ENERGY EFFICIENCY AND PERFORMANCE EVALUATION OF HYBRID PHOTOVOLTAIC SYSTEM FOR FAN-PAD OF GREENHOUSES

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Abstract

This research is oriented towards improving energy efficiency in the cooling of greenhouses equipped with pad fan system. Once the least amount of energy was achieved, it was installed and evaluated the performance of a hybrid photovoltaic system inexpensive. Two of the most important branches of research related to agricultural greenhouses were unified to achieve the objective, namely, 1) the mathematical modeling of greenhouse climate and 2) experiments of solar energy for greenhouses. The results of the mathematical model allowed selecting the adequate ventilation rate and avoiding unnecessary consumption of electrical energy in the exhaust fans. After determining the optimal ventilation rate, it was calculated the energy required by fans depending on the hours of operation. A hybrid photovoltaic system with backup was evaluated with a solar resource of 5.5 kWh per day (Central region of Mexico). The main objective of the hybrid system is not to provide all the energy required for extraction fans, but reduce the power consumption from the local grid. The reduction of energy consumption was 32% and the total cost of hybrid system was USD $3,400.00.

Keywords: *agricultural greenhouse, photovoltaics, mathematical modeling, saved energy*

1. Introduction

In general, mathematical models applied to agricultural greenhouses have two main objectives: 1) to get knowledge about greenhouses systems, without the need to reproduce expensive experiments, 2) to optimize the greenhouses processes of operation and management. In literature, mathematical models predict variables such as natural ventilation rate, temperature, relative humidity, carbon dioxide concentration, growth of crops, among others (Tiwari, 2003). However, their practical application to solving real specific problems of agricultural engineering is limited by their low reliability. On the other hand, photovoltaic systems are a source of electrical power also with low reliability, mainly because generated energy is variable along the day and year (Duffie, 1993). The above objections foster distrust the implementation of photovoltaic systems in greenhouses. The objective of this work is to solve these problems in order to reduce the electric power consumed by the cooling system of a greenhouse. First, a mathematical model is used to predict the energy required by the cooling system and its reliability is increased by validating with experimental data. After, the performance of a hybrid photovoltaic system that supplies electric energy for fans from solar panels and local grid was evaluated.

Greenhouses without cooling systems often have 11 °C higher indoor temperatures than the surrounding environment. The detrimental effects of high temperatures on crops are characterized by loss of strength in the stem, reduced flower size, delay flowering, among other (ASABE, 1995). The pad fan system can lower the temperature of incoming air by about 80 percent of the difference between the dry and wet bulb temperatures, by example, air at a dry bulb temperature of 32 °C and relative humidity of 20%, the wet bulb temperature would be of 17 °C and the pad fan system could lower temperature to approximately 20 °C. To generate the cooling effect, it is required a ventilation rate created by exhaust fans located on the opposite side wall of the evaporative pad. There are different technical methods based on experience for calculating this ventilation rate, for example, the National Greenhouse Manufacturers Association (NGMA) indicates in its 1993 standards for cooling greenhouses that a rate of removal of 13.6 m^3 per hour per m^2 is sufficient. Nelson (1998) suggests a method of applying correction factors to the previous value considering, elevation

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above sea level of greenhouse, light intensity and distance between fans and evaporative pad. Likewise, several authors have shown that steady-state mathematical models describe with high level of accuracy the thermal behavior of a greenhouse with pad fan system (Jain et al (2002), Ganguly, Kittas (2001)). In the field of energy efficiency applied to pad fan systems, Franco and Valera (2014) used a steady state model to describe the operation parameters of three different materials for evaporative pads. Since that results they concluded that evaporative cooling boxes are better option than cellulose pads for cooling non-hermetic greenhouses such as those most frequently used in the Mediterranean basin. The use of steady-state models has yielded practical results for the improvement of operation of greenhouses.

In reference to the use of photovoltaic systems in greenhouses, Al-Helal (2004) realized a study of pad-fan performance for a photovoltaic application and his results showed that electricity consumption increased non-linearly as the ambient temperature increased, only through experimentation and without modeling. Ahmed (2011) evaluated a standalone photovoltaic system for cooling a greenhouse, he proposed the design of isolated photovoltaic system to supply power to the fans of greenhouses in remote areas, however, the system is expensive and its main objective is to provide all energy, since it is off-grid and electrical loads are inductive decreases the reliability of the battery bank, jeopardizing the control of the depth of discharge and decreasing the life of the same. There are several studies related to photovoltaic for cooling greenhouses, Al-Shamiry et al (2007), Carlini et al (2012), Ganguly et al (2010), Al- Ibrahim et al (2006), however, none of them involves mathematical modeling to calculate electrical energy consumed by the fans, i.e. that the dimensioning of the photovoltaic system is not a function of the greenhouse temperature or ventilation rate.

2. Materials and methods

2.1 Instrumentation for greenhouse climate study.

The experiment was conducted in an experimental even-span greenhouse equipped with a pad-fan system (height: 5 m, length: 30 m and width: 11 m) of the Center for Plant Breeding Research of Autonomous University of Chapingo, located in Texcoco, Mexico (19.30 °N, 98.53 °W). The greenhouse is equipped with two fans located on the opposite side to the pad and their function is exhaust air, sucking there through pad. The rated power of each electric fan motor is 1.125 kW, and the electric motor of the water pump is 0.375 kW, all them work at 220 V. Greenhouse instrumentation was performed to measure air temperatures inside the greenhouse on a typical day at high temperatures (01/03/2015). Campbell Scientific datalogger (CR1000) was used. Variables recorded were: air temperatures (seven sensors, model108, Camp. Sc.) (before and after the pad and along the greenhouse), relative humidity (four sensors, model HMP50, Camp. Sc.). For measurements of solar radiation a sensor CMP3 (Camp. Sc.) was used.

2.2 Thermal modelling for energy demand.

In order to predict the thermal behavior inside agricultural greenhouses when ventilation rate varies, different mathematical models have been studied. Jain and Tiwari (2002) developed a detailed model in order to predict the temperatures inside the greenhouse divided in two zones (Eqs. 1-2), where $T_{r,i}$: temperatures of zones, I_i: solar radiation on various walls and roof (W/m^2), V_1: volume of greenhouse under zone-I (m^3), $\boldsymbol{H_{z2}}$: height of walls in zone II, $\boldsymbol{A_{nw1}}$: area in north wall (m^2), $\boldsymbol{U_i}$: overall heat transfer coefficient from cover zone-II to ambient air, A_i: area of walls and roofs of greenhouse (m^2) , $\boldsymbol{T_a}$: ambient temperature (°C), $\boldsymbol{h_{r1r2}}$: convective heat transfer coefficient of air from zone-I to zone-II (W/m^2°C), $\boldsymbol{A_g}$: area of floor under greenhouse (m^2) , $\boldsymbol{T_{r2}}$: room air temperature in zone-II (°C) , L: length of greenhouse (m) , a: constant value , $T_{r2,0}$: room air temperature in zone-II at 0 m (°C) . The model can be simplified and applied for the greenhouse of case study of this research which is equipped with pad-fan system (Eq. 3), from an energy balance Kittas (2001) developed the model solution. Ganguly and Ghosh (2006) compared the results of this model with other similars, demonstrating that steady state models described with acceptable accuracy the interior temperatures of greenhouses.

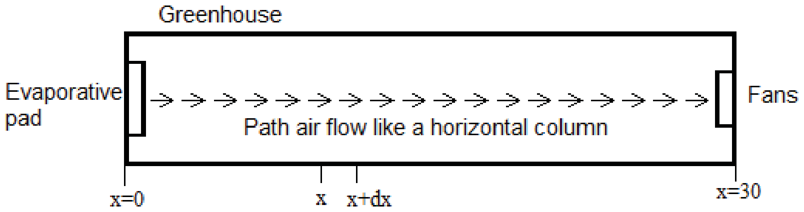

Fig.1. Greenhouse cross section scheme used for mathematical modeling. The main assumption is that air flow moves only in the horizontal direction. No temperature gradients in the vertical direction.

$$\boldsymbol{T_{r,1}} = \frac{I_1V_1 + I_2H_{z2}A_{nw1} + (U_{r2a}A_{nw1} + \sum U_iA_i)T_a + h_{r1r2}A_gT_{r,2}}{h_{r1r2}A_g + \sum U_iA_i + U_{r2a}A_{nw1}} \qquad \text{(eq. 1)}$$

$$T_{r,2} = \frac{1}{L}\int_0^L T_{r,2}dx = \frac{F(t)}{a}\left[1 - \frac{1-e^{-aL}}{aL}\right] + T_{r,2,0}\frac{1-e^{-aL}}{aL} \qquad \text{(eq. 2)}$$

Where $F(t) = \frac{T_{r2}}{dx} + aT_{r\mathbf{2}}$, and the simplified model applied to this case study is:

$$T_{in}(x) = \left[T_o + \frac{\tau(1-\alpha)R_ow_{gh}}{K_cL_{cov}}\right]\left[1 - e^{-\left(\frac{K_cL_{cov}}{\dot{V}\rho C_p}[x]\right)}\right] + T_{pad}e^{-\left(\frac{K_cL_{cov}}{\dot{V}\rho C_p}[x]\right)} \qquad \text{(eq. 3)}$$

In Eq. 3, $\mathrm{T_{in}}$(x) is the interior temperature at x position (Fig. 1), $\mathrm{T_o}$ is exterior dry bulb temperature, τ is transmissivity of the plastic cover; α: coefficient representing the fraction of radiation incident on the plant that is directly converted into latent heat by transpiration and losses of radiation through soil, $\mathrm{R_o}$: outside solar radiation in $\mathrm{W/m^2}$, $\mathrm{w_{gh}}$: greenhouse width en m, $\mathrm{K_c}$: coefficient of sensible heat losses through the plastic cover in $\mathrm{W/m^2}$ °C, $\mathrm{L_{cov}}$: length of cover proportional to the width of greenhouse to obtain the differential heat transfer area, $\dot{V}$: ventilation rate generated by fans; C_p: heat capacity of air in $J/kg°C$; ρ is air density, T_{pad} is air temperature when leaves evaporative pad.

This model can describe the temperature of a completely sealed greenhouse without evaporative pad, this is achieved when the ventilation rate, $\dot{V}$, approaches zero, this way the temperature is not a function of x, and there is only one temperature along greenhouse:

$$T_{in} = \left[T_o + \frac{\tau(1-\alpha)R_ow_{gh}}{K_cL_{cov}}\right] \qquad \text{(eq. 4)}$$

The term x represents the length of the greenhouse, that is, the distance between the pad and the fans. This term is limited by the exponential function, so it is expected that the model cannot be applied to great lengths greenhouses because the exponential function generates an asymptote and the value of the temperature would be the same. This asymptote is physically correct. Since for large distances between the evaporative pad and the fans, there would be no cooling effect inside the greenhouse. In general, the model describes physically a greenhouse with or without pad fan system. The main results of interest for research will be the temperature depending on the ventilation rate. That is, it is necessary to know the ventilation rate required to obtain a given temperature of 25 °C, which is most recommended for crops.

2.3 Hybrid photovoltaic system.

The amount of energy obtained from the mathematical model served as the first parameter design of a hybrid system. The hybrid system consists of four PV modules of 200 W each one, one hybrid inverter charger of 3.0 kW and four batteries of 150 Ah each one. The operation mode of Fig. 2 is described below:

t1: To ensure the battery is at full capacity, when the inverter is turned ON, the CPU will execute the bypass mode automatically connecting the AC main to the load. In the meantime, it will activate both the AC charger and solar charger to simultaneously charge the batteries.

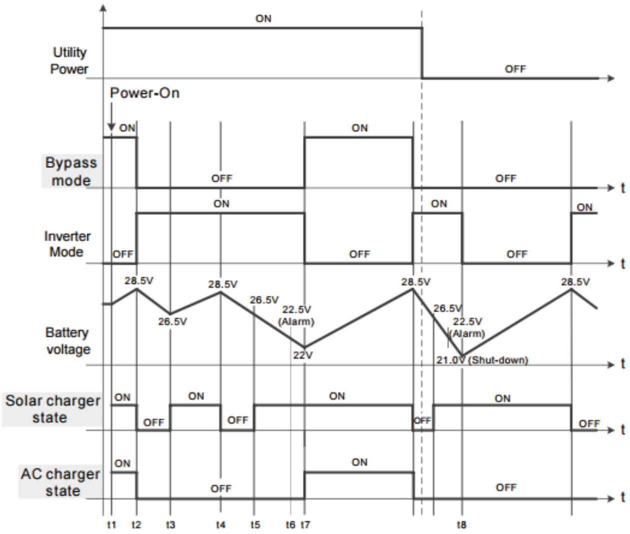

Fig. 2. Diagram of hybrid photovoltaic system mode control logic.

t2: When the batteries are full (battery voltage around 28.5 V), both the AC and solar charger will be turned OFF by the CPU to prevent overcharging and reducing battery lifetime. The CPU will also switch to inverter mode and the AC electricity provided to the loads will be coming from the batteries.

t3: When the batteries are depleted to around 90% of their capacity (battery voltage around 26.5V), CPU will start up the solar charger but not the AC charger to achieve the goal of energy saving.

t4: If the energy provided by the solar panels is larger than the load requirement, voltage of the battery bank will increase gradually until reaching battery voltage around 28.5 V and then the solar charger will shut off to prevent the batteries from overcharging.

t5: When the capacity of batteries goes down to battery voltage around 26.5 V, solar charger will restart and begin charging.

t6: If the energy provided by the solar panels is lower than consumed by the loads, voltage of the battery bank will decrease gradually to battery voltage around 22.5 V. The built-in buzzer will sound to inform the user that battery power is very low.

t7: If the power consumption of the loads does not decrease and the AC main is normal, the CPU will detect this and the unit will be transferred to "bypass mode." The utility will provide electricity to the loads and charge the battery bank at the same time in order to prevent the unit from shutting off. If the solar current is higher than 3 A, the CPU will not activate the AC charger and just let the solar changer charge the batteries to achieve energy saving target.

t8: When there is no AC main, the CPU will shut down the whole system if the external battery bank voltage is less than 21 V in order to prevent over-discharging and reducing its lifetime. After shutdown, the CPU will still provide LED indication so the user knows why the inverter has shut off.

Fig. 3: PV array and hybrid inverter with battery bank.

Fig. 4: View of PV system outside greenhouse.

3. Results and discussion

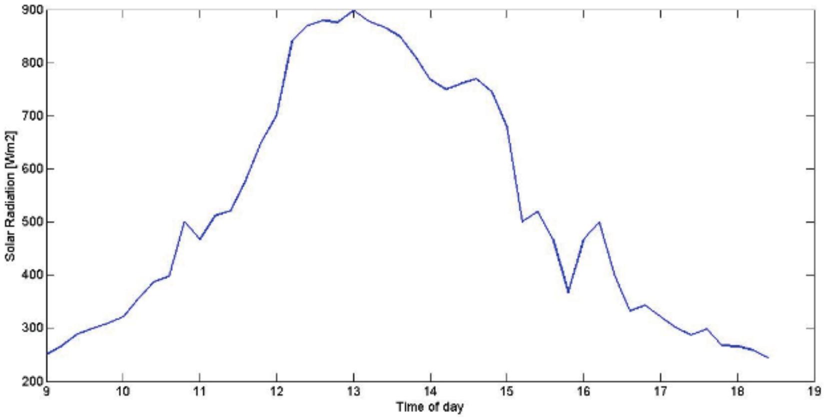

Fig. 5: Variation of exterior solar radiation on experimentation day (03 june 2015)

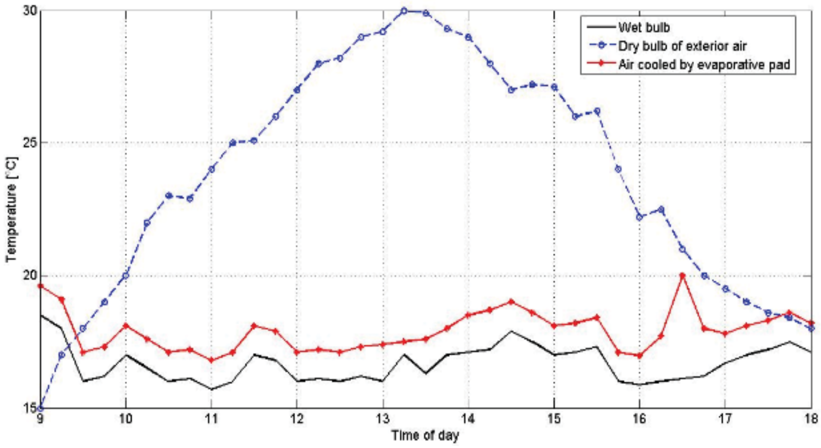

Fig. 6: Operation air temperatures involved in performance of evaporative pad on experimentation day.

The mathematical model does not include factors such as partial cloudy, rain and outside wind speed and these represent sources of random errors that generates uncertainty of non-reproducible measurements. For that reason, it was taken a specific day with the least influence of these factors, environmental conditions and operation temperatures of evaporative pad are shown in Figs. 5 and 6.

MatLab software was used to program solution. Figures 5 and 6 shows that when exterior dry bulb temperature was 30 °C the outside solar radiation achieve 890 W/m^2, and the wet bulb temperature was 17 °C. The average dry bulb air temperature leaving the pad was 18 °C. That means 85% of pad efficiency. This value is less than the common pads and could rise with an increase in ventilation rate, however the temperature of 25 °C is reached successfully with the current rate of ventilation. Here lies the importance of implementing energy efficiency plan to avoid consuming more energy than necessary.

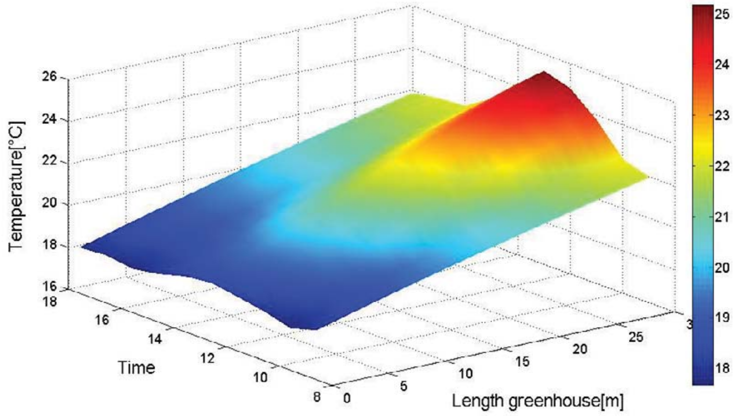

Fig. 7: Spatial temperature distribution inside the greenhouse during experimentation.

With the steady-state model it can obtain a spatial temperature distribution inside the greenhouse along the day when operating the evaporative wall and exhaust fans. To determine the temperature versus time, experimental data of radiation was used in the model along the day while keeping constant the ventilation rate. Fig. 7 shows the spatial temperature distribution inside the experimental greenhouse along the experimentation day. Fig. 8 shows predicted temperature inside greenhouse at x = 30 m, ie maximum temperatures at outlet in fans.

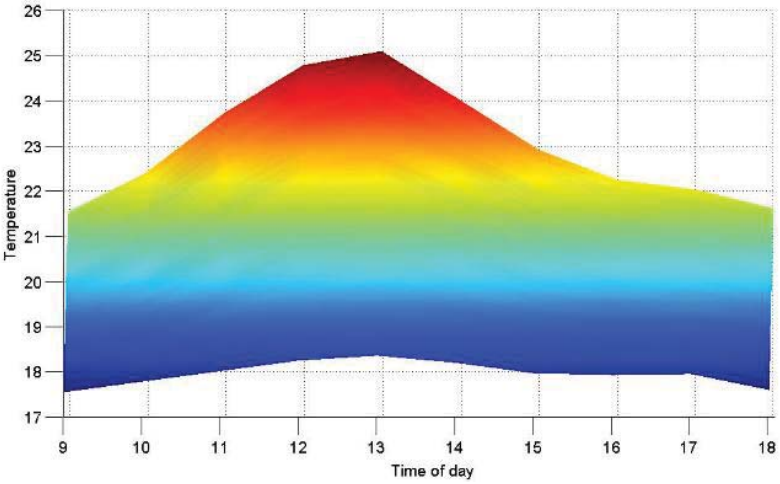

Fig. 8: Predicted temperature inside greenhouse at x= 30 m, ie maximum temperatures at outlet in fans.

To improve the energy efficiency of exhaust fans is necessary to determine the value of adequate ventilation rate, then to know the number of fans and power to meet that ventilation rate. As an example, in Table 1, three models of fans are proposed:

Tab. 1: Characteristics of three models of fans commonly used for pad fan system

Model	**Power** W	**Airflow rate** $m^3/h - m^3/s$
1	563	24666 - 6.9
2	746	27630 - 7.7
3	1119	31663 - 8.9

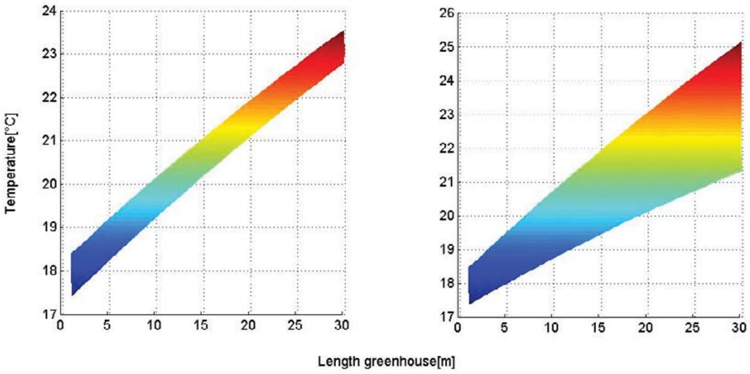

Fig. 9: Temperature profiles along greenhouse with 16 m^3/s (left) and 14 m^3/s (right).

Under same exterior conditions but different ventilation rate, it can obtained different temperature profile. It is expected that energy consumed is proportional to cooling effect. Fig. 9 shows two different temperature profiles generated from two ventilation rates. It is necessary to select fans to meet these ventilation rates of 16 and 14 m^3/s from the models proposed in table1. Table 2 shows results:

Tab. 2: Fans selected for different ventilation rates

Ventilation Rate [m^3/s]	**Maximum temperature obtained at x=30 [°C]**	**Fans selected**	**Energy consumed in 1 hour kWh**
16	23.5	Two fans of model 3	2.22
14	25	Two fans of model 2	1.49

In order to create a flow as uniform as possible, ASABE (2008) suggest that the fans should have the same air flow capacity and technical features, that is, they are the same model. For that reason, it is important to select just necessary ventilation rate, it would be creating more cooling effect than necessary for crops. Also, it would be wasting electrical energy. The ventilation rate of 14 m^3/s is sufficient to maintain 25 °C or lower temperature. If we use this design parameter, for 5 hours of operation of the pad fan system during the hottest part of the day, the energy consumed would be 7.45 kWh/day.

Fig. 10. Fans in opposite face to evaporative pad.

Theoretically energy generated by solar panels at Central Region of Mexico during experimentation is 3.3 kWh per day (5.5 sunshine hours equivalent to 1000 W/m^2 multiplying 600 W of power pv array). However, the energy measured with a Power Quality Analyzer (Fluke 434 Three phase power) during the 30 days of 2015 summer, the percentage of energy saving was 32%. The specific technical characteristics and unit prices of all major components shown, including also expenses for construction and installation of the support tower of photovoltaic modules, electrical cables, and labor. The total cost of the installed system was USD$3,400.00 and the recovery time is 10.5 years. After that the system will have an estimated 15 years additional life.

4. Conclusions

If a pad works at maximum efficiency, it does not guarantee that energy consumption is minimal. The increased pad efficiency could cause unnecessary energy consumption, this is because the optimum temperature inside a greenhouse can be obtained with a ventilation rate lower than that corresponding to the maximum ventilation pad efficiency. Consequently, the power of the fans must be lower and energy consumption as well. After this energy analysis, a hybrid photovoltaic system of low cost and high reliability was obtained. Energy savings of nearly a third of the total are obtained by keeping the greenhouse temperature close to 25 °C during the hours of maximum solar radiation. 32% of the energy saved is the best choice because other settings create more cost and time of return is achieved close to the life of the system. Even when AC utility is available, the main source of power will still be solar, AC utility will supplement only when necessary. This type of design cuts back the use of paid electricity thus reaching the goal of energy conservation.

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NOMENCLATURE

C_p	heat capacity of air $[J/kg°C]$
K_c	coefficient of sensible heat losses through the plastic cover $[W/m^2\ °C]$
L_{cov}	length of polyethilene cover proportional to greenhouse width to obtain the differential heat transfer area $[m]$
$\dot{m}$	mass airflow generated by fans $[kg/s]$
$\dot{Q}_{accum}$	heat rate accumulated in greenhouse $[J/s]$
R_o	outside solar radiation $[W/m^2]$
T_{in}	interior temperature $[°C]$

T_o	outside temperature $[°C]$
T_{pad}	air temperature cooled by pad of cellulose corrugated $[°C]$
$\dot{V}$	ventilation rate $[m^3/s]$
w_{gh}	greenhouse width $[m]$
τ	coefficient transmissivity of the plastic cover
α	coefficient of losses that involves transpiration and radiation through soil
ρ	air density $[kg/m^3]$