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Controlled Environment Agriculture (CEA): A Promising Solution with Challenges in Tropical Zones

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Abstract. In tropical greenhouses, single-stage direct evaporative cooling systems (DECS) are commonly employed. However, a key limitation of DECS is the inability to achieve desired temperature reductions. This study investigated and compared a two-stage pre-cooling system utilizing a heat exchanger in combining with DECS, against the traditional single-stage approach. The findings revealed a significant improvement in the two-stage system, achieving a 1.2°C lower in dry bulb temperature. Additionally, the magnitude of air condition change was 1.26 greater compared to the single-stage system. Furthermore, the two-stage system exhibited greater data consistency, indicating a more precise cooling effect.

1 Introduction

The escalating global population presents a multifaceted challenge for researchers, demanding innovative solutions to ensure food security. A seemingly modest 2°C rise in global average temperature, as projected by NASA [1], could exacerbate existing drought conditions, leading to a significant threat to traditional agricultural practices. Additionally, plant cultivation is susceptible due to its reliance on freshwater resources. While agriculture accounts for approximately 70% of global freshwater consumption annually, only 30% is directly utilized for crop growth and livestock production [2]. The remaining 40% is lost through evapotranspiration and inefficient water management practices. The Controlled Environment Agriculture (CEA) offers a potential solution alongside hydroponics. This synergistic approach holds promise for improved water efficiency, resource utilization, and crop yield [3]. Differ from temperate and frigid zones, the tropical CEA systems frequently utilize evaporative cooling methods to maintain optimal growing conditions [4],[5]. Thailand, a hot and humid country, experiences average temperatures ranging from 24.1°C to 33.8°C and humidity levels between 73% and 83% [6]. In such conditions, employing evaporative cooling for CEA might not achieve sufficient temperature reduction beyond the adiabatic saturation temperature. Increasing of the thickness of a cooling pad and fogging play a crucial role in system efficiency [7]. Thicker pads enhance cooling efficiency by increasing the surface area, which allows for more effective heat exchange between the water and air. Furthermore, the fogging involves atomizing water into small droplets (smaller than 50

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micron). Water droplets absorb heat from the air for evaporating to lowers the air temperature and raises the relative humidity. However, the minimum air temperature produced would not be lower than the saturation temperature that follows the adiabatic line (constant enthalpy) of certain initial air conditions. In practical applications, the temperature of the water used in conjunction with a cooling pad is typically lower than the ambient air temperature. Consequently, the changes in ambient conditions following passage through the cooling pad do not adhere to the adiabatic process. Instead, the actual temperature reduction tends to be less pronounced due to heat loss from the ambient air to the cooler intake water.

Furthermore, aside from the reduction in ambient air temperature due to the lower intake water temperature, an additional strategy is to pre-cool the approach ambient air. This pre-cooling process effectively lowers the air temperature while maintaining the existing moisture content, thereby following the dewpoint temperature line. The intake water circulates through pipes within the heat exchanger, where it facilitates the transfer of sensible heat from the intake air to the cooler water. This heat exchange process results in a reduction in air temperature. As the air temperature decreases, the amount of heat entering the evaporative cooling system is reduced, which tends to make the air temperature drop more effectively than in a traditional evaporative cooling system. Therefore, the pre-cooling the air entering the evaporative cooling system has been proposed as a potential strategy to enhance its effectiveness. By reducing the temperature of the air, pre-cooling can theoretically lead to a greater temperature decrease during the evaporation process.

This study, therefore, aims to evaluate the effectiveness of two cooling strategies for CEA systems in tropical climates. The first approach utilizes a traditional CEA system with evaporative cooling as well as the second approach implements a two-stage cooling system, combining pre-cooling with evaporative cooling.

2 Materials and methods

2.1 Environment control chamber preparation

The experiment utilized two types of chambers for a single-stage cooling system and a two-stage cooling system, illustrated in Fig. 1. Each chamber is combined by 2 types of modules which are rectangular tube modules and the standard top-flat pyramid. The rectangular module was a 0.75 m x 0.5 m cross-sectional area. The standard top-flat pyramid module was connected with rectangular module while the other end connected with 1 m x 1 m size of fan frame. The length of the chamber for the single-stage cooling system was 8 m, while the two-stage cooling system was 1 m longer.

2.2 Fixed parameters for experiment

The single-stage cooling system was the DECS using the evaporative pad. While the two-stage cooling system was the combination of the pre-cooling system (PCS) and the DECS. The heat exchanger, employing convection process of heat transfer between water in tube and air, was employed for the PCS. The air was, then, passed through the DECS. The water supply for the DECS was 10 kg/min (point 1), and for the pre-cooling system was 37.0 kg/min and the average air ventilation measured at point B and D was about 1.0 m/min.

2.3 Data collection

The dry bulb temperature (T_{db}) and the relative humidity of the air were measured at position A, B, C and D at the same time, the dew point temperatures (T_{dp}) were, then, calculated. The water temperature used in DECS and PCS were also measured at position 1 through 9. The data were recorded for 30 samples throughout the 17 days (about 1-2 times per day).

2.4 Data analysis through Psychrometric chart

Psychrometric charts were used to visualize the air properties. Each data point on the chart represents a state of air, with Tdb plotted on i-direction and T_{dp} plotted on j-direction. The magnitude of the vector, $T_{db}\,\vec{i} + T_{dp}\,\vec{j}$, connecting the initial state (A) to the final state (B or D) quantified the change in air conditions achieved by the single-stage and two-stage cooling systems, respectively. Data variation was assessed by the area of a rectangle on the psychrometric chart encompassing the data points. The width of this rectangle represents the difference in T_{db}, while the height represents the difference in T_{dp}. Finally, the heat transfer from air to water was calculated using the enthalpy difference between the initial and final states.

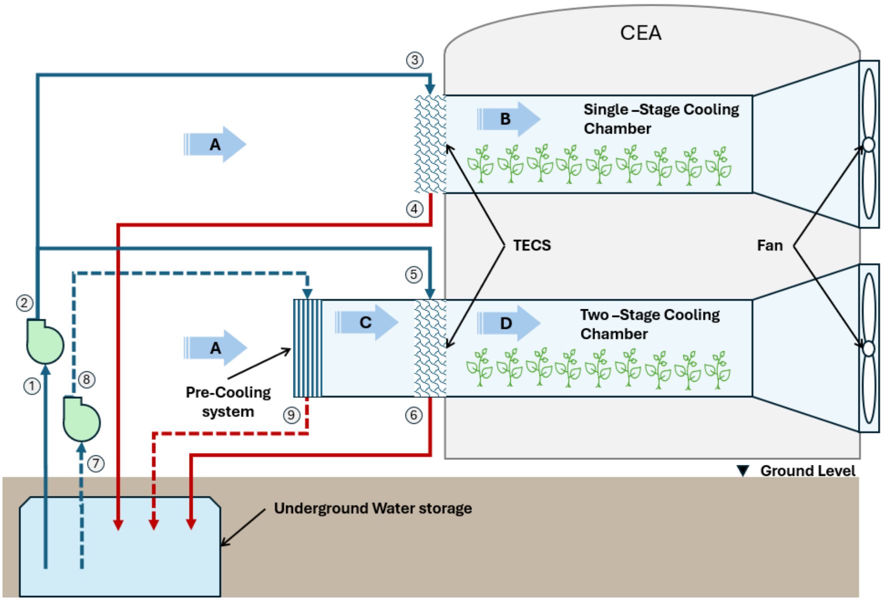

Fig. 1 Schematic diagram of the experiment

3 Results and discussions

3.1 The water temperature and the initial air conditions throughout the experiment

The average intake water temperature in the underground water storage was 27.0 C and the standard deviation was 1.0 C which showed the more stable intake water temperature throughout the experiment (illustrated in Fig 2). Therefore, the intake water temperature at point 1 and 7 were as same as the water temperature from the underground water storage at the certain day of experiment. The initial air conditions, including dry bulb temperature and relative humidity, were recorded and are depicted in Fig. 2. The data indicate that the ambient dry bulb temperatures were relatively stable, with an average of 33.4°C and a standard deviation of 1.62°C. In contrast, relative humidity values exhibited greater variability, with an average of 66.3%, and ranged from a maximum of 86.3% to a minimum of 44.8%. This wide range resulted in a larger standard deviation of 9.88%.

To combine each initial air condition (T_{db} and %RH) was practical; however, to combine the standard deviation of initial air condition was challenging. The idea of uniting the data deviation was created by plotting initial ambient conditions (T_{db} and %RH) in psychrometric chart (sea level). Then, the lines were created to cover all data represented in rectangular shape presented in Fig 3. The height was determined by the difference between the maximum dew point temperature (T_{dp}) at the highest point on the psychrometric chart and the minimum dew point temperature at the lowest point on the chart. The width was defined as the difference between the highest and lowest dry bulb temperatures. Greater variation in the data resulted in a larger height and width of the rectangle, thereby increasing the covered area. Therefore, the data variation of initial ambient condition could be performed by the covered rectangular area.

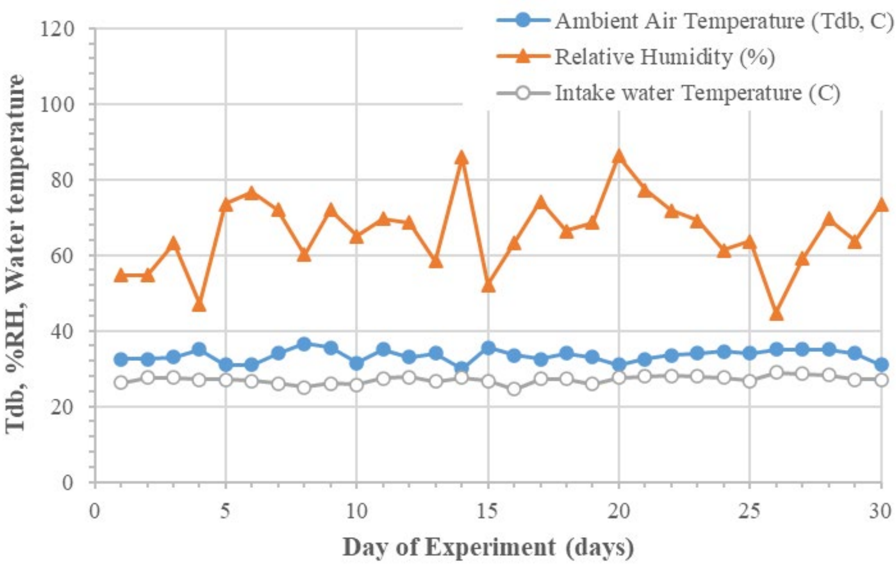

Fig. 2 Initial ambient conditions of each day of experiment

Fig 3 illustrates the initial ambient conditions at position A, as shown earlier in Fig 1. The average T_{db} was 34.4°C, with a minimum of 30.5°C and a maximum of 38.5°C resulting in the width of 8 C for the rectangle. The T_{dp} ranged from a minimum of 22.2°C to a maximum of 30.3°C, with an average of 27.1C, giving the rectangle a heigh of 8.1 C. The area of rectangular representing data variation was 64.8°C².

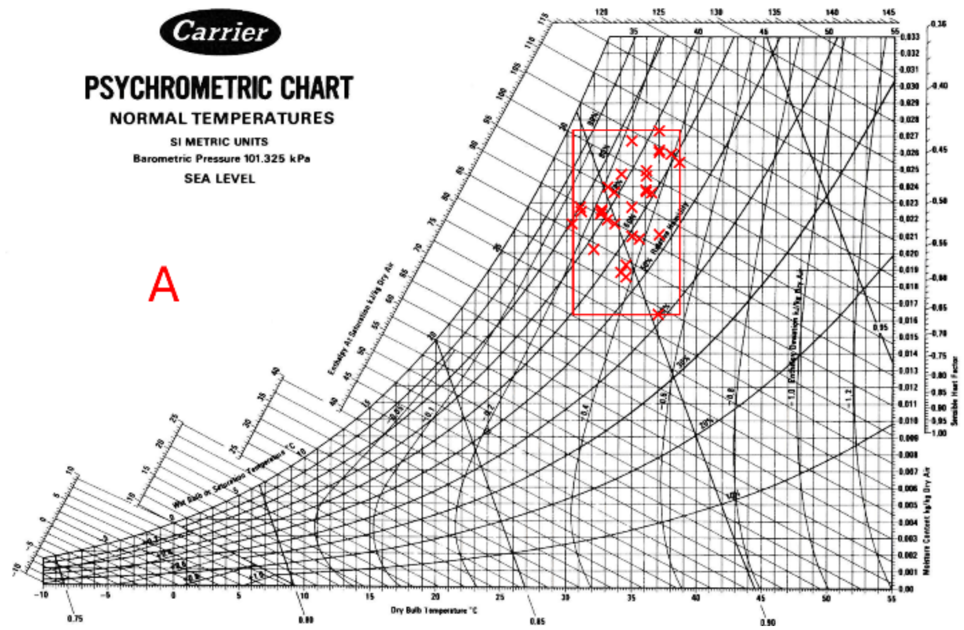

Fig. 3 The initial ambient conditions of each day of experiment plotted in psychrometric chart (sea level)

3.2 The ambient conditions of the single-stage cooling chamber and of the two-stage cooling chamber

The results of the single-stage cooling chamber (B), and the two stages cooling chamber (C and D) were plotted individually in the psychrometric charts illustrated in Fig 4 and Fig 5 respectively. The single-stage cooling chamber (B) achieved an average T_{db} of 27.4°C, with a minimum of 25.0°C and a maximum of 29.5°C resulting in the width of 4.5°C for the rectangle. The average T_{dp} was 26.1°C, with a minimum and a maximum T_{dp} were 20.1°C and 28.2°C respectively, resulting in the heigh of 8.1°C for the rectangle. After passing through point B, the ambient air condition reached a humidity ratio of approximately 90%. The area of rectangular representing data variation was 36.45°C². As a result, the variation of the ambient air condition after passing through the single-stage cooling chamber was more stable compared to the initial air condition, as evidenced the reduction in data variation from 64.8°C² to 36.45°C². This improvement suggests the potential for better control of the ambient air conditions to meet desired specifications. However, the 90%RH.

The results of the two-stage cooling chamber were explained here. At point C, the average T_{db} was 28.8°C, with corresponding T_{dp} of 25.1°C. The minimum and the maximum of T_{db} were 26.5°C and 32.0°C respectively, resulting in the width of 5.5°C. while the minimum and the maximum of T_{dp} were 18.3°C and 29.2°C respectively, resulting in the width of 10.7°C. The area of rectangular representing data variation was 59.95°C². The trend of the results showed the alignment with the previous deviation results of the single-stage cooling chamber (B).

After the ambient air stream passed through point C, the average dry bulb temperature (T_{db}) decreased from 34.4°C to 28.8°C. However, this reduction in air temperature occurred along a constant moisture ratio line, which resulted in an increase in relative humidity. At point C, the average relative humidity of the ambient air was approximately 80%. To achieve the same humidity ratio of 90% as in the single-stage cooling chamber, approximately 10% of the vapor needed to be evaporated into the air to reach this level.

The results of the two-stage cooling chamber (point D) achieved average T_{db} and T_{dp} of 26.2°C and 25.2°C respectively. The minimum and the maximum of T_{db} were 24.1°C and 28.0°C respectively, resulting in the width of 4.0°C. while the minimum and the maximum of T_{dp} were 20.3°C and 28.0°C respectively, resulting in the width of 7.7°C. At point C, the average relative humidity of the ambient air was approximately 90%, which is nearly equivalent to the relative humidity of the air stream that passed through the single-stage cooling chamber (B). The area of rectangular representing data variation was 30.03°C². The results indicated that the two-stage cooling chamber achieved a lower temperature, reducing the T_{db} from 28.8°C to 26.2°C. Additionally, the variation in ambient data decreased compared to point B, suggesting that the two-stage cooling chamber not only produced a lower temperature but also resulted in more stable ambient conditions.

The magnitude of the vector from point A to point B represents the change in ambient air conditions achieved by the single-stage system (7.54 units). Similarly, the magnitude of the vector from point A to point D represents the change achieved by the two-stage system (8.80 units). Furthermore, the amount of heat removed from the air and transferred to the water in the single-stage and two-stage cooling systems was calculated using the enthalpy difference between points. The enthalpy change from point A to B (single-stage) was 2.70 kJ/min, while the changes from A to C (two-stage, stage 1) and C to D (two-stage, stage 2) were 2.47 kJ/min and 1.36 kJ/min, respectively. Theoretically, the heat removed in stages A-B and C-D likely contributed to the latent heat of vaporization for water evaporation. The remaining heat removed in stage A-C represents sensible heat that increased the water temperature from pre-cooling system by a small amount (approximately 0.06°C). However, the limitations of the thermometer used may have prevented detecting such a small temperature rise, leading to the recorded value of 27.1°C

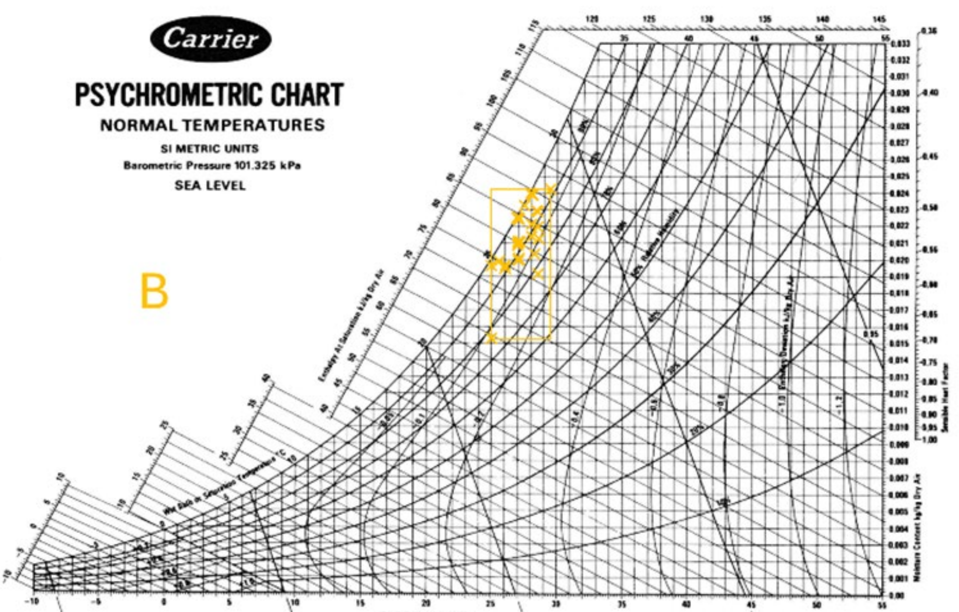

Fig. 4 The ambient conditions for each day of the experiment were plotted on a psychrometric chart (at sea level) for the single-stage cooling system (B).

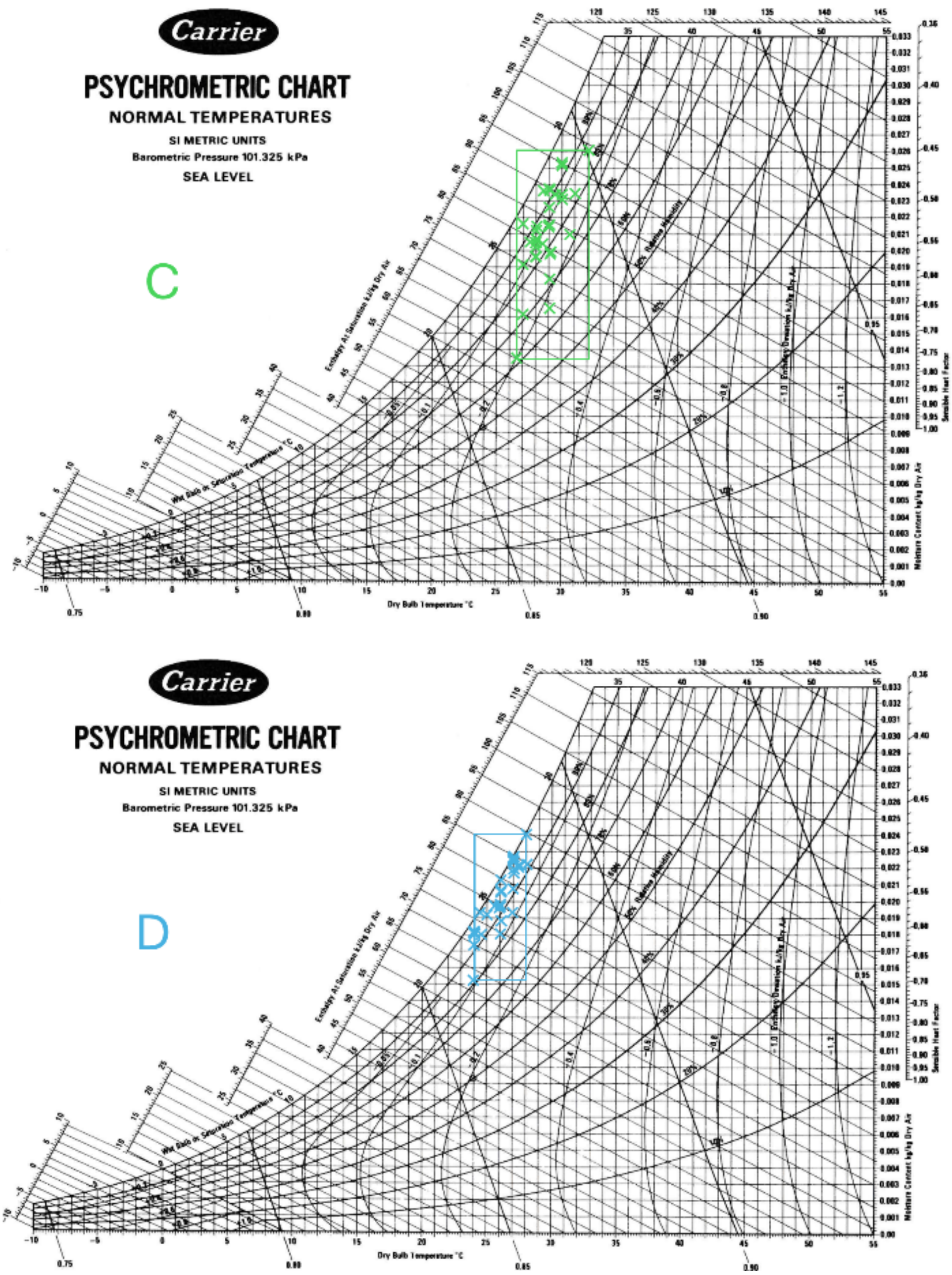

Fig. 5 The ambient conditions for each day of the experiment were plotted on a psychrometric chart (at sea level) for the two-stage cooling systems (C and D).

4 Conclusions

The single-stage cooling system reduced the dry bulb temperature from 33.4°C to 27.4°C, achieving a decrease of approximately 6°C while maintaining a humidity ratio of about 90% and exhibiting a data variation of 59.95°C². In contrast, the two-stage cooling system lowered the dry bulb temperature to 26.2°C, a reduction of about 7.2°C, with a similar humidity ratio of approximately 90% and a data variation of 30.03°C². This result is also supported by the magnitude of the vectors, which were 7.54 units for the single-stage system and 8.80 units for the two-stage system. The two-stage cooling system thus demonstrated a greater reduction in temperature and a lower variation in data, indicating a more efficient and consistent cooling process compared to the single-stage system. Therefore, the two-stage system holds promise for application in greenhouses in tropical zones to more effectively reduce internal temperatures.

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