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Investigation of Climate Control Techniques for Tropical Lowland Greenhouses in Malaysia

[1]R. Shamshiri and [2]W.I. Wan Ismail
[1]Department of Agricultural and Biological Engineering, University of Florida, Gainesville, FL 32611, United States of America
[2]Department of Biological and Agricultural Engineering, Universiti Putra Malaysia, Serdang, Selangor, Malaysia

Abstract: The efficiency of greenhouse production in tropical lowland greenhouses of Malaysia was investigated and found to be less than optimal due to inefficiency of the employed climate control systems. Major constraints in this region are the high temperature, high humidity, typhoons and heavy seasonal rains; therefore, selecting the right method for manipulating greenhouse environment under specific exterior conditions is critical and important to improve efficiency and increase benefits. Preliminary observation of three major greenhouse sites in lowland Malaysia indicates that new technologies from other countries have been utilized widely without appropriate modification for tropical environments. The objective of this study was to address the interactions of environment and evaporative cooling methods for the situations of lowland Malaysia. An engineering analysis approach was used for this purpose. Results show that the amount of cooling that an evaporative system can achieve, highly depends on the water content of the outside air. It was then concluded that, these systems are most effective in regions with relative humidity less than 65%.

Key words: Greenhouse production, controlled environment, evaporative cooling, tropical lowland Malaysia

INTRODUCTION

A modern commercial greenhouse is considered a Controlled Environment Plant Production System (CEPPS) that provides high yields at low expenses. The purpose of CEPPS is to keep the production competitive through automatic control of the environmental parameters, including air temperature, Relative Humidity (RH), light level and Carbon Dioxide (CO_2) concentration. This can be quiet challenging due to the dynamic system that is influenced by changes of internal and external factors, such as changes in wind speed or the temperature outside the greenhouse which affects the RH of the inside climate (Fuchs *et al*., 2006). In addition, the greenhouse structure design and material can indirectly affect other microclimatic factors like air RH and CO_2 concentration (Jaafar, 2006).

The main purpose of using greenhouse in hot and humid environments such as lowlands of Malaysia is to protect plants from extreme temperature, rain, wind and insects, in such a way that optimal plant growing conditions are attained. Tropical Lowland Greenhouses (TLGs) in Malaysia was introduced in the mid-1980's using simple technology of plastic covered rain shelters (Hawa, 1998). Their performance showed significant potential in terms of economic and year-round production capability with increased productivity (Hawa, 1990); however, developments of plants disease due to the high temperature and misting condition have always been serious problems. A comparison between greenhouse production in Malaysia and the Netherlands by Jaafar (2006), shows that temperate crops have been grown locally but the production is still insufficient in lowland environments and does not meet the large market demand. Study of TLG in Malaysia (Hawa, 2003), reveals that with optimal crop growing temperature (27°C) and CO_2 enrichment, tomato, sweet pepper and celery yields under TLG can increase between 10-15 times more than open cultivation and crop maturity period will be shortened about 30% while products keeping quality enhances.

Air temperature and RH inside greenhouse are usually controlled by natural and mechanical ventilation, evaporative cooling techniques and shading. Evaporative cooling by means of pad-and-fan, misting or high-pressure fog systems is based on the evaporation of water into an air stream which results in reducing air temperature. In a misting system, high pressure water is passed through nozzles with orifice sizes usually less

Corresponding Author: R. Shamshiri, Department of Agricultural and Biological Engineering, University of Florida, Gainesville, FL 32611, United States of America

than 10 μm. A fan then blows the extremely small droplets of water into greenhouse air and reduces temperature through an evaporative process. A major drawback with this method is that it creates high humidity climate inside canopies which facilitates development of bacterial diseases, such as Algae. Pad-and-fan systems, depending on the size of greenhouse, utilize one or more exhaust fans installed at one end of the greenhouse with a large pad at the opposite end. A pump circulates water through and over the pad which allows the thermal energy of the air to be absorbed by water.

The cooling potential of fan-and-pad ventilated greenhouse was addressed long back by Morris (1956). Measurement and data analysis for greenhouse evaporative cooling has been discussed by Kittas *et al*. (2001). Decrease in air temperature by 4-5°C inside greenhouse with pad-and-fan evaporative cooling has been reported by Jain and Tiwari (2002). Performance of a two stage pad cooling system in broiler houses was analyzed by Petek *et al*. (2012). Modeling and Simulation of Evaporative Cooling System in greenhouse was done by Fahmy *et al*. (2012). Low and high pressure fogging systems in a naturally ventilated greenhouse have been studied and compared by Li and Willits (2008). A thermal model for prediction of microclimate factors inside a greenhouse with mechanical ventilation and evaporative cooling system was introduced by Willits (2003). A comprehensive review of ventilation systems in greenhouse is available in the work of Ganguly and Ghosh (2011).

For the purpose of this study, three major TLGs sites in Malaysia, including the Malaysian Agricultural Research and Development Institute (MARDI), the Seremban site and the Taman Pertanian University (the Union Agricultural Park, TPU) were visited. Preliminary observation revealed several issues such as high RH and ambient temperatures. The direct solar radiation imposing excess heat to the closed environment, would cause substantial amount of increase in the inside temperature. It was also observed that greenhouse infrastructure and control systems, mostly imported from Australia and the Netherlands were operating using evaporative cooling systems (misting or high-pressure fog and pad-and-fan), without proper modifications for the environment where the RH is as high as 90% in some days and ambient temperature is around 38 to 41°C. The objective of this study was to employ an engineering approach to determine the efficiency of these cooling methods in TLGs of Malaysia.

MATERIALS AND METHODS

The pad and fans at the two ends of a greenhouse visited at Seremban site are shown in Fig. 1 and 2. The schematic views of misting and pad-and-fan evaporative cooling system are provided in Fig. 3a, b, respectively. Another type of evaporative cooling devices that use airflow over wet pads are swamp coolers (Fig. 4), also

Fig. 1: Evaporative cooling, view of the pad end

Fig. 2: Evaporative cooling, view of the fan end

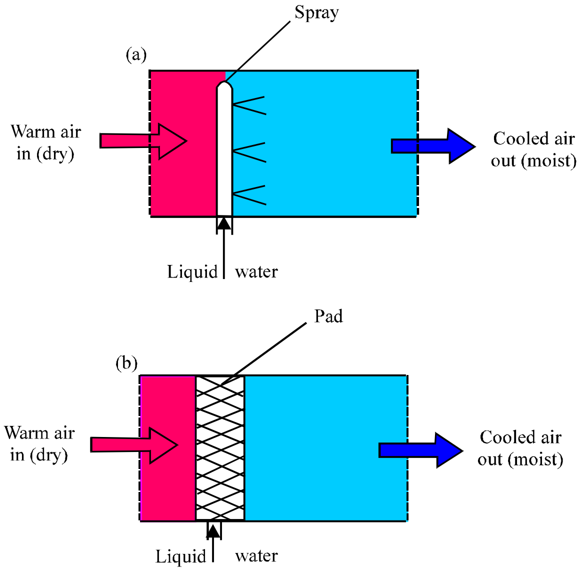

Fig. 3(a-b): Schematic diagram of evaporative cooling method, (a) Misting and (b) Pad and fan systems

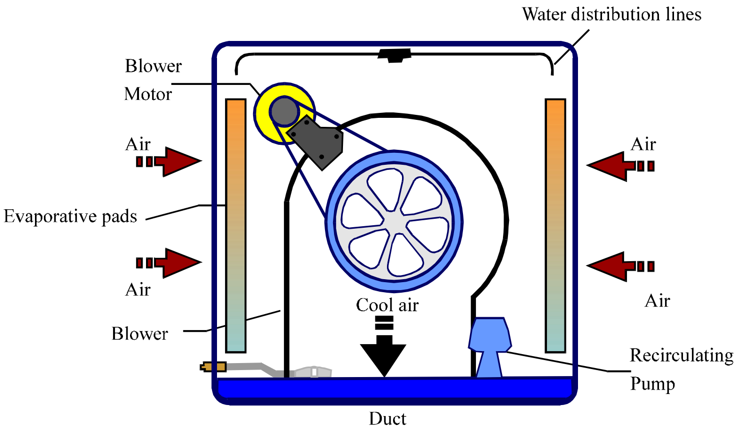

Fig. 4: Schematic front-view of a swamp cooler showing motor, pump, fan and water circulation on pads at left and right sides

referred to as direct evaporative coolers. Swamp coolers are widely utilized by greenhouse owners in lowland Malaysia as an alternative for misting and pad-and-fan. Their compactness, lower installation-operation cost and higher efficiency have made them more practical for small greenhouses. These devices simply consist of a metal box mounted outside the greenhouse with evaporative pads on its three sides. A pump supplies water to the three saturated pads over which the supply air is blown. The air stream is then cooled by evaporated water. As a result of this process, the RH of the air stream increases, causing the dry-bulb temperature to decrease. These coolers are categorized by volumetric flow rate capacity in cubic feet per minute (cfm) and are available at 1.42 m^3 sec^{-1} (3000 cfm), 2.12 m^3 sec^{-1} (4500 cfm), 3.3 m^3 sec^{-1} (7000 cfm), 4.25 m^3 sec^{-1} (9000 cfm) or higher. They can also provide ventilation because they typically condition and supply 100% outside air.

In order to investigate the performance of swamp cooling methods in TLGs of Malaysia, a control volume for a general swamp cooler was considered as shown in Fig. 5. The mass and energy conservation in Eq. 1 and 2 were solved to find the absolute humidity (ω_3) at the outlet Eq. 3.

$$\begin{gathered}\sum \dot{m}_a = \text{Constant} \\ \therefore \dot{m}_{a,1} + \dot{m}_{a,1}\omega_1 + \dot{m}_{H_2O\,(l),2} = \dot{m}_{a,3} + \dot{m}_{a,3}\omega_3\end{gathered} \tag{1}$$

$$\begin{gathered}\sum \dot{E}_{in} = \sum \dot{E}_{out} \\ \therefore \dot{m}_{a,1}h_{a,1} + \dot{m}_{a,1}\omega_1 h_{H_2O\,(v),1} + \dot{m}_{H_2O\,(l),2}\omega_1 h_{H_2O\,(l),1} + \dot{W}_{elec} \\ = \dot{m}_{a,3}h_{a,3} + \dot{m}_{a,3}\omega_3 h_{H_2O\,(v),3}\end{gathered} \tag{2}$$

$$\omega_3 = \frac{(h_{a,1} - h_{a,3}) + \omega_1 (h_{H_2O(v),1} - h_{H_2O\,(l),2})}{h_{H_2O(v),3} - h_{H_2O\,(l),2}} + \frac{W_{elec}}{\dot{m}_a (h_{H_2O\,(v),3} - h_{H_2O\,(l),2})} \tag{3}$$

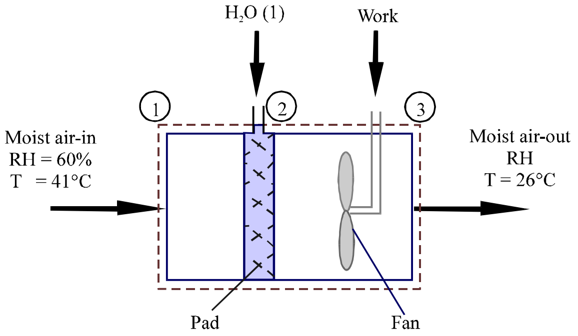

Fig. 5: Control volume diagram of a swamp cooler

Here $\dot{m}$ (kg sec^{-1}) and $\dot{E}$ [kW] are the mass and energy flow rate, respectively h [j kg^{-1}] is the enthalpy, ω [kg of water vapor/kg of dry air] is the absolute humidity and $\dot{W}$[Watt] is the power. In order to find RH of the outlet air using these equations, the enthalpy of the water vapor at station 3 was assumed to be equal to the enthalpy of saturation ($h_{H_2O(v),3} = h_{sat,3}$). Writing the mass conservation equation for water (H_2O) to find the total mass of air, $\dot{m}_a$, yields the following expression in 4:

$$\begin{gathered}\sum_{inlets} \dot{m}_{H_2O,i} = \sum_{Outlets} \dot{m}_{H_2O,i} \\ \dot{m}_{H_2O(v),1} + \dot{m}_{H_2O(l),2} = \dot{m}_{H_2O(v),3} \\ \dot{m}_{a,1}\omega_1 + \dot{m}_{H_2O(l),2} = \dot{m}_{a,3}\omega_3\end{gathered} \tag{4}$$

The available expressions for volumetric flow rate, mass flow rate over the flow area, humidity ratio and water vapor mole fraction (Reference) were used to determine the following in Eq. 5, 6 and 7, describing total mass of air, absolute humidity and RH at the outlet, respectively.

$$\dot{m}_a = \frac{\rho_1 \dot{V}_1}{1+\omega_1} \qquad (5)$$

$$\omega_3 = 0.622 \frac{P_{H_2O(v),3}}{P_3 - P_{H_2O(v),3}} \qquad (6)$$

$$\varphi_3 = \frac{P_{H_2O(v),3}}{P_{sat}(T_3)} \qquad (7)$$

Here:

$$\rho\left[\frac{kg}{m^3}\right]$$

is the specific weight:

$$\dot{v}\left[\frac{m^3}{sec}\right]$$

is the volumetric flow rate:

$$P\left[\frac{N}{m^2}\right]$$

is the pressure and φ is the relative humidity. To expand this analysis, a control volume representing of a general evaporative cooling system is shown in Fig. 6. In this scheme, air with temperature T_1 (°C) and RH of φ_1 enters a long duct and exits with temperature T_3 (°C) and RH of φ_3. Assuming ideal-gas behavior for air and water vapor, constant-pressure and an adiabatic process and in the absence of any work interactions, solving mass and energy balance equations for this system yields 8 and 9, respectively. Using the water mass conservation expression to eliminate $\dot{m}_{1,2}$ in Eq. 9, the expression in Eq. 10 is resulted after simplifying and rearranging. Assuming that the air in the outlet is cooled down to the temperature of the supplying water ($T_2 = T_3$ and $h_{1,2} = h_{1,3}$) yield the expression given in 11 which can be used to determine the enthalpy difference of the air and to approximate the enthalpy of the water vapor as expressed by 12. Substituting for $h_{v,3}$, $h_{v,1}$ in the combined mass/energy conservation expression to solve for ω_1 and by using the definition of humidity ratio (mass of water vapor per unit mass of dry air) and its relation with water vapor partial pressure, the RH of air at the inlet is determined from Eq. 14. In addition, based on the efficiency of the evaporative cooling system (η), a relationship between air temperature at the inlet and outlet is given by 15:

$$\dot{m}_{1,2} = (\omega_3 - \omega_1)\dot{m}_a \qquad (8)$$

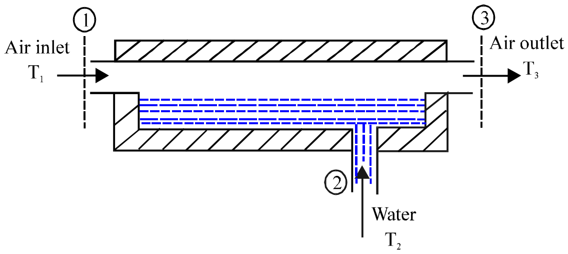

Fig. 6: Schematic view of a genera1 l evaporative cooling system, T_2 is less than T_1

$$\dot{m}_a = (h_{a,1} + \omega_1 h_{v,1}) + \dot{m}_{1,2} h_{1,2} = \dot{m}_a (h_{a,3} + \omega_3 h_{v,3}) \qquad (9)$$

$$h_{a,1} = (h_{a,1} + \omega_3 (h_{v,3} - h_{1,2}) + \omega_1 (h_{v,1} - h_{1,2}) = 0 \qquad (10)$$

$$h_{v,3} - h_{1,2} = h_{v,3} - h_{1,3} = h_{fg})(T_3) \qquad (11)$$

$$\begin{gathered} h_{a,1} - h_{a,3} = C_{p,avg}(T_1 + T_3) \\ h_{v,1} \approx h_g(T_1) \end{gathered} \qquad (12)$$

$$\omega_1 = \frac{C_{p,avg}(T_1 - T_3) + \omega_3 h_{fg}(T_3)}{h_f(T_3) - h_g(T_1)} \qquad (13)$$

$$\varphi_1 = \frac{P_{v,1}}{P_{sat}(T_1)} \qquad (14)$$

$$T_3 = T_1 - \eta(T_1 - T_w) \qquad (15)$$

RESULTS AND DISCUSSION

According to Kamaruddin *et al.* (2002), in lowland Malaysia, temperature ranges from 25 to 33°C, RH: 80 to 90%, solar radiation: 12 to 20 MJ m^2, wind speed: 2 to 22 m sec^{-1} and heavy rainfall 2032 to 2540 mm. Because of the direct solar radiation which imposes excess heat to the greenhouse closed environment, the indoor ambient temperature can become as high as 38 to 41°C. The performance of the swamp cooler in Fig. 5 was determined for TLGs of Malaysia by considering a case in which air with a volumetric flow rate of 1.416 m^3 sec^{-1} and RH of 60% enters a swamp cooler at 41°C, with a fan driven by a 1/8 hp (93.2 W) electric motor. In order to reduce air temperature by 15°C to reach a desired temperature of 26°C at the outlet, according to the Eq. 7, the RH of the outlet air will theoretically increase up to 170%. For the same change in air temperature with 25 and 16% RH at the inlet, the RH of the outlet air becomes 90 and 65%, respectively. The properties of air, water vapor and liquid water and other thermodynamics values for this particular

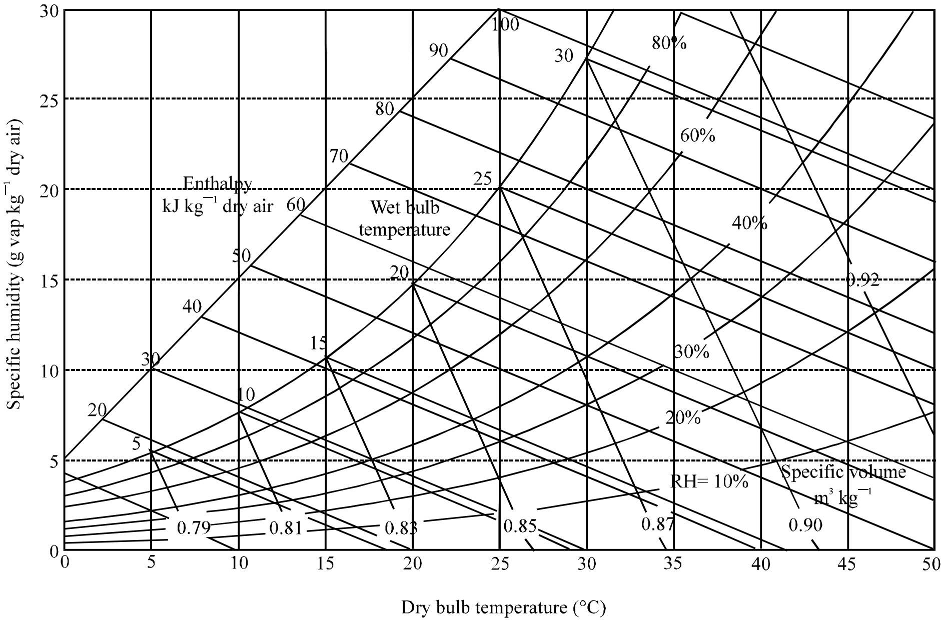

Fig. 7: A simplified psychometric chart at 1 atm total pressure

example are provided in Table 1. According to the temperature and relative humidity values (T_1, T_2, T_3, φ_1 and φ_3) in Table 1, removal of energy from air lowers the temperature inside the greenhouse; however, this does not happen when RH is higher than a certain point. From the psychrometric charts in (Fig. 7), the wet bulb line represents the removal of sensible heat by adding moisture to the air at a constant temperature measured by a thermometer whose bulb is covered by a wetted wick, also known as Wet Bulb Temperature (WBT). This is the process of evaporative cooling, which increases the absolute humidity, the RH and decreases the air temperature. Assuming 80% efficiency for the evaporative cooling system in Fig. 6, the temperature and RH at the outlet (station 3) were calculated for air temperature of 30 and 35°C and RH of range 20 to 80% at the inlet (station 1). Results of these calculations are provided in Table 2. According to Fig. 6, with evaporative cooling, the air temperature can be decreased to the temperature of the supplying water, or $T_2 = T_3$ and $h_{l,\,2} = h_{f,\,3}$. In addition, the calculations in 8 to 15 imply that the amount of cooling that an evaporative system can achieve, highly depends on the water content of the outside air. For example, according to the thermodynamic values provided in Table 3, if air temperature is to be decreased from 25 to 20°C with 100% RH at the outlet, the RH of air at the inlet (station 1, Fig. 6) should be 63.5%. These results clearly show that the higher the RH of air at inlet of an evaporative cooling system, the less potential it has to

Table 1: Thermodynamic table for swamp cooler in Fig. 5

Property	Value	Units
T_1	41	°C
T_2	26	°C
T_3	26	°C
φ_1	60	%
P_{sat}(26°C)	7.726299	kPa
$P_{H_2O(v),1}$ (26°C)	4.6851	kPa
$P_{H_2O(v),2}$	100	kPa
$P_{H_2O(v),3}$	5.6671	kPa
$h_{H_2O(v),1}$	2577.5	kj kg^{-1}
$h_{H_2O(v),2}$	107.22	kj kg^{-1}
$h_{H_2O(v),3}$	2547.5	kj kg^{-1}
$h_{a,1}$	440.34	kj kg^{-1}
$h_{a,3}$	424.73	kj kg^{-1}
$C_{p,a,1}$	1.0072	kj kg-K^{-1}
$C_{p,a,3}$	1.007	kj kg-K^{-1}
$X_{H_2O(v)}$	0.046851	
M_a	28.4567	kg kmol^{-1}
ρ_1	1.0892	kg m^{-3}
$\dot{m}_a$	1.4965	kg sec^{-1}
$\dot{m}_{H_2O(l),2}$	0.01017	kg sec^{-1}
ω_1	0.03057	
ω_3	0.037367	
φ_3	173	%

Table 2: Performance of evaporative cooling system

Temperature (°C)			RH (%)	
T_1	T_w	T_3	φ_1	φ_3
30	16	18	20	75
30	19	21	35	82
30	22	24	50	87
30	25	26	65	92
30	27	28	80	93
35	19	22	20	75
35	23	25	35	82
35	26	28	50	87
35	29	30	65	92
35	32	32	80	93

Table 3: Thermodynamic table for evaporative cooling system shown in Fig. 6

Property	Values	Units
T_1	25	°C
T_2	20	°C
T_3	20	°C
$C_{p,a}$ (25.5°C)	1.0065	$kj\ kg\text{-}K^{-1}$
h_g (20°C)	2537.4	$kj\ kg^{-1}$
h_f (20°C)	83.914	$kj\ kg^{-1}$
P_{sat} (20°C)	2.3393	kPa
h_g (25°C)	2546.5	$kj\ kg^{-1}$
h_{fg} (20°C)	2453.5	$kj\ kg^{-1}$
$P_V (T_3) = P_{sat} (T_3)$	2.3393	kPa
$P_{v,1}$	2.0122	kPa
ω_3	0.014	
ω_1	0.012603	
φ_3	100	%
φ_1	63.5	%

reduce its temperature. Therefore, as a conclusion, these systems are most effective in regions with RH less than 65%.

CONCLUSION

Evaporative cooling in the form of misting, pad-and-fan and swamp cooling are currently utilized in major commercial CEPPS in Malaysia. The performance of each system was investigated, it was shown that evaporative methods, either by means of pad-and-fan, misting or swamp cooling, is not an efficient method for making a significant drop in air temperature as required at certain hours of greenhouse operation when maximum cooling load is required. Perhaps the best way to improve the greenhouse performances is to use the information gathered from the environment and the crop itself. A good understanding of the environmental such as long-term temperature and RH trends is then necessary in order to design any system capable of working in greenhouse conditions. It can be concluded that high temperature, RH and their interaction with crop growth microenvironment beside inappropriate deployment of resource management are the major factors that reduce the production potential of lowland greenhouses in Malaysia. Improving control systems for greenhouse production systems that are suitable for Malaysian environment can potentially increase the number of greenhouse production systems in Malaysia, thus increasing the production of locally grown fruits and vegetables.

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