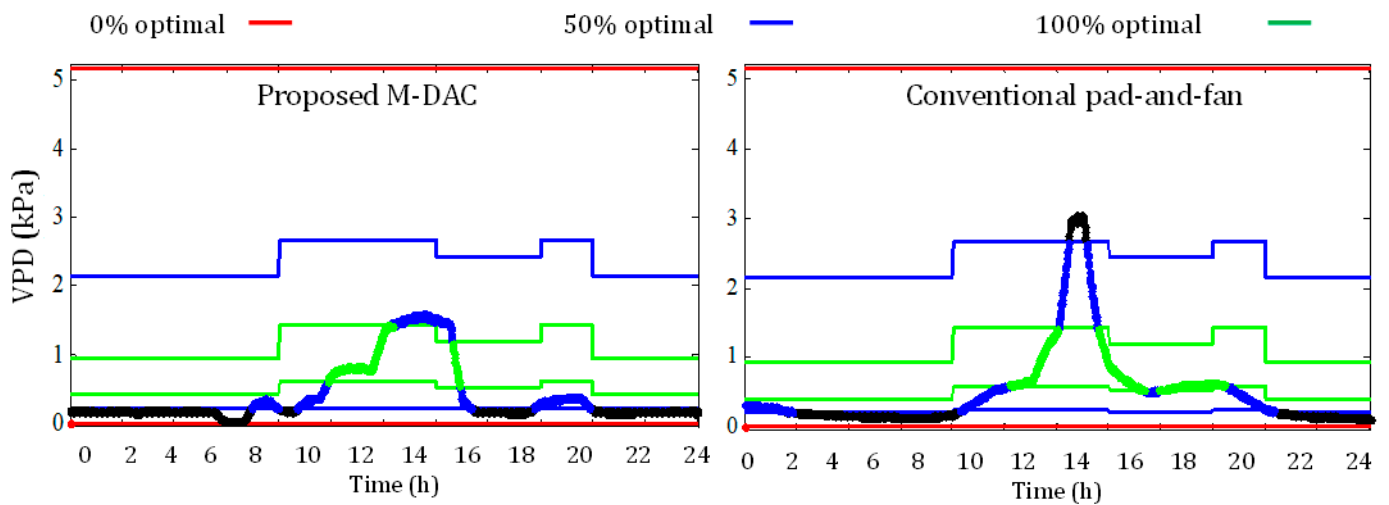




**Figure 14.** Validation of the performance of the proposed M-DAC with the OptDeg model for 24 h in greenhouse cultivation of tomato under vegetative to mature fruiting growth stages.



**Figure 15.** Variation in the VPD data (a), and optimality degree of VPD (b) resulting from the air temperature and relative humidity data of the proposed M-DAC system simulated for greenhouse cultivation of tomato in 130 days.



**Figure 16.** Comparison between comfort ratios of VPD for 24 h resulting from the proposed M-DAC system and VPD data from a conventional evaporative (pad-and-fan) cooled greenhouse.

**Table 3.** Performance validation of the proposed M-DAC system with the comfort ratio model at  $\alpha_s = 1$  in comparison with experimental data from a tomato greenhouse with evaporative pad-and-fan system [57] for 11 days under different growth stages.

Days	Vegetative Growth Stage		Flowering to Mature Fruiting	
	Pad-and-fan Ref. [57]	M-DAC (This Study)	Pad-and-Fan Ref. [57]	M-DAC (this Study)
1	8.6	21.2	11.1	24.7
2	4.9	17.2	19.6	26.7
3	27	11.9	31.9	18.2
4	13.5	20.4	32.1	24.9
5	21.9	16.4	40.9	31.0
6	25.2	36.0	27.5	43.4
7	4.6	17.5	25.8	39.0
8	32.3	51.4	36.3	65.2
9	5	6.4	8.8	25.5
10	4.3	46.8	8.3	51.0
11	18.8	20.5	32.2	29.4
Mean	15.1	24.1	24.9	34.5
SD	10.4	14.3	11.4	14.0
Min	4.3	6.4	8.3	18.2
Max	32.3	51.4	40.9	65.2

## 5. Conclusions

The present study aimed to investigate the applicability of desiccant dehumidification-based Maisotsenko cycle evaporative cooling (M-DAC) system for greenhouse air-conditioning application for the climatic conditions of Multan (Pakistan). The study area (i.e., Multan) lies in a warm desert climate of the Köppen–Geiger climatic classification. In this regard, air-conditioning is required in greenhouses to optimize the temperature and humidity conditions, which could potentially increase the production level. Therefore, the present study proposed a desiccant dehumidification-based Maisotsenko cycle evaporative cooling system (M-DAC) for greenhouse air-conditioning application. The standalone DAC system created a (maximum) dehumidification gradient (i.e., 16.8 g/kg) at 24.3 g/kg ambient air conditions, and a (maximum) temperature gradient (i.e., 8.4 °C) at 38.6 °C ambient air conditions. The DAC+HX system created a temperature gradient roughly equal to the ambient air conditions, which are unsuitable for greenhouse air-conditioning. The M-DAC system created a (maximum) temperature gradient (i.e., 21.9 °C) at 39.2 °C ambient

air conditions, which lies well within the required optimum temperature conditions of the greenhouse air-conditioning. Additionally, the M-DAC created a dehumidification gradient (i.e., 16.8 g/kg) at 24 g/kg ambient conditions, which also lies inside the range of optimum humidity conditions inside the greenhouse. Moreover, the vapor pressure deficit (VPD) profile of the greenhouse environment related to the growth of plants indicated that a VPD of 0.8–1.2 kPa (for leaf temperature equal to surrounding air temperature) is most suitable for healthy transpiration from the plant leaves at late vegetation and early flowering stages. These results were validated with OptDeg and Cft-ratio microclimate models, which showed that the M-DAC system maintained the VPD of the product air inside the healthy transpiration zone throughout most of the summer months (i.e., May to August). Moreover, the comfort ratio model results of the proposed M-DAC system were compared with experimental results from a tomato greenhouse, indicating that the M-DAC system could potentially achieve the required comfort level compared to a pad-and-fan type air-conditioning system. Thus, the study indicates the M-DAC system is a feasible option for greenhouse air-conditioning for the climatic conditions of Multan (Pakistan).

**Author Contributions:** Conceptualization, H.A., M.S. and F.A. (Farrukh Abbas); Data curation, H.A. and F.A. (Farrukh Abbas); Formal analysis, H.A., M.S., R.R.S. and F.A. (Farrukh Abbas); Funding acquisition, M.S. and R.R.S.; Investigation, M.S., R.R.S., U.S. and F.A. (Fiaz Ahmad); Methodology, H.A., R.R.S., M.F. and H.M.-T.; Project administration, M.S.; Resources, M.S.; Software, M.H.M.; Supervision, M.S.; Validation, M.S. and R.R.S.; Visualization, U.S., H.M.-T., Y.R.T., A.S. and B.M.K.N.; Writing—original draft, H.A., M.S., R.R.S. and F.A. (Farrukh Abbas); Writing—review & editing, M.F., M.H.M., F.A. (Fiaz Ahmad), Y.R.T., A.S. and B.M.K.N. All authors have read and agreed to the published version of the manuscript.

**Funding:** This research received no external funding.

**Data Availability Statement:** Data is contained within the article.

**Acknowledgments:** This research work has been carried out in the Department of Agricultural Engineering, Bahauddin Zakariya University, Multan-Pakistan. Some of this work is part of the M.Sc. (Hons.) research of Farrukh Abbas. This research was funded by Bahauddin Zakariya University, Multan-Pakistan under the Director Research/ORIC grant entitled “Development and performance evaluation of prototypes of direct and indirect evaporative cooling-based air conditioning systems” awarded to Principal Investigator Muhammad Sultan. The authors acknowledge the financial support by the Open Access Publication Fund of the Leibniz Association, Germany, and the partial research funding and editorial supports from Adaptive AgroTech Consultancy International.

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

## References

1. Pandelidis, D.; Anisimov, S. Numerical analysis of the heat and mass transfer processes in selected M-Cycle heat exchangers for the dew point evaporative cooling. *Energy Convers. Manag.* **2015**, *90*, 62–83. [[CrossRef](#)]
2. Caliskan, H.; Dincer, I.; Hepbasli, A. A comparative study on energetic, exergetic and environmental performance assessments of novel M-Cycle based air coolers for buildings. *Energy Convers. Manag.* **2012**, *56*, 69–79. [[CrossRef](#)]
3. Zhan, C.; Duan, Z.; Zhao, X.; Smith, S.; Jin, H.; Riffat, S. Comparative study of the performance of the M-cycle counter-flow and cross-flow heat exchangers for indirect evaporative cooling—Paving the path toward sustainable cooling of buildings. *Energy* **2011**, *36*, 6790–6805. [[CrossRef](#)]
4. Caliskan, H.; Dincer, I.; Hepbasli, A. Exergetic and sustainability performance comparison of novel and conventional air cooling systems for building applications. *Energy Build.* **2011**, *43*, 1461–1472. [[CrossRef](#)]
5. Chua, K.J.; Chou, S.K.; Yang, W.M.; Yan, J. Achieving better energy-efficient air conditioning—A review of technologies and strategies. *Appl. Energy* **2013**, *104*, 87–104. [[CrossRef](#)]
6. Pandelidis, D.; Anisimov, S.; Worek, W.M. Performance study of the Maisotsenko Cycle heat exchangers in different air-conditioning applications. *Int. J. Heat Mass Transf.* **2015**, *81*, 207–221. [[CrossRef](#)]
7. Rogdakis, E.D.; Koronaki, I.P.; Tertipis, D.N. Experimental and computational evaluation of a Maisotsenko evaporative cooler at Greek climate. *Energy Build.* **2014**, *70*, 497–506. [[CrossRef](#)]
8. Cui, X.; Chua, K.J.; Yang, W.M. Use of indirect evaporative cooling as pre-cooling unit in humid tropical climate: An energy saving technique. *Energy Procedia* **2014**, *61*, 176–179. [[CrossRef](#)]

9. Rianguvilaikul, B.; Kumar, S. Numerical study of a novel dew point evaporative cooling system. *Energy Build.* **2010**, *42*, 2241–2250. [[CrossRef](#)]
10. Sultan, M. Study on Sorption Characteristics of Water Adsorbents for Agricultural Air-Conditioning Systems. Ph.D. Thesis, Kyushu University, Kyushu, Japan, 2015.
11. Anisimov, S.; Pandelidis, D.; Jedlikowski, A.; Polushkin, V. Performance investigation of a M (Maisotsenko)-cycle cross-flow heat exchanger used for indirect evaporative cooling. *Energy* **2014**, *76*, 593–606. [[CrossRef](#)]
12. Zhao, X.; Li, J.M.; Riffat, S.B. Numerical study of a novel counter-flow heat and mass exchanger for dew point evaporative cooling. *Appl. Therm. Eng.* **2008**, *28*, 1942–1951. [[CrossRef](#)]
13. Zube, D.; Gillan, L. Evaluating Coolerado Corporation's Heat and mass exchanger performance through experimental analysis. *Int. J. Energy Clean Environ.* **2011**, *12*, 101–116. [[CrossRef](#)]
14. Weerts, B. Coolerado and modeling an application of the Maisotsenko Cycle. *Int. J. Energy Clean Environ.* **2011**, *12*, 287–307. [[CrossRef](#)]
15. Maisotsenko, V.; Treyger, I. Way to energy abundance can be found through the Maisotsenko cycle. *Int. J. Energy Clean Environ.* **2011**, *12*, 319–326. [[CrossRef](#)]
16. Khalatov, A.; Karp, I.; Isakov, B. Prospects of the Maisotsenko thermodynamic cycle application in Ukraine. *Int. J. Energy Clean Environ.* **2011**, *12*, 141–157. [[CrossRef](#)]
17. Anisimov, S.; Pandelidis, D. Heat-and mass-transfer processes in indirect evaporative air conditioners through the Maisotsenko cycle. *Int. J. Energy Clean Environ.* **2011**, *12*, 273–286. [[CrossRef](#)]
18. Rogdakis, E.D.; Tertipis, D.N. Maisotsenko cycle: Technology overview and energy-saving potential in cooling systems. *Energy Emiss. Control Technol.* **2015**, *3*, 15.
19. Nagaya, K.; Senbongi, T.; Li, Y.; Zheng, J.; Murakami, I. High energy efficiency desiccant assisted automobile air-conditioner and its temperature and humidity control system. *Appl. Therm. Eng.* **2006**, *26*, 1545–1551. [[CrossRef](#)]
20. Lee, S.H.; Lee, W.L. Site verification and modeling of desiccant-based system as an alternative to conventional air-conditioning systems for wet markets. *Energy* **2013**, *55*, 1076–1083. [[CrossRef](#)]
21. Ismail, M.Z.; Angus, D.E.; Thorpe, G.R. The performance of a solar-regenerated open-cycle desiccant bed grain cooling system. *Sol. Energy* **1991**, *46*, 63–70. [[CrossRef](#)]
22. Sultan, M.; Miyazaki, T.; Koyama, S.; Saha, B.B. Utilization of Desiccant Air-Conditioning System for Improvement in Greenhouse Productivity: A Neglected Area of Research in Pakistan. *Int. J. Environ.* **2014**, *4*, 1–10.
23. Zheng, G.; Zheng, C.; Yang, G.; Chen, W. Development of a new marine rotary desiccant airconditioning system and its energy consumption analysis. *Energy Procedia* **2011**, *16*, 1095–1101. [[CrossRef](#)]
24. Bourdoukan, P.; Wurtz, E.; Joubert, P. Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components. *Energy* **2010**, *35*, 1057–1067. [[CrossRef](#)]
25. Enteria, N.; Yoshino, H.; Takaki, R.; Yonekura, H.; Satake, A.; Mochida, A. First and second law analyses of the developed solar-desiccant air-conditioning system (SDACS) operation during the summer day. *Energy Build.* **2013**, *60*, 239–251. [[CrossRef](#)]
26. Dezfouli, M.M.S.; Mat, S.; Pirasteh, G.; Sahari, K.S.M.; Sopian, K.; Ruslan, M.H. Simulation Analysis of the Four Configurations of Solar Desiccant Cooling System Using Evaporative Cooling in Tropical Weather in Malaysia. *Int. J. Photoenergy* **2014**, *2014*, 843617. [[CrossRef](#)]
27. Enteria, N.; Yoshino, H.; Satake, A.; Mochida, A.; Takaki, R.; Yoshie, R.; Mitamura, T.; Baba, S. Experimental heat and mass transfer of the separated and coupled rotating desiccant wheel and heat wheel. *Exp. Therm. Fluid Sci.* **2010**, *34*, 603–615. [[CrossRef](#)]
28. Ge, T.S.; Ziegler, F.; Wang, R.Z.; Wang, H. Performance comparison between a solar driven rotary desiccant cooling system and conventional vapor compression system (performance study of desiccant cooling). *Appl. Therm. Eng.* **2010**, *30*, 724–731. [[CrossRef](#)]
29. Sultan, M.; El-Sharkawy, I.I.; Miyazaki, T.; Saha, B.B.; Koyama, S.; Maruyama, T.; Maeda, S.; Nakamura, T. Water vapor sorption kinetics of polymer based sorbents: Theory and experiments. *Appl. Therm. Eng.* **2016**, *106*, 192–202. [[CrossRef](#)]
30. Sultan, M.; Miyazaki, T.; Koyama, S.; Khan, Z.M. Performance evaluation of hydrophilic organic polymer sorbents for desiccant air-conditioning applications. *Adsorpt. Sci. Technol.* **2018**, *36*, 311–326. [[CrossRef](#)]
31. Sultan, M.; El-Sharkawy, I.I.; Miyazaki, T.; Saha, B.B.; Koyama, S.; Maruyama, T.; Maeda, S.; Nakamura, T. Insights of water vapor sorption onto polymer based sorbents. *Adsorption* **2015**, *21*, 205–215. [[CrossRef](#)]
32. Sultan, M.; Miyazaki, T.; Mahmood, M.H.; Khan, Z.M. Solar assisted evaporative cooling based passive air-conditioning system for agricultural and livestock applications. *J. Eng. Sci. Technol.* **2018**, *13*, 693–703.
33. Noor, S.; Ashraf, H.; Sultan, M.; Khan, Z.M. Evaporative Cooling Options for Building Air-Conditioning: A Comprehensive Study for Climatic Conditions of Multan (Pakistan). *Energies* **2020**, *13*, 3061. [[CrossRef](#)]
34. Sultan, M.; Miyazaki, T. Energy-Efficient Air-Conditioning Systems for Nonhuman Applications. In *Refrigeration*; Ekren, O., Ed.; InTech: London, UK, 2017; pp. 97–117.
35. Sultan, M.; El-Sharkawy, I.I.; Miyazaki, T.; Saha, B.B.; Koyama, S. An overview of solid desiccant dehumidification and air conditioning systems. *Renew. Sustain. Energy Rev.* **2015**, *46*, 16–29. [[CrossRef](#)]
36. Mahmood, M.H.; Sultan, M.; Miyazaki, T.; Koyama, S.; Maisotsenko, V.S. Overview of the Maisotsenko cycle—A way towards dew point evaporative cooling. *Renew. Sustain. Energy Rev.* **2016**, *66*, 537–555. [[CrossRef](#)]

37. Raza, H.M.U.; Ashraf, H.; Shahzad, K.; Sultan, M.; Miyazaki, T.; Usman, M.; Shamshiri, R.R.; Zhou, Y.; Ahmad, R. Investigating Applicability of Evaporative Cooling Systems for Thermal Comfort of Poultry Birds in Pakistan. *Appl. Sci.* **2020**, *10*, 4445. [[CrossRef](#)]
38. Kanwal, R.; Ashraf, H.; Sultan, M.; Babu, I.; Yasmin, Z.; Nadeem, M.; Asghar, M.; Shamshiri, R.R.; Ibrahim, S.M.; Ahmad, N.; et al. Effect of 1-Methyl Cyclopropane and Modified Atmosphere Packaging on the Storage of Okra (*Abelmoschus esculentus* L.): Theory and Experiments. *Sustainability* **2020**, *12*, 7547. [[CrossRef](#)]
39. Shamshiri, R.R.; Kalantari, F.; Ting, K.C.; Thorp, K.R.; Hameed, I.A.; Weltzien, C.; Ahmad, D.; Shad, Z.M. Advances in greenhouse automation and controlled environment agriculture: A transition to plant factories and urban agriculture. *Int. J. Agric. Biol. Eng.* **2018**, *11*, 1–22. [[CrossRef](#)]
40. Shamshiri, R.R.; Jones, J.W.; Thorp, K.R.; Ahmad, D.; Man, H.C.; Taheri, S. Review of optimum temperature, humidity, and vapour pressure deficit for microclimate evaluation and control in greenhouse cultivation of tomato: A review. *Int. Agrophysics* **2018**, *32*, 287–302. [[CrossRef](#)]
41. Shamshiri, R.; Ismail, W.I.W. A review of greenhouse climate control and automation systems in tropical regions. *J. Agric. Sci. Appl.* **2013**, *2*, 176–183. [[CrossRef](#)]
42. Shamshiri, R.; Ismail, W.I.W.; bin Ahmad, D. Experimental evaluation of air temperature, relative humidity and vapor pressure deficit in tropical lowland plant production environments. *Adv. Environ. Biol.* **2014**, *8*, 5–14.
43. Sultan, M.; Miyazaki, T.; Koyama, S. Optimization of adsorption isotherm types for desiccant air-conditioning applications. *Renew. Energy* **2018**, *121*, 441–450. [[CrossRef](#)]
44. Sultan, M.; Miyazaki, T.; Saha, B.B.; Koyama, S. Steady-state investigation of water vapor adsorption for thermally driven adsorption based greenhouse air-conditioning system. *Renew. Energy* **2016**, *86*, 785–795. [[CrossRef](#)]
45. Song, J.; Sobhani, B. Energy and exergy performance of an integrated desiccant cooling system with photovoltaic/thermal using phase change material and maisotsenko cooler. *J. Energy Storage* **2020**, *32*, 101698. [[CrossRef](#)]
46. Miyazaki, T.; Nikai, I.; Akisawa, A. Simulation analysis of an open-cycle adsorption air conditioning system—numeral modeling of a fixed bed dehumidification unit and the maisotsenko cycle cooling unit. *Int. J. Energy Clean Environ.* **2011**, *12*, 341–354. [[CrossRef](#)]
47. Pandelidis, D.; Anisimov, S.; Worek, W.M.; Drag, P. Analysis of different applications of Maisotsenko cycle heat exchanger in the desiccant air conditioning systems. *Energy Build.* **2017**, *140*, 154–170. [[CrossRef](#)]
48. Goldsworthy, M.; White, S. Optimisation of a desiccant cooling system design with indirect evaporative cooler. *Int. J. Refrig.* **2011**, *34*, 148–158. [[CrossRef](#)]
49. Sultan, M.; El-Sharkaw, I.I.; Miyazaki, T.; Saha, B.B.; Koyama, S. Experimental Study on Carbon Based Adsorbents for Greenhouse Dehumidification. *Evergreen* **2014**, *1*, 5–11. [[CrossRef](#)]
50. Lim, X. How heat from the sun can keep us all cool. *Nature* **2017**, *542*, 23–24. [[CrossRef](#)]
51. Beck, H.E.; Zimmermann, N.E.; McVicar, T.R.; Vergopolan, N.; Berg, A.; Wood, E.F. Present and future Köppen-Geiger climate classification maps at 1-km resolution. *Sci. Data* **2018**, *5*, 180214. [[CrossRef](#)]
52. Panaras, G.; Mathioulakis, E.; Belessiotis, V.; Kyriakis, N. Theoretical and experimental investigation of the performance of a desiccant air-conditioning system. *Renew. Energy* **2010**, *35*, 1368–1375. [[CrossRef](#)]
53. Panaras, G.; Mathioulakis, E.; Belessiotis, V. Solid desiccant air-conditioning systems—Design parameters. *Energy* **2011**, *36*, 2399–2406. [[CrossRef](#)]
54. ASHRAE. *ASHRAE Handbook—Fundamentals*; American Society of Heating, Refrigerating, and Air-Conditioning Engineers: Atlanta, GA, USA, 2017.
55. Shamshiri, R. Measuring optimality degrees of microclimate parameters in protected cultivation of tomato under tropical climate condition. *Measurement* **2017**, *106*, 236–244. [[CrossRef](#)]
56. Shamshiri, R.R.; Bojic, I.; van Henten, E.; Balasundram, S.K.; Dworak, V.; Sultan, M.; Weltzien, C. Model-based evaluation of greenhouse microclimate using IoT-Sensor data fusion for energy efficient crop production. *J. Clean. Prod.* **2020**, *263*, 121303. [[CrossRef](#)]
57. Rezvani, S.M.; Abyaneh, H.Z.; Shamshiri, R.R.; Balasundram, S.K.; Dworak, V.; Goodarzi, M.; Sultan, M.; Mahns, B. IoT-Based Sensor Data Fusion for Determining Optimality Degrees of Microclimate Parameters in Commercial Greenhouse Production of Tomato. *Sensors* **2020**, *20*, 6474. [[CrossRef](#)] [[PubMed](#)]