Thermal Model Development of a Biomass Regenerated Desiccant Supported Greenhouse Cooling for Orchid Cultivation

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Abstract. In this paper, a novel scheme of biomass regenerated desiccant supported greenhouse cooling with distributed fan-pad evaporative system has been proposed. The system aims to provide suitable thermal condition inside the greenhouse for cultivation of high value flowers like varieties of Orchid which require a temperature between 21 °C to 26 °C and humidity from 50 to 70% for the sub-tropical climate prevailing in the plains of India. In the proposed system two stage desiccant based cooling is used to obtain a low humidity ratio. A biomass based heating system is coupled for regeneration of the desiccant. Thermal modeling of desiccant wheel and greenhouse are done to estimate air temperature inside the greenhouse. A comparative study has been made based on estimated greenhouse air temperature with the results of the reference model available in the literature. From the performance analysis of the proposed system, it is revealed that the maximum temperature of the greenhouse air can be restricted within 25 °C and 25.8 °C for a hot and humid day prevailing in the month of July and September respectively. However the system can maintain a lower temperature of 19 °C during the morning time which is more conducive for target plantation (Orchid) in the given region.

1. Introduction
Greenhouse technology is a method of cultivation of plants under a suitable environment for their optimum growth. It is used to protect the plants from the adverse climatic conditions and promotes off seasonal cultivation. The plains of sub tropical countries like India receive abundant solar radiation for major part of a calendar year resulting in high ambient temperature coupled with high humidity levels. Most of the time of a calendar year, the values of ambient temperature and humidity highly exceed the prescribed ranges for Orchid (Orchidaceaeous) cultivation in the open field. The cultivation of high value target Orchids require optimum temperature ranges between 21 °C to 26.5 °C and relative humidity ranges between 50% - 70% [1]. The greenhouse cultivation of Orchids can be a potential solution to this problem. Thus, the maintenance of proper inside temperature and humidity are major challenges behind the greenhouse cultivation of the Orchids.

The desiccant wheel based air conditioning system for greenhouse cultivation is a promising alternative to maintain low humidity and low temperature as per requirement. Desiccant wheels driven by waste heat or solar thermal heat [2]. From the literature, it is observed that few works are available for greenhouse cooling using desiccant. There liquid desiccants were used and the regeneration was done using solar energy.
Till now, no work has been reported on two stage composite desiccant based greenhouse cooling, where biomass based heating system is used for regeneration of the desiccant. This study aims to propose a scheme of a biomass regenerated two stage desiccant cooling system for high growing value Orchid cultivation in a subtropical greenhouse. In this paper, a thermal model has been developed and using the thermal model the temperature and humidity inside the greenhouse have been estimated.

2. Description of the proposed system
Figure 1 shows the scheme of the proposed biomass regenerated desiccant supported greenhouse cooling system for Orchid cultivation. It consists of a free standing even span greenhouse oriented lengthwise in the east-west direction. The process air in this system is obtained from the atmosphere, which passes through desiccant wheel 1 (DW₁) made of synthesized metal silicate. After dehumidification, the air is passed through heat exchanger 1 (HE₁) for sensible cooling.

![Figure 1. Scheme of a biomass regenerated desiccant supported greenhouse cooling for Orchid cultivation in subtropical region.](image)

It may be noted the moisture adsorption capacity of the desiccant wheel decreases with increase in temperature. Considering this aspect the air at the outlet of HE₁ is again passed through desiccant wheel 2 (DW₂) and heat exchanger 2 (HE₂) in series. After passing through the second heat exchanger (HE₂), the dehumidified process air goes to the cooling pads located in the both longitudinal sidewalls of the greenhouse, where the process air is cooled due to evaporative cooling and finally supplied into the greenhouse. A circulating water pump (CWP) is used for circulation of water from the Ground
well to the pad to keep the same wet. The induced draft (ID) fans are mounted vertically at the top of the greenhouse, the arrangement is known as distributed fan pad cooling system [3]. A portion of the return air is mixed with the ambient air in an air mixing plenum and the same is allowed to pass through a direct evaporative cooler (DEC) to obtain further lower temperature. Then the same is equally divided into two groups operating in parallel, one flowing through HE2, air heater and DW 2, while the other passing through HE1, air heater and DW1. In the air heater the air is heated to the regeneration temperature of the desiccant wheel using a biomass based heating system as shown in Fig.1.

Hot water from biomass boiler is supplied to the Heater 1 and then it flows through the Heater 2. The water coming from Heater 2 is stored into a storage tank from where the water is supplied to the boiler using a feed water pump. Greenhouse crop residues (GCR) and wood pellets are used as the biomass fuel. The biomass based fuels from the biomass storage are fed to the boiler through a screw conveyor. The heat energy produced in the biomass combustion is transferred to the water to produce steam.

3. Thermal model development

3.1. Desiccant wheel

Relative humidity and enthalpy of the process air coming from the desiccant wheel can be obtained from equations (1) and (2), respectively [4].

\[ \phi_{p, out} = \phi_{p, in} - \eta_{\phi} \left( \phi_{r, in} - \phi_{r, out} \right) \]  
(1)

\[ h_{p, out} = h_{p, in} - \eta_{h} \left( h_{r, in} - h_{r, out} \right) \]  
(2)

where, \( \eta_{\phi} \) and \( \eta_{h} \) are the humidity and enthalpy effectiveness respectively. In Eq.s (1) and (2), \( h_{p, in} \) and \( \phi_{p, in} \) are enthalpy and relative humidity of the inlet process air respectively. Therefore, humidity ratio (\( X_{p, out} \)), temperature (\( T_{p, out} \)) and water vapour saturation pressure (\( P_{p, out, vsat} \)) of the outlet process air are obtained using Eq.s (3) through (5).

\[ X_{p, out} = \frac{\phi_{p, out}}{\phi_{p, out, vsat} - \phi_{p, out}} \]  
(3)

\[ T_{p, out} = \frac{h_{p, out} - \mu X_{p, out}}{C_{p, da} X_{p, out} + C_{p, wv}} \]  
(4)

\[ P_{p, out, vsat} = \exp \left( 23.196 \times \frac{3816}{T_{p, out} + 273.15} - 46.13 \right) \]  
(5)

In Eq. (4), \( \mu \) is latent heat of vaporization of water in kJ kg\(^{-1}\). \( C_{p, da} \) and \( C_{p, wv} \) are specific heat of dry air and water vapour. (\( \eta_{h} \)) is obtained from equation (6) with specific correlation coefficients In Eq.s (7) through (11).

\[ \eta_{\phi} = \alpha_{\phi, X} \times \alpha_{\phi, v} \times \alpha_{T} \times \alpha_{X} \times \alpha_{N} \]  
(6)

\[ \alpha_{\phi, r} = C_{1} \phi_{r, in} + C_{2} \phi_{r, in} + C_{3} \]  
(7)

\[ \alpha_{\phi, p} = C_{4} \phi_{p, in} + C_{5} \phi_{p, in} + C_{6} \]  
(8)

\[ \alpha_{T} = C_{7} \ln \left( T_{r, in} - T_{p, in} \right) + C_{8} \]  
(9)

\[ \alpha_{X} = C_{9} X_{p, in} - C_{10} X_{r, in} + 1 \]  
(10)

\[ \alpha_{N} = C_{11} N + C_{12} \]  
(11)
Similarly, \( \eta_h \) can be calculated using equation (12) with respective correlation coefficients in Eq.s (13) through (17).

\[
\eta_h = \beta_v \times \beta_v \times \beta_T \times \beta_X \times \beta_N \tag{12}
\]

\[
\beta_v = K_1 (v_{r,in})^{K_2} \tag{13}
\]

\[
\beta_v = K_3 (v_{p,in})^{K_4} \tag{14}
\]

\[
\beta_T = K_5 T_{r,in} + K_6 T_{p,in} + K_7 \tag{15}
\]

\[
\beta_X = K_8 X_{r,in} + K_9 X_{p,in} + 1 \tag{16}
\]

\[
\beta_N = K_{10} N + K_{11} \tag{17}
\]

where \( v_{p,in} \) and \( v_{r,in} \) are process air and regeneration air velocity (m \( s^{-1} \)) at inlet of the desiccant wheel respectively. \( T_{r,in} \) and \( N \) are regeneration air temperature (\( ^\circ C \)) and revolution speed (rev h\(^{-1} \)). \( C_1 \) through \( C_{12} \) and \( K_1 \) through \( K_{11} \) are the correlation constant are taken from the work of [5].

### 3.2. Thermal model of greenhouse

To develop the thermal model of the greenhouse, it has been assumed that heat flux of the soil is negligible and the transmissivity of the greenhouse cover material is constant. The heat transfer due to radiation from the crop is negligible and the absorptivity of the structural material is negligible. Then the steady state energy balance equation of the greenhouse can be expressed as [3]:

\[
(1 - S)\alpha G_0 \tau + U_g \Delta T = K_s (T_{amb} - T_{pad}) - K_s \Delta T + \lambda E \tag{18}
\]

In Eq. (18), \( S, \alpha, G_0, \tau, U_g, K_s, T_{amb}, T_{pad}, \lambda \) and \( E \) are shading factor, absorbance of the canopy, global solar radiation intensity (kJ \( m^{-2} \)), roof transmittance, total heat transfer coefficient (W \( m^{-2} K^{-1} \)) of the greenhouse covering material, sensible heat transfer coefficient (W \( m^{-2} K^{-1} \)) of air, ambient temperature, temperature of the cooling pad end, latent heat of vaporization (kJ kg\(^{-1} \)) and crop canopy transpiration (kg \( m^{-2} s^{-1} \)) respectively. Where, \( \Delta T \) denotes the temperature difference between the atmospheric air and greenhouse air.

In Eq. (18), the heat transfer coefficient (\( K_s \)) due to sensible load inside the greenhouse can be given by [6]:

\[
K_s = \rho_a C_{pa} (ACM) V_g / (60 \cdot A_{gf}) \tag{19}
\]

The rate of plant transpiration rate in greenhouse can also be calculated from the Penman-Monteith formula as given below [7]:

\[
\lambda E = \frac{\delta (\lambda E + H) + \frac{2I_a \rho_a C_{pa} W_g}{r_s}}{\delta + \gamma \left(1 + \frac{r_s}{r_a}\right)} \tag{20}
\]

where, \( I_a \) indicates area index of leaf for the given plantation, while \( \gamma \) indicates a psychrometric constant (Pa \( K^{-1} \)). In Eq. (20), the stomatal resistance of the crop (\( r_s \)) can be written as.

\[
r_s = 200 \left[1 + 1/ \exp \left\{0.05 \left(S_g - 50\right)\right\}\right] \tag{21}
\]

In Eq. (20), the resistance due to aerodynamic effect, \( r_a \) (s \( m^{-1} \)) can be written as.
where, \( L_c \) is the characteristic length of the leaf and \( u_i \) denotes the average air velocity inside the greenhouse. In Eq. (20), sensible heat load (H) can be expressed as:

\[
H = K_s \left( T_{amb} - T_{pad} \right) - K_s \Delta T
\]  

(23)

In Eq. (20), \( \delta \) denotes the slope of the water vapour saturation curve in (Pa K\(^{-1}\)) at any temperature \( T \) (K) and may be expressed as [6].

\[
\delta = \left( \frac{5385}{T} \right) \times 2.229 \times 10^{11} \exp \left( \frac{-5385}{T} \right)
\]  

(24)

In Eq. (20), \( W_g \) indicates water vapour pressure deficit and can be expressed as.

\[
W_g = \delta \cdot \Delta T - \Delta e + W_a
\]  

(25)

In Eq. (25), \( \Delta e \) denotes the water vapour pressure difference (Pa) between the ambient air and greenhouse air. \( W_a \) denotes the water vapour pressure deficit (Pa) of the ambient air [6]. The expression for the greenhouse plant transpiration rate can be written as [8].

\[
\lambda E = K_L \cdot \Delta e
\]  

(26)

In Eq. (26), \( K_L \) denotes the latent heat transfer coefficient (W m\(^{-2}\) Pa\(^{-1}\)) and may be written as.

\[
K_L = \lambda \cdot \zeta \cdot \rho_a \cdot \left( ACM \right) \cdot V_g / \left( 60 \cdot A_g \right)
\]  

(27)

In Eq. (27), \( \zeta \) is a conversion factor for the water vapour present in the air at standard temperature and pressure. In this present model the value of \( \zeta \) is considered to be 6.25 \times 10^{-6} \text{ kgw kg}_{\text{a}}^{-1} \text{ Pa}^{-1} \) [8]. Now combining the above following equations (20), (23), (25) and (26), we get.

\[
\Delta T = \left[ \frac{\left( \delta \cdot P_1 + P_2 \frac{W_a}{r_a} \right)}{P_3} - \left( S_g - C \right) \cdot \left( 1 - \frac{\delta \cdot P_2}{P_3} \right) \right] - \left[ \left( U_g + K_1 \right) \cdot \left( 1 - \frac{\delta \cdot P_2}{P_3} \right) \right]
\]  

(28)

In Eq. (28), \( P_1, P_2 \) and \( P_3 \) represent the constants, which can be written further as:

\[
P_1 = K_s \left( T_{amb} - T_{pad} \right)
\]  

(29)

\[
P_2 = 2 \cdot I_{rad} \cdot \rho_a \cdot C_{pa}
\]  

(30)

\[
P_3 = \delta + \gamma \cdot \left( 1 + r_s / r_a \right)
\]  

(31)

Therefore, the average greenhouse temperature can be written as.

\[
T_{gh} = T_{amb} - \Delta T
\]  

(32)
The conditioned air that is supplied to the greenhouse is to be cooled in a distributed fan-pad cooling system where evaporative cooling is taken place. The expression of the air temperature at the end of the cooling pad can be given by

\[ T_{pad} = T_{he2} - \varepsilon_{cp} \cdot (T_{he2} - T_{wb,he2}) \]  
(33)

In Eq. (33), \( \varepsilon_{cp} \) and \( T_{wb,he2} \) are the effectiveness of the cooling pads and wet bulb temperature of the air coming from the second heat exchanger (HE2) respectively. The heat exchanger is modelled based on the work of Asadi and Roshanzadeh [5]. A computer code in Engineering Equation solver (EES) has been written to solve the above equations. The input parameters to the computer code are listed in Table 1. The initial temperature of the greenhouse is guessed as ambient temperature for the first iteration. After each successive iteration, the same is shifted to its previous value. In this study the greenhouse is assumed to be located in the plains of Gangetic Bengal (Kolkata) in the Indian subcontinent. Thus, the weather data for Kolkata as available from the Indian Meteorological Department, New Delhi [9] have been used for the simulation.

**Table 1.** Value of the input parameters to the thermal model.

| Variable                      | Value       | Variable                      | Value       |
|-------------------------------|-------------|-------------------------------|-------------|
| Absorptivity (\( \alpha \))   | 0.7 [10]    | Effectiveness of the cooling pad (\( \varepsilon_{cp} \)) | 0.88 [6]    |
| Roof area (\( A_{gf} \))      | 240 m\(^2\) [11] | Effectiveness of the heat exchange (\( \varepsilon_h \)) | 0.8 [5]     |
| Greenhouse volume (\( V_g \))  | 728 m\(^3\) [11] | Diameter of the desiccant wheel (\( D \)) | 60 cm       |
| Shad factor (\( S \))         | 0.5 [3]     | Axial length of the wheel (\( L \)) | 20 cm       |
| Transmissivity (\( \tau \))   | 0.8 [10]    | Aspect ratio (\( a/b \)) | 0.51       |
| Overall heat transfer coefficient (\( U_g \)) | 4.5 W m\(^{-2}\) | Speed of the wheel (\( N \)) | 5-30 rev h\(^{-1}\) |
| Psychometric constant (\( \gamma \)) | 72.08 Pa K\(^{-1}\) [6] | Regeneration air temperature at inlet (\( T_{r,in} \)) | \( 45\, ^\circ\text{C}-78\, ^\circ\text{C} \) [4] |
| Leaf area index of lettuce (\( I_{la} \)) | 0.5 m\(^2\) m\(^{-2}\) | Process air temperature at inlet (\( T_{p,in} \)) | \( 17\, ^\circ\text{C}-35\, ^\circ\text{C} \) [4] |
| Latent heat of vaporization of water (\( \lambda \)) | 2,260,000 J kg\(^{-1}\) | Velocity of the inlet regeneration air (\( v_{r,in} \)) | 1.75-2.86 m s\(^{-1}\) [4] |
|                             |             | Velocity of the inlet process air (\( v_{p,in} \)) | 1.81-2.5 m s\(^{-1}\) [4] |

### 4. Results and discussions

#### 4.1. Model validation for desiccant wheel

The validation of present thermal model of desiccant wheel is carried out by comparing the numerical results estimated by our model with the experimental results reported by the reference work [4]. Figure 2 shows the comparison of numerical results and experimental results on specific humidity and temperature of the process air outlet from the desiccant wheel, for different inlet conditions. From the Fig. 2 (a) and (b), it is observed that the present model agrees well with the reference model, having average absolute error of 4.3% and 4.5 % respectively.

#### 4.2. Performance analysis

Figure 3 shows the difference of atmospheric temperature and average greenhouse air temperature for a typical day of monsoon season (24\(^{th}\) July, 2009). The ambient air temperature and relative humidity both remain high during the month of July. The results of the present model have been compared with the reported work of Banik and Ganguly [3] and Kittas et al. [12] respectively. To maintain the suitable air temperature inside the greenhouse, a traditional fan-pad cooling system is suggested by...
Kittas et al., while, liquid desiccant based distributed fan-pad evaporative cooling system is proposed by Banik and Ganguly[10].

Figure 2. Numerical and experimental values of (a) specific humidity and (b) temperature of the process air at the outlet from the desiccant wheel for different inlet conditions.

From the figure 3 (a), it is evident that our proposed system (two stage composite desiccant based distributed fan-pad evaporative cooling) can maintain the greenhouse air temperature within 24 °C even during the peak sunshine hour (12 Noon), while during the same period the temperature of other two considered models, T Kittas and T Banik and Ganguly can maintain a temperature at 28.4 °C and 27 °C respectively.

Figure 3. Difference of atmospheric temperature and average greenhouse air temperature for a typical day of (a) July and (b) September.

In the present model for a typical day July, the maximum greenhouse air temperature reaches at 25 °C, which is 7.9 °C less than the corresponding ambient temperature. From the figure, it is also seen that during the morning time (6:00 to 8:00 am), the system can maintain a temperature about 19 °C due to efficient desiccation and recirculation of the process air.
Figure 3 (b) epitomizes the difference of atmospheric air and greenhouse air temperature for a typical day in the autumn season (12th September, 2009) in Kolkata. It is observed from Fig. 3 (b) that the maximum greenhouse air temperature can be maintained within 25.8°C, while the same reaches to about 28°C with a traditional fan-pad cooling system and 26.1°C with that of distributed fan-pad ventilated system with liquid desiccation respectively during the peak sunshine hours. It is also noticed that during the same period, the present system can reduce the greenhouse air temperature by about 10°C below that of ambient (35.8°C), while the other considered two systems can reduce the same by only about 7.8°C and 9.5°C respectively. During the morning and evening time, present system can very well maintain the required suitable low temperature conducive for the growing of the considered plantation.

5. Conclusions
In this present study, a scheme of a biomass regenerated composite desiccant supported greenhouse cooling for Orchid cultivation is presented. A thermal model of the greenhouse and its desiccation system is presented and validated against available reference study in the literature. The performances of the proposed system have been analyzed for a hot and humid day in the month of July and September respectively. From the study, it is observed that the proposed model can conserve the greenhouse air temperature between 19°C to 25.8°C by using two stage desiccants cooling with distributed fan-pad evaporative system. From the study, it is also revealed that proposed model can reduce the specific humidity of the ambient air by a maximum value of 11 g kg⁻¹. The system can very well maintain the desired climate in terms of temperature and humidity ratio inside the greenhouse for the year round cultivation of the target flora. The study thus reinforces the need and viability of such system for the sub-tropical countries like India.

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