Thermal Modeling and Economical Analysis of a Solar Desiccant Assisted Distributed Fan-Pad Ventilated Greenhouse

P. Banik, A. Ganguly

***Abstract*— In the present work, a novel configuration of distributed fan-pad evaporative cooling along with solar desiccation has been proposed for agricultural greenhouse application. In the proposed configuration solid desiccation along with solar regeneration has been used to dehumidify the air prior to the evaporative cooling pads to enhance its performance even under high ambient humidity condition. A thermal model of the proposed system has been developed, taking into consideration the transpiration phenomenon of target flora along with sensible heating of the greenhouse air. A computer code in 'C' language has been developed for the proposed model. The analysis has been done for the climatic condition of Kolkata, a place that witness a high temperature coupled with high humidity level for a considerable part of a year, especially during the monsoon. The study reveals that a signification amount of performance enhancement in terms of temperature reduction can be achieved using the proposed system over the distributed fan-pad ventilated system (without desiccant) throughout the year. Finally, an economical analysis has also been made to examine the economical feasibility of the proposed system for cultivating some target flora like Gerbera.**

***Index Terms*—Desiccant evaporative cooling, distributed fan-pad ventilation, greenhouse, economical analysis.**

I. INTRODUCTION

GREENHOUSE technology promotes cultivation of target plantation under a controlled environment. The greenhouse technology had emanated in the western countries of the world primarily to protect the plants from the extreme cold and frost. But in the plains of India, it is being practiced mainly to protect the plants from excessive heat from solar radiation, high temperature and humidity. So, the main objective of a greenhouse located in the plains of Indian subcontinent is the reduction of the inside temperature. The cooling can be achieved by using fan-pad evaporative cooling system. A conventional fan-pad ventilated greenhouse comprise of the induced draught fan(s) and cooling pads mounted on the opposite end walls while in a distributed fan-pad system, the fan(s) are installed at the top and the pads are mounted on the side walls of the greenhouse. This distributed fan-pad configuration eliminates the problem of temperature gradient along the length of the greenhouse that is generally encountered in a conventional fan-pad ventilated system. But, it may be noted that the performance of all types of evaporative cooling systems depend heavily on the ambient humidity level. They fail to provide satisfactory results at high humidity levels. To enhance the cooling performance even under high ambient humidity condition, desiccant materials can be employed. The desiccant materials adsorb the moisture from the air and thus, the air gets dehumidified before its entry to the evaporative pads.

The objective of the present work is to develop a thermal model of a solar desiccant supported distributed fan-pad evaporative cooling system for greenhouse cultivation of Gerbera flower. Gerbera can be best grown at temperatures ranging between 23-27 ºC [1], which is difficult to be maintained in the open field especially during the hours of peak sunshine in summer season. Gerbera flower has wide demand in the domestic and international market. So, finally an economical analysis has been done to evaluate the payback period of the proposed system.

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II. DESCRIPTION OF THE PROPOSED SYSTEM

The schematic diagram of the proposed system is shown in the Fig.1. The system comprises of a ridge ventilated greenhouse (10m x 4m x 3.5m), oriented in east-west direction. Cellulose cooling pads are installed in the north and south walls of the greenhouse. The induced draught fans are mounted on the canopy of the greenhouse. This type of arrangement is called distributed fan-pad system as available in the literature [2]. In the present work, to further enhance the performance of the greenhouse under high ambient humidity condition, a desiccant wheel (made of silica-gel) and a heat exchanger are installed prior to the cooling pads at both north and south walls of the greenhouse. When the air flows through the desiccant wheel, it gets dehumidified, but its dry bulb temperature increases due to the heat of adsorption. Thus, to reduce its temperature, the air is passed through the heat exchanger, where water is used as the coolant. After passing through the heat exchanger, the cold and dehumidified air passes through the cooling pads, where further temperature reduction occurs due to evaporative cooling. The conditioned air then enters the greenhouse, where it absorbs the heat and gets vented out from the top under suction from the induced draught fans.

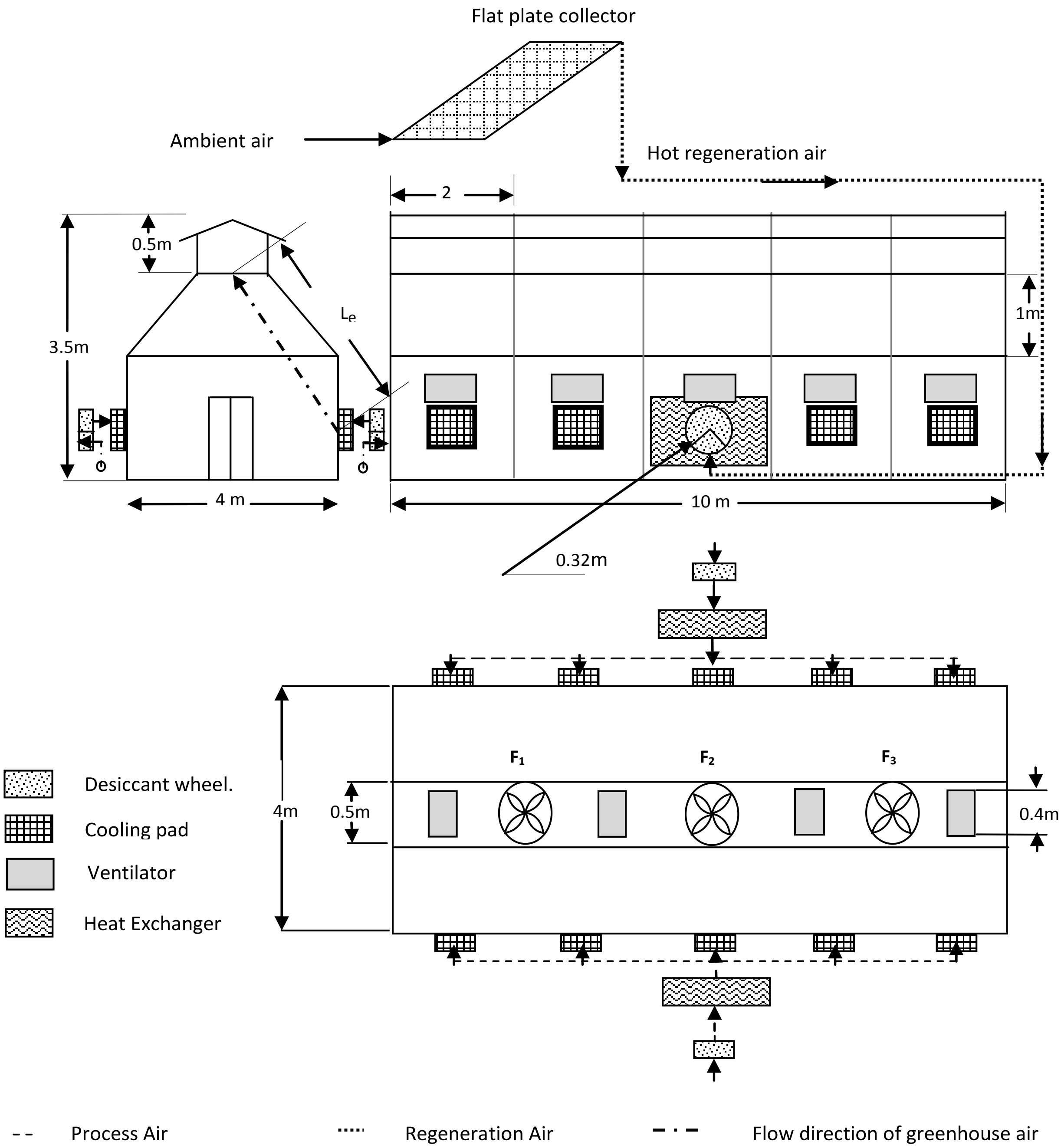

Fig. 1: Schematic diagram of the proposed system

For the regeneration of the desiccant materials, hot air derived from the solar flat plate collector(s) is allowed to pass through a portion of the desiccant wheel in the opposite direction to that of the process air.

The dehumidification process is mainly required during the summer, monsoon and autumn season, when the temperature and humidity of the ambient air remains moderate to high for a considerable period of time of the day. But during the winter season, both the temperature and relative humidity remains low.

Therefore, during the winter period (December, January and February), instead of passing the ambient air through the desiccant wheels, it is directly fed to the evaporative pads.

III. Model Development

Following assumptions have been made while developing the thermal model:

i. The modeling has been done under steady state condition.
ii. The chemical dehumidification in the desiccant wheel is assumed to be adiabatic in nature.
iii. Water temperature in the heat exchanger is assumed to be equal to the wet bulb temperature corresponding to the atmospheric condition.
iv. No energy is absorbed by the structural elements of the proposed system.
v. The effects of pressure drop in the heat exchanger have been neglected.

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The greenhouse can be considered to be a solar heat collector or storage system, where both sensible and latent heat exchange takes place. The energy balance equation for the greenhouse considering the transpiration phenomenon of the target plantation can be given by:

$$\begin{aligned} S_{gi} + U.\Delta T &= K_s.(T_g - T_{pad}) + \lambda.E \\ &= Q - K_s.\Delta T + \lambda.E \end{aligned} \tag{1}$$

Where, $Q = K_s.(T_a - T_{pad})$

In Eq. (1), the term 'S_{gi}' denotes the part of the incident solar radiation that gets transmitted inside the greenhouse, while 'U' denotes the overall heat transfer coefficient of the greenhouse cladding material, whose value is considered to be 4.5 W/m^2-K [3]. The term 'ΔT' represents the temperature difference between the ambient and greenhouse inside air. The term 'T_g' is the mean greenhouse temperature, which is considered to be the average of the pad and the fan end temperature [2], while 'λ' represents the latent heat of vaporization of water, whose value is assumed to be 2.45×10^6 [4]. The term 'E' denotes the canopy transpiration (kg m^{-2} s^{-1}).

The sensible heat transfer coefficient of the greenhouse air can be given by [4]:

$$K_s = \rho_a.C_{pa}.(ACM).V_g / (60.A_g) \tag{2}$$

Where, 'ACM' and 'V_g' denote the number of air changes per minute and total volume of the greenhouse respectively. In the present work, the value of 'ACM' is considered to be 1.2 [2]. The term 'A_g' represents the floor area of the greenhouse.

The greenhouse crop transpiration rate can also be deduced from the Penman-Monteith formula as given below [5]:

$$\lambda.E = \left[\delta.(\lambda.E + H) + 2.I_{la}.\rho_a.C_{pa}.D_i / r_a\right] / \left[\delta + \gamma.(1 + r_s / r_a)\right] \tag{3}$$

In Eq. (3), 'I_{la}' denotes the leaf area index of the given plant species. In the present work the value of 'I_{la}' is assumed to be 3.5 [4], considering the plant to be Gerbera. The term 'γ' denotes the psychrometric constant. In Eq. (3), the term 'H' represents the sensible heat load of the greenhouse air, which can be represented as:

$$H = Q - K_s.\Delta T \tag{4}$$

The slope of the water vapour saturation curve at a temperature, T (K) can be expressed as [4]:

$$\delta = \left(5385 / T^2\right) \times 2.229 \times 10^{11}.\exp(-5385/T) \tag{5}$$

The water vapour pressure deficit of the greenhouse inside air can be given by [4]:

$$D_i = \delta.\Delta T - \Delta e + D_o \tag{6}$$

In Eq. (6), 'Δe' represents the water vapour pressure difference between the inside and the ambient air. The water vapour pressure deficit of the ambient air can be given by the following relation [4]:

$$D_o = e^*(T_a) - e_o \tag{7}$$

Where, the terms '$e^*(T_a)$' and 'e_o' denote the saturated water vapour pressure and actual water vapour pressure of the ambient air respectively.

The saturated water vapour pressure of the air at a temperature, T (K) can be given by [4]:

$$e^*(T) = \exp(25.317 - 5144/T) \tag{8}$$

The stomatal resistance of the greenhouse plants can be given by the expression [4]:

$$r_s = 200.\left[1 + 1/\exp\{0.05(S_{gi} - 50)\}\right] \tag{9}$$

The aerodynamic resistance, r_a depends on the geometry of the leaves as well as on the average speed of inside air, which can be expressed as [5]:

$$r_a = 220.\left(d^{0.2} / V_i^{0.8}\right) \tag{10}$$

In Eq. (10), 'V_i' denotes the average air speed inside the greenhouse, which has been calculated considering the greenhouse to be a parallelepiped, while 'd' represents the characteristic length of the leaf, whose value is assumed to be 0.025 m [4].

The greenhouse crop transpiration can also be given by [4]:

$$\lambda.E = K_L.\Delta e \tag{11}$$

Where, the term 'K_L' represents the latent heat transfer coefficient, which can be given by [4]:

$$K_L = \lambda.\zeta.\rho_a.(ACM).V_g / (60.A_g) \tag{12}$$

In Eq. (12), the term 'ζ' denotes the conversion factor between air moisture content at standard temperature and the air water vapour pressure. The value of 'ζ' has been considered to be 6.25×10^{-6} kg_w kg_a^{-1} Pa^{-1} [4].

Combining Eq. (6) and (11), we get:

$$D_i = \delta.\Delta T - \lambda.E / K_L + D_o \tag{13}$$

Again combining the Eq. (3), (4) and (13) we get:

$$\begin{aligned} \lambda.E = \{&\delta.\lambda.E + \delta.Q - \delta.K_s.\Delta T \\ &+ P_1.(\delta.\Delta T - \lambda.E / K_L + D_o)\} / P_2 \end{aligned} \tag{14}$$

Considering,

$P_1 = 2.I_{la}.\rho_a.C_{pa}$ and, $P_2 = \delta + \gamma.(1 + r_s / r_a)$

The term '$\lambda.E$' can be removed from the Eq. (14) by combining Eq. (1) with Eq. (14). Therefore, after simplification, the equation for 'ΔT' can be obtained as:

$$\Delta T = \frac{\left[\dfrac{\left(\delta.Q + P_1.\dfrac{D_o}{r_a}\right)}{P_2} - (S_{gi} - Q)\left\{1 - \dfrac{\left(\delta - \dfrac{P_1}{K_L.r_a}\right)}{P_2}\right\}\right]}{\left[(U + K_s)\left\{1 - \dfrac{\left(\delta - \dfrac{P_1}{K_L.r_a}\right)}{P_2}\right\} - \dfrac{\left(\dfrac{P_1.\delta}{r_a} - \delta.K_s\right)}{P_2}\right]} \tag{15}$$

Thus, the mean greenhouse temperature can be given by:

$$T_g = T_a - \Delta T \tag{16}$$

The temperature of the greenhouse air at the pad end depends on a number of processes. Those are, the chemical dehumidification at the desiccant wheels, sensible cooling at the heat exchanger and the evaporative cooling at the cooling pads.

The air temperature at the pad end can be represented as:

$$T_{pad} = T_{he} - \varepsilon_p.(T_{he} - T_{wbhe}) \tag{17}$$

In Eq. (17), 'ε_p' denotes the effectiveness of the cooling pads, while 'T_{wbhe}' represents the wet bulb temperature of the air at the exit of the heat exchanger. In the present work

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the effectiveness of the cooling pad is assumed to be 0.88 [2]. The term 'T_{he}' represents the temperature of air at the exit of the heat exchanger, which can be given by:

$$T_{he} = T_{de} - \varepsilon_{he}.(T_{de} - T_{water}) \tag{18}$$

Where, 'ε_{he}' is the effectiveness of the heat exchanger, whose value is considered to be 0.8 [6]. The term 'T_{water}' denotes the temperature of water, which is assumed to be equal to the wet bulb temperature of the ambient air, while 'T_{de}' represents the temperature of air at the exit of the desiccant wheel, which in turn can be estimated from the specific humidity of the air at the exit of the desiccant wheel, which in turn can be given by:

$$\omega_{de} = \omega_a - m_a / m \tag{19}$$

Where, 'ω_a' represents the specific humidity of the ambient air. The term, 'm' denotes the mass flow rate of the process air.

The rate of adsorption of the moisture (Kg/s) due to chemical dehumidification can be given by the following expression [7]:

$$m_a = \frac{\left\{2.\sqrt{(3600.D_s.\beta_s)}.\sqrt{n}.\rho_d.a.m_e.A_d.L_d.\rho_b\right\}}{\left(3600.\sqrt{\pi}\right)} \tag{20}$$

Where, 'β_s' represents the fraction of the regeneration area to the adsorption area of the desiccant wheel, whose value is assumed to be 0.215 [7]. The term, 'n' is the rotational speed of the wheel in RPH, which is considered to be equal to 12 [7]. The terms 'ρ_d' and 'ρ_b' represent the density of the desiccant material and bulk density of the desiccant wheel respectively. In the present work the values of 'ρ_b' and 'ρ_d' are considered to be equal to 240 kg/m^3 [8] and 720 kg/m^3 [9] respectively. The term 'a' denotes the specific surface area of the desiccant material whose value is assumed to be 450 m^2/g [7]. The term 'A_d' denotes the cross sectional area of the desiccant wheel and 'L_d' represents the length of the desiccant wheel. In this work, the values of 'A_d' and 'L_d' are considered to be 0.08 m^2 and 0.2 m respectively. In Eq. (20), the term 'm_e' represents the equilibrium amount of moisture adsorbed by the desiccant materials, which in turn can be given by [7]:

$$m_e = 0.24 \times \varphi^{(1/1.5)} \tag{21}$$

Where, 'φ' denotes the relative humidity of the air, entering the desiccant wheel from the atmosphere.

The surface diffusion coefficient of the desiccant material can be expressed as [10]:

$$D_s = 2.27 \times 10^{-7} \exp(-h_s / RT) \tag{22}$$

In Eq. (22), 'h_s' represents the heat of adsorption. The heat of adsorption for the silica-gel varies between 2100-2300 kJ/kg [7]. In the present work its value is considered to be 2300 kJ/kg. The term 'T' represents the temperature at which diffusion process occurs. In the present study, the average temperature of ambient and regeneration air is considered to estimate the diffusivity [7].

The temperature of the air at the exit of the regenerator, which is used for the regeneration of the desiccant material can be given by:

$$T_r = F_R.A_p.S_c / (m_r.C_{pa}) + T_a \tag{23}$$

In Eq. (23), 'm_r' denotes the mass flow rate of the regenerative air, while 'A_p' represents the area of the absorber plate. The term 'S_c' denotes the solar flux absorbed by the absorber plate, while 'F_R' represents the collector heat removal factor, which in turn can be given by [11]:

$$F_R = \frac{m_r.C_{pa}}{U_l.A_p}.\left[1 - \exp\left\{-\frac{F'.U_l.A_p}{m_r.C_{pa}}\right\}\right] \tag{24}$$

In Eq. (24), 'U_l' is the overall heat transfer coefficient of the collector, whose value is considered to be 5.08 W/m^2-K [11]. The term 'F'' represents collector efficiency factor, which can be expressed as [11]:

$$F' = \frac{1}{W.U_l.\left[\frac{1}{U_l.\{(W - d_o).\eta + d_o\}} + \frac{1}{\pi.d_i.h_f}\right]} \tag{25}$$

In Eq. (25), 'd_o' and 'd_i' denote the outer and inner diameter of the tube, through which the air flows and 'η' denotes the absorber plate efficiency. The term 'h_f' represents the air to tube heat transfer coefficient, which can be given by [11]:

$$h_f = Nu.k_a / D_i \tag{26}$$

Where, 'k_a' denotes the thermal conductivity of air, which is considered to be 0.0259 W/m-K [11]. The Nusselt Number can be expressed as [11]:

$$Nu = 0.0158 \times Re^{0.8} \tag{27}$$

In Eq. (27), 'Re' is the Reynolds Number, which can be given by:

$$Re = (\rho_a.V_{avg}.d_i) / \mu \tag{28}$$

In Eq. (28), 'μ' denotes the dynamic viscosity of air, whose value is considered to be 18.1×10^{-6} N-s/m^2 [11]. The term 'V_{avg}' denotes the average velocity of the air inside the tube of the flat plate collector.

During regeneration, the water molecules diffuse out of the desiccant materials according to the same theory as discussed in Eq. (20) [7]. Therefore, as per literature [7], during the process of regeneration same amount of moisture will be ejected to the regenerative air.

For the economical analysis of the proposed system, a model for cumulative cash flow (CCF) has been developed, which can be given by:

$$\begin{aligned} CCF = {} & \frac{C_y}{(d - i_y)}.\left[1 - \left(\frac{1 + i_y}{1 + d}\right)^n\right] \\ & - \frac{i_l.f_l.C}{\left[1 - 1/(1 + i_l)^{n_l}\right]}.\frac{1}{d}.\left[1 - 1/(1 + d)^{n_l}\right] \\ & - \frac{C_e.E_c}{(d - i_e)}.\left[1 - \left(\frac{1 + i_e}{1 + d}\right)^n\right] - \frac{C_m}{(d - i_m)}.\left[1 - \left(\frac{1 + i_m}{1 + d}\right)^n\right] \\ & - \frac{C_{plant}}{d}.\left[1 - \frac{1}{(1 + d)^n}\right] - \frac{C_{Labour}}{d}.\left[1 - \frac{1}{(1 + d)^n}\right] \\ & - (1 - f_l).C \end{aligned} \tag{29}$$

In Eq. (29), the term 'C_y' denotes the total revenue from the flower yield per year; while 'i_y' is the rate of increase (%) of the flower cost per year. The term, 'd' represents the market discount rate, while 'n' denotes the service life of the proposed system. The terms 'f_l' and 'i_l' denote the fraction of capital taken as loan and interest on loan respectively, while the terms 'C' and 'n_l' represents the capital cost and time required to pay back the loan respectively. The terms 'C_e', 'C_m', 'C_{plant}' and 'C_{labour}' represent the electricity cost, maintenance cost, plant associated cost and labour cost respectively (on the first year), while 'i_e' and 'i_m' denote the increase in electricity and maintenance cost per year; 'E_c' is the electricity consumed by the system (kWh/year).

IV. Results and Discussions

A computer code in C language has been developed for the thermal model presented in the earlier section. The model considers the hourly data for solar radiation intensity, ambient air temperature and the relative humidity as input parameters and predicts the average greenhouse temperature for a given degree of shading. In the present work, the weather data for Kolkata as available from R. M. C. Kolkata [12] have been used. The performance of the proposed system (in terms of the mean greenhouse temperature) has been compared with that of a distributed fan-pad ventilated greenhouse (without desiccant materials).

Figure 2 shows the hourly variation of ambient and average greenhouse air temperature for a representative day in summer (15th May). During the month of May, both the intensity of solar radiation and the ambient air temperature remain high, while the humidity level remains moderate to high. From Fig. 2, it is evident that with the proposed system the maximum greenhouse temperature could be restricted within 26 ºC, when the corresponding ambient air temperature reaches about 34.5 ºC (with 61% RH). On the other hand, with the distributed fan-pad cooling system without desiccation, the maximum temperature reaches about 29 ºC.

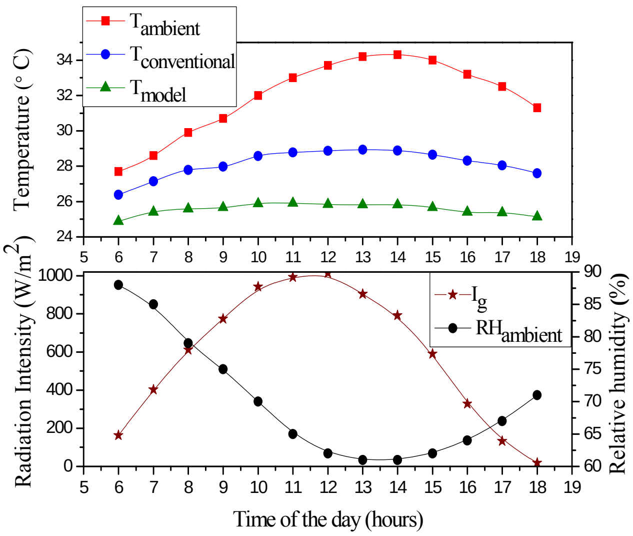

Fig. 2: Hourly variation of ambient and greenhouse air temperature on 15th May, 2009.

Therefore, it is evident from the figure that the conventional system fails to provide ideal temperature for Gerbera cultivation during the peak sunshine hours.

Figure 3 shows the hourly variation of ambient and greenhouse air temperature for a representative day in the rainy season (16th August, 2009).

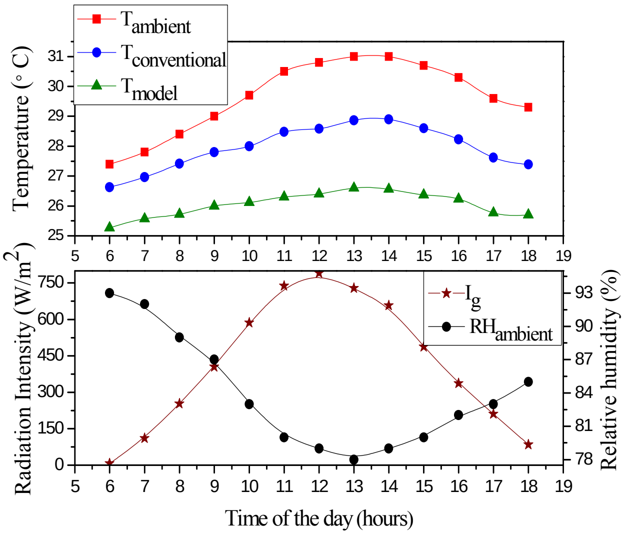

Fig. 3: Hourly variation of ambient and greenhouse air temperature on 16th August, 2009.

It is found that the distributed evaporative cooling system alone is able to reduce the greenhouse temperature marginally by around 2 ºC, while, the proposed system is able to reduce the same by about 4.5 ºC even during the hours of peak sunshine. Also as evident from the Fig. 3, for the daylong operation of the proposed system, the maximum greenhouse temperature reaches about 26.5 ºC when the ambient air temperature is 31 ºC (with 78% RH).

Figure 4 shows the hourly variation of ambient and greenhouse air temperature for a representative day in the autumn season (15th October, 2009).

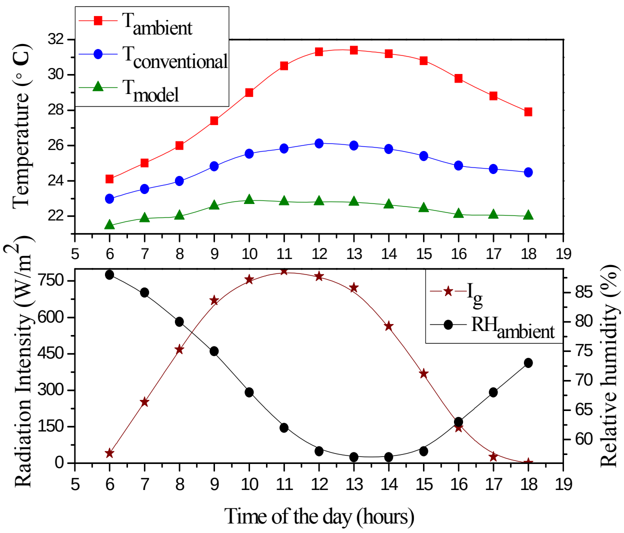

Fig. 4: Hourly variation of ambient and greenhouse air temperature on 15th October, 2009.

It is conspicuous from the figure that during the time of high ambient air temperature and moderate relative humidity (31.4 °C and 57% RH), the distributed evaporative cooling system alone is able to restrict the mean greenhouse temperature within 26 °C but at the same time the proposed system is able to restrict the same within 23 °C.

The economical model considered the market discounting rate to estimate the discounted payback period. From the analysis it was found that with the increase in the flower price, the payback period was decreasing. The Gerbera

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flower is generally sold at Rs. 10-12 per flower [13]. In the present analysis a lower price has also been considered to estimate the payback period of the system at that price range.

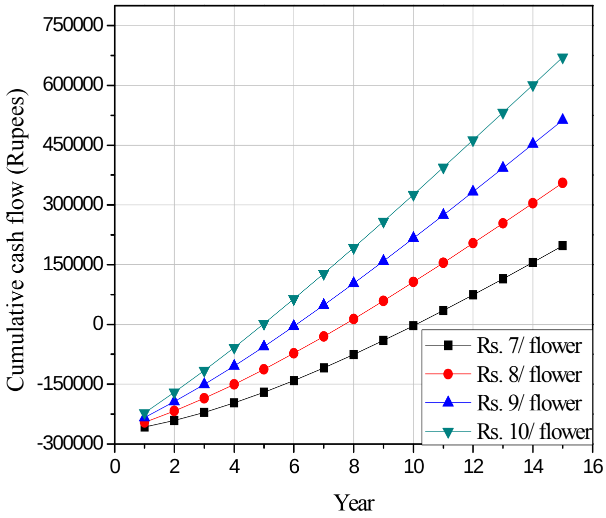

Fig. 5: Cumulative cash flow at various flower prices

Figure 5 shows the variation of the cumulative cash flow with the year of operation (at various rates of flower price). From the Fig. 5, it is evident that at Rs. 10 per flower, the cumulative cash flow (CCF) becomes zero at about 5 years, which is the payback period of the system at that price range, while at Rs. 7 the payback period is about 10 years.

V. Conclusion

In the present wok a thermal model and a cumulative cash flow model have been presented for a solar desiccant supported distributed fan-pad ventilated greenhouse. From the analysis, it is revealed that the proposed system is able to maintain a lower temperature inside the greenhouse compared to that achieved through distributed evaporative cooling system. The economical analysis reveals that the payback period of the proposed system is about 5 years, considering the flower cost to be Rs. 10.

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