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Research Article

EVOLUTION OF DESIGN CRITERIA FOR EARTH AIR -PIPE COOLING SYSTEM OF GREENHOUSE

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Abstract: A mathematical model has been developed to evolve criteria for optimally designing an earth air–pipe cooling system employed in greenhouses. Relations have also been developed to compute the size of earth air pipe cooling system required for the space to be cooled. This cooling system is characterized by two parameters *i.e.* pipe length and cross-section and volumetric air flow rate ($m^3s^{-1}m^{-2}$ floor area). These two parameters have been optimized to remove the maximum heat flux from the greenhouse enclosure. For a given environmental parameter φ (φ = (Tr-Ts)/(Ta-Ts)), the optimum air flow rate ($m^3s^{-1}m^{-2}$ floor area) cab be computed. A polynomial relation between environmental factor (φ) and climatic factor (CF) was given. For creating a desirable temperature inside greenhouse the value of φ is decided and then CF is calculated. The ratio A/At has been related to φ and CF. For a given value of flow velocity (v_t), we can determine the cross-sectional area of pipe in m^2. This model has proved to be very useful in designing the optimal cross-sectional area of pipe and optimal volumetric air flow rate for removing the maximum heat load from the greenhouse.

Keywords: *Air-pipe cooling, Optimization, Maximum heat removal, Climatic factor, Environmental factor*
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Introduction

A greenhouse is an enclosure of a transparent plastic/glass cover for ensuring required environment to grow desired crops. The temperature inside the greenhouse increases due to greenhouse effect which needs to be reduced during summer. The ground heat exchanger (GHE) reduces temperature by upto 6°C. Such system can be used for cooling greenhouses.

Various methods for cooling the greenhouses are used. Natural ventilation is the first step due to its low cost, but is generally not sufficient for exiting the excess energy during sunny summer days. Therefore, other cooling methods can be used in combination with ventilation. Air passed through a pipe buried inside the earth at a depth where earth temperature remains constant gets cooled in summer and heated in winter. In hot climates, the use of earth air pipe system for space cooling is a well-known concept. Sodha *et al.* [1] presented the results for theoretical and experimental investigation for a large earth air pipe system. Bharadwaj *et al*. [2] reported that the temperature of the earth varies considerably up to 1m depth and remains relatively constant beyond that.

The ground at certain depths can work as both heat source and sink during different parts of the year. Singh [3] provided a model for cooling through earth tube. Santamouris *et al.* [4] developed a greenhouse of 1000 m^2 area and obtained heat gain through latent heat material and they validated the data with measured values which were found in close proximity [5] and ground heat exchanger (GHE) [6] reported that ground heat exchanger worked well to regulate the nighttime temperature required for crop. The stable temperature of ground at certain depth help in cooling during daytime and heating during nighttime. The ground heat exchanger (GHE) was also tried in arid zone of India by [7]. In Kucth region of Gujarat, India, the ground heat exchanger (GHE) system for greenhouse application is a single-pass GHE reported by Sharan and Jadhav [8]. After testing it was single loop, they went for a closed loop using optimal length and diameters of pipes [9]. The system reduced the temperature by 7°C below the ambient temperatures. They also tested ground heat exchanger (GHEs) [10] for cooling or heating for greenhouse and animal dwelling application in Gujarat, India. An earth coupled thermal modeling of a greenhouse was developed at the Indian Institute of Technology Delhi, India [11-13]. The greenhouse was integrated with a GHE. It is also considered to be a good option to cool the human residence [14]. The earth air pipe system can be very useful in cooling the greenhouse space. The extent to which the ambient temperature can be brought close to that of the pipe surface depends on various parameters *viz*, dimension of the pipe, condition of the surface and flow velocity *etc*. A mathematical model has been developed to evaluate the effect of these parameters on the performance of the system. It is found that for a required thermal environment inside a greenhouse, there exist optimum values of size and fan parameters. The cooling through air pipe cooling system is based on the principle that the ambient air entering the pipe dissipates heat through convection, which then passes into the ground through conduction, and in the process ambient air gets cooled

The size of cooling system is usually characterized by the diameter of fan and air mass flow rate. The other parameters are pipe diameter or cross-section and length. In this communication, we have made an attempt to optimize the parameters *viz*. Optimum fan power and pipe cross-section for maximizing heat removal from a greenhouse for the purpose of our calculations we have taken unit floor area. For various numerical ratio of (Ac/Ag), the fan power has been optimized to remove the entire heat entering a greenhouse enclosure. The required fan power is found to be linearly dependent on the (Ac/Ag) of greenhouse. The relationships have been developed for computing optimum fan power and pipe cross-section for various ratios of (Ac/Ag). These relations provide a rule of thumb to decide the optimum fan power and pipe diameter and length. The schematic of an earth-air tunnel system along greenhouse is shown in Fig. 1. The objective of the paper is to develop a mathematical model for design criteria for optimal flow rate and dimension of earth tube system including length and diameter of pipe.

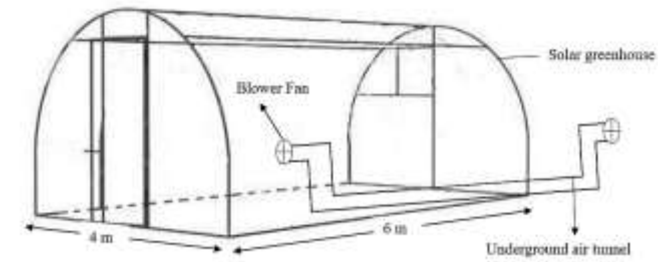

Fig-1 Earth air heat exchangers tunnel system installed to the greenhouse

Material and Methods

Thermal analysis

Rate of heat flux removal by an air-pipe system

The rate at which heat is removed by air pipe system is given by Bansal *et al.* [15]

$$Q_r = m_a C_a (T_r - T_L)$$

$$Q_r = \rho A_t V_t C_a [(T_r - T_s) - (T_a - T_s)\left[1 - \exp\left\{\frac{-(h_c + h_D C_a) bL}{\rho A_t V_t C_a}\right\}\right] \text{-------(1)}$$

Following Threlkeld (16), heat and mass transfer coefficients are related as,

$$\frac{h_c}{h_D C_a} = 1$$

The equation (1) can be rearranged as follows,

$$Q_r = \rho A_t V_t C_a [(T_r - T_s) - (T_a - T_s)\left[1 - \exp\left\{\frac{-(2h_c bL)}{\rho A_t V_t C_a}\right\}\right] \text{-------(2)}$$

and, $h_c = \delta + \tau V_t$

where, $\delta = 5.7$ and $\tau = 3.85$

and

$$Q_r = \rho A_t V_t C_a [(T_r - T_s) - (T_a - T_s) \exp\left\{\frac{-2(\delta + \tau V_t) bL)}{\rho A_t C_a}\right\} \text{-------(3)}$$

$$Q_r = \rho A_t V_t C_a \ (T_a - T_s)\left[\frac{(T_r - T_s)}{(T_a - T_s)}\right] \exp\left\{\frac{-2(\delta + \tau V_t) bL)}{\rho A_t C_a}\right\}$$

$$Q_r = \rho A_t V_t C_a \ (T_a - T_s)\left[\varphi - \exp\left\{\frac{-2(\delta + \tau V_t) bL)}{\rho A_t C_a}\right\}\right]$$

$$Q_r = B A_t \left[\varphi - \exp\left\{\frac{-A}{A_t}\right\}\right]$$

Where,

$B = \rho V_t C_a \ (T_a - T_s)$

$A = \frac{-2(\delta + \tau V_t) bL}{\rho C_a}$

$\varphi = \left[\frac{(T_r - T_s)}{(T_a - T_s)}\right]$ which is also defined as environmental factor and is given as the ratio of difference of room temperature and wet–bulb temperature per degree wet–bulb depression.

Optimization of fan power

The optimum value of fan power can be obtained from the condition $\left(\frac{dQ_r}{dA_t}\right) = 0$.

For which environmental parameter and optimum cross-section is related as:

$$\varphi = \left(\frac{1 + A}{A_t}\right) . e^{\frac{A}{A_t}} \text{-------(4)}$$

The corresponding maximum heat removed from the thermal load is given by

$$(Q_r)_{max} = \rho A_t V_t C_a \ (T_a - T_s)\left[\varphi - exp\left\{\frac{2(\delta + \tau V_t) bL)}{\rho A_t C_a}\right\}\right]$$

On substituting the value of φ from equation (4) into equation (5), we get maximum heat removed as,

$$(Q_r)_{max} = \rho A_{topt} C_a \ (T_a - T_s)\left[\left(\frac{A}{A_t}\right) \ e^{\frac{A}{A_t}}\right]$$

The second derivative $\left(\frac{d^2 Q_r}{dF_f^2} = -ve\right)$ is negative which clearly indicates that the heat removed will be maximum.

Thermal load of greenhouse

The sensible heat balance of a greenhouse fitted with an earth air-pipe cooling system is given by a first order differential equation,

$$\rho V C_a \ \frac{dT_r}{dt} = \tau\alpha S A_g \ \left(\frac{\beta}{\beta + 1}\right) + U A_c \ (T_a - T_r) - \rho A_t V_t C_a \ (T_a - T_s) \ \left[\left(\frac{A}{A_t}\right) \ e^{\frac{A}{A_t}}\right] \text{-(7)}$$

The first term on the right hand side of the equation (7) corresponds to the sensible heat gain from the floor and the plant mass, the second term corresponds to the overall heat gain from cover to the inside and third term represents the heat removed by the cooling system.

For a steady- state condition,

(Q_r) removed = Q_i heat entering the greenhouse

$$\tau\alpha S A_g \ \left(\frac{\beta}{\beta + 1}\right) + U A_c \ (T_a - T_r) = \rho A_t V_t C_a \ (T_a - T_s) \ \left[\left(\frac{A}{A_t}\right) \ e^{\frac{A}{A_t}}\right] \text{----(8)}$$

$$\rho A_t V_t C_a \ (T_a - T_s) \ \left[\left(\frac{A}{A_t}\right) \ e^{\frac{A}{A_t}}\right] = A_g D + U A_c \ (T_a - T_s)(1 - \varphi)$$

or

$$B A_{top} \left[\left(\frac{A}{A_t}\right) \ e^{\frac{A}{A_t}}\right] = A_g D + U A_c (1 - \varphi)$$

$$\text{or } A_{top} = \frac{A_g D + U A_c \left[1 - \left(\frac{A}{A_t}\right) e^{\frac{A}{A_t}}\right]}{B \left(\frac{A}{A_t}\right) e^{\frac{A}{A_t}}} \text{-------(9)}$$

Where,

$B = \rho C_a (T_a - T_s) V_t$

$D = \tau\alpha S \ (\beta/(\beta + 1))$

$$\text{Or, } V_t A_{top} = \frac{D}{\rho C_a \ (T_a - T_s)\left(\frac{A}{A_t}\right) e^{\frac{A}{A_t}}} + \frac{\frac{A_c}{A_g U\left\{1 - \left(1 + \frac{A}{A_t}\right) e^{\frac{A}{A_t}}\right\}}}{\rho C_a \ \left(\frac{A}{A_t}\right) e^{\frac{A}{A_t}}}$$

$$\text{Or } F_{FOP} = \frac{\left[\alpha\tau S\left\{\frac{\beta}{\beta + 1}\right\} + U\left(\frac{A_c}{A_g}\right)(1 - \varphi)(T_a - T_s)\right]}{1200 \ (T_a - T_s) CF} \text{------(10)}$$

Climatic factor (CF) can be defined as the rate of heat removal per m^3 of cooled air flow per degree Celsius wet-bulb depression given as,

$$CF_0 = \frac{(Q_r)_{max}}{\rho F_{Fop} C_a (T_a - T_s) \frac{(T_a - T_r)}{(T_a - T_s)}}$$

Determination of A/A_t ratio

Optimum climatic factor is given as,

$$CF_0 = \frac{(Q_r)_{max}}{\rho A_{topt} C_a (T_a - T_s)} \left(\frac{A}{A_t}\right) e^{\frac{A}{A_t}} \text{-----(11)}$$

$$\varphi = \left(1 + \frac{A}{A_t}\right) e^{\frac{A}{A_t}}$$

$$(\varphi - CF_0) = \left(1 + \frac{A}{A_t}\right) e^{-\frac{A}{A_t}} - \left(\frac{A}{A_t}\right) e^{-\frac{A}{A_t}}$$

or = ln (φ-CF)---------------(12)

Thus, the ratio A/A_t can be computed using equation (12)

Results and Discussion

Effect of environmental factor (φ) on optimum fan flow

For calculating maximum heat removed by a cooler equation (6) can be used. For computing climatic factors for various values of environmental factor a fourth order polynomial relationship was established as given by equation $CF = -0.9419\varphi^4 + 1.089\varphi^3 - 0.7979\varphi^2 + 0.8722\varphi$ ($R^2 = 0.9999$) as given in [Fig-2], which indicates that there is a strong correlation between these two parameters. Using this relationship for any value of φ, the value of CF can be determined.

The required fan power increases with increasing ratio of A/A_t because the value of environmental factor, φ decreases with increasing ratio A/A_t. Thus the low value of, φ increases the value of $(T_a - T_r)$ which is given as $(T_a - T_s)(1 - \varphi)$ and heat entering the greenhouse through the cover increases tremendously. The value of optimum flow rate FOP will be somewhat lower for a shaded greenhouse than that for an unshaded one as the value of sensible heat load $[\alpha\tau S\{\beta/(\beta + 1)]$ in equation (10) will be lower due to lower value of transmittance.

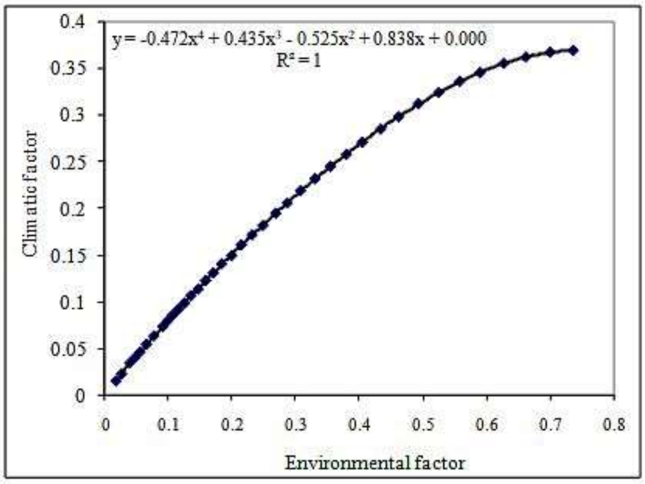

Fig-2 Climatic factor as affected by environmental factor

Determining optimum flow rate and size of cooler

The value of optimum A/A_t ratio can be determined with the help of equation (12). Once the value of optimum A/A_t has been determined, the optimum fan power can be computed by using equation (10). The structure of the greenhouse (semi-circular type), its size, properties of material and climatic parameters are given in [Table-1]. The values of [Table-1] can be used for a partially cropped shaded greenhouse (6m×4m) size (Quonset type) for determining optimum flow rate and size of air-pipe (cross-section and length of pipe). If the value of φ is taken as 0.405, the corresponding value of CF and A/A_t come to 0.27 and 2.0 from equations $CF= -1.9419\varphi^5 +1.9419\varphi^4 +1.3129\varphi^3 -0.8983\varphi^2 +0.8507\varphi$ and equation (12).

The sensible heat load for a unit floor area can be determined for an irradiance level S= 940 $W.m^{-2}$ in respect of an unshaded partially cropped greenhouse as 238.29 Wm^{-2}. The total flow requirement is 0.075 $m^3.s^{-1}$ per m^2 floor area. The flow rate for a 24 m^2 greenhouse comes to 1.809 m^3s^{-1}. The value of A_{top} (optimum cross-section) comes to 0.258m^2 at V_t = 7ms^{-1}. The radius of pipe is computed as 0.286m and its periphery is 1.8m. Putting the value of A_{top} in (A/A_t = 2.0), the value of A is calculated as 0.529. By substituting the values of ρ, V_t, C_a, δ, τ, b, ρ and A in $A = 2(\delta/v_t + \tau)\, b\, L/\rho C_a$, the value of length of pipe comes to 37.8m.

Table-1 *Parameters used in calculations*

Parameter	Value	Unit
ρ	1.2	Kg m^{-3}
C_a	1000	J kg °C^{-1}
U	10	Watt m^{-2}°C^{-1}
(T_a-T_s)	18	°C
A_g	24	m^2
τ	0.65 (unshaded greenhouse) 0.30 (shaded greenhouse)	
α	0.65	
S	940	Watt.m^{-2}
β	1.5 for partially cropped 1.0 for fully cropped	
A_c/A_g	1.8	

Conclusion

The developed model can prove very useful in determining the design criteria for optimal air flow rate and dimension of earth tube system including length and diameter of pipe. It is very useful in ensuring the desired environment inside a greenhouse. Greenhouse systems can be equipped with several solar or energy storage systems to provide a favorable microclimate for different type of crops.

Application of research: The developed model can prove very useful in determining the design criteria for optimal air flow rate and dimension of earth tube system including length and diameter of pipe in greenhouse cooling system.

Research Category: Energy and Environmental Studies

Abbreviations: α= absorptivity of floor, β= Bowen ratio
φ = environmental factor, τ= over all transmittivity, ρ = density of air (Kg m^{-3}),
T_a-T_s = wet-bulb depression (°C), T_s = temperature of tunnel surface (°C)
U = over all heat transfer coefficient (Watt m^{-2}°C^{-1})
h_c = convective heat transfer coefficient (Watt m^{-2}°C^{-1})
h_D = mass transfer coefficient (Watt Kg^{-1}°C^{-1})
A_t = cross-section of air pipe (m^2), A_c = cover area (m^2), A_g = floor area (m^2)
F_{Fop} = fan power or air flow rate (m^3s^{-1}), Q_r = heat removed (W),
Q_i= heat entering the greenhouse (W), S= Solar radiation ($W.m^{-2}$),
V_t = air flow velocity through pipe (ms^{-1}), C_a = specific heat of air (J kg °C^{-1})

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Author statement: All authors read, reviewed, agreed and approved the final manuscript. Note-All authors agreed that- Written informed consent was obtained from all participants prior to publish / enrolment

Study area / Sample Collection: Kucth region of Gujarat, India

Cultivar / Variety / Breed name: Nil

Conflict of Interest: None declared

Ethical approval: This article does not contain any studies with human participants or animals performed by any of the authors
Ethical Committee Approval Number: Nil

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