Feasibility Study of a Combi-PV Panel for Greenhouse Energy Supply and Water Recovery by Nightly Radiation towards the Sky

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Abstract

In southern European areas, characterized by high irradiation, the use of water for both evaporative cooling systems and hydroponic fertigation, represents a serious drawback for crop cultivation under cover. Water recovery systems seem to be an attractive solution, especially when they are integrated in the greenhouse construction. In this research, a feasibility study of applying a water recovery system driven by a combi-PV panel, in a semi-closed greenhouse was carried out. The prototype combi-PV panel was made by coupling an amorphous silicon panel with a sump stacked on the rear PV panel surface and filled with saline water. The system is driven by a cold-heat sink which is the PV panel itself. During night, the combi-PV panel exploits the radiative cooling of a 'gray' surface towards clear sky, chilling the water in the sump. In opposition, during day-time, the water in the sump is heated at a temperature higher than the environment. Thus, the water vapour will be condensing on the rear panel surface during night, being the warm air circulation facilitated by bouyancy effect. The evaluation of the system is in progress in order to assess the real amount of energy irradiated and consequently the water-drips to be collected on a proper surface inside the sump. The condensed water can be mixed with saline water to reduce the salinity and be used for fertigation.

INTRODUCTION

Many studies focused on the possibility of reducing water consumption by recovering water from the condensation of vapour available in greenhouses, most importantly for those located in a semi-arid climate (Lovichit et al., 2008; Hardin et al., 2008), and on the effect of water quality on the crop yield (Stanghellini et al., 2005). Even the energy saving is a compelling force in future development of environmentally friendly greenhouse activities. Thus, a new approach to those problems was attempted, coupling the issues. Thermal behaviour of a closed box filled with saline water and roofed with a PV panel, called Combi-PV, was investigated to evaluate its suitability for producing non-saline water by evaporation. A simulation software called Simul-CPV has been developed for estimating interior surface temperature and inner air temperatures. Using outside air temperature, direct and diffuse solar radiation, location and orientation, tilt angle and physical properties of the materials as input parameters, the model transforms the Fourier's differential equation into a system of finite difference equations depending on time, taking into account the net radiant heat emitted from each inner surface of the structure. The 'grey body' theory was applied to a close cavity, and, involving the view factors and its reciprocal theorem, a system of equation was derived and solved to find out the net radiant heat.

The Combi-PV system can be regarded as a heat phase shifter: during night-time it

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exploits the temperature difference between the warmed water and the cold roof radiating towards the sky; during day-time, the temperature difference between the warm water and the hot roof being irradiated by the sun. The simulation software predicts the evapo-condensation of saline water stored inside. Moreover, different materials and layer thickness can be tested. At the same time, the Combi-PV generates some electric power which could supply any needed device, mainly for dynamic ventilation, artificial lighting and pumps. An estimation of the electric energy requirements of a typical Mediterranean greenhouse (Campiotti et al., 2008) reports that the yearly average power electrical requirement ranges from a maximum of 90 000 $kWh_{el}.y^{-1}$ ha^{-1} for a greenhouse with a good climate control (heating, cooling or ventilation) to 20 000 kWh_{el} y^{-1} ha^{-1} for a very low technological greenhouse structure. Designing a PV field leads to take into account 0.65 kWp.(1 000 $kWh.year^{-1})^{-1}$ to cover the energy demand of the greenhouse in that region.

Thus, coupling a large available PV field area with the condensation-device could be both cost effective and environmentally friendly. Under such a working hypothesis, the use of amorphous silicon PV panels results compulsory in order to maximize the PV panel area.

MATERIALS AND METHODS

The Heat Transfer Model

The transient heat conduction model was derived from the Fourier's equation applied to each layer considering the thermal flux mono-dimensional (Kreith, 1973). The Fourier's equation was transformed into a finite difference equation, splitting each envelope wall (roof, floor and walls) into 'n' surfaces and 'n-1' layers of thickness 'Δx_j'. Number '1' was assigned to the outside surface and each layer was characterized by its physical and thermal properties. A time-increment, a finite interval, 'Δθ' was fixed to 240 s, denoting with 't' the number of intervals elapsed.

In correspondence of each 't·Δθ' interval (360 intervals were chosen to cover 24 h), a system of 'n' equations with 'n' unknown variables was written for each surface. Then, all the systems were solved by the Excel® program using Kramer's method. The layer temperatures at '(t+1)·Δθ' time were calculated depending on previous 't·Δθ' interval values. The following system of equations resulted for each time interval and for each envelope wall:

$$\left(T_1^{t+1} - T_1^{t}\right)C_1 = A\,\Delta\theta\left(\frac{T_e^{t+1}-T_1^{t+1}}{\frac{1}{h_e}}+\frac{T_2^{t+1}-T_1^{t+1}}{\frac{\Delta x_1}{k}}+q_e^{t+1}\right) \quad \text{outside layer} \tag{1}$$

$$\left(T_j^{t+1} - T_j^{t}\right)C_j = A\,\Delta\theta\left(\frac{T_{j-1}^{t+1}-T_j^{t+1}}{\frac{\Delta x_{j-1}}{k_{j-1,j}}}+\frac{T_{j+1}^{t+1}-T_j^{t+1}}{\frac{\Delta x_j}{k_{j,j+1}}}\right) \quad \text{layers} \tag{2}$$

$$\left(T_n^{t+1} - T_n^{t}\right)C_n = A\,\Delta\theta\left(\frac{T_i^{t+1}-T_n^{t+1}}{\frac{1}{h_i}}+\frac{T_{n-1}^{t+1}-T_n^{t+1}}{\frac{\Delta x_{n-1}}{k_{n-1,n}}}+q_i^{t+1}\right) \quad \text{inside layer} \tag{3}$$

The $T_e^{t+1}(\theta)$ and $q_e^{t+1}(\theta)$ values on external surfaces were measured, while $h_i^{t+1}(\theta)$ and $h_e^{t+1}(\theta)$, internal and external heat conduction coefficients, were fixed on the basis of a value-list correlated to wind speed. $T_i^{t+1}(\theta)$ and $q_i^{t+1}(\theta)$ values on internal surfaces were approximated by calculations at the previous iteration.

To evaluate $T_i^t(\theta)$ function, a simple balance equation of sensible heat was applied to air volume inside the structure, taking into account the volume flow due to air circulation. Thus, the equivalent inner temperature was calculated by the following equation:

$$T_i^t(\theta) = (T_{1-\mathrm{int}}(\vartheta) + T_{4-\mathrm{int}}(\vartheta))/2°C \tag{4}$$

To evaluate the net radiant heat emitted, or absorbed, from the surface 'i', the interior surfaces were described as 'opaque gray bodies' enveloping a close cavity. Thus, being the emittance 'ε' equal to the absorptance 'α', it results:

$$\varepsilon_i = \alpha_i \tag{5}$$

$$\rho_i = (1-\varepsilon_i) \tag{6}$$

$$J_i = \rho_i G_i + \varepsilon_i E_{bi} \tag{7}$$

where: 'J' is the surface radiosity, 'G' is the incident radiant heat, and 'E_b' is the black body emission power. The net radiant heat $q_i(\theta)$ emitted from the surface 'i' can be calculated by equation:

$$q_i/A_i = J_i – G_i \tag{8}$$

Using the principle of conservation of energy and the reciprocal theorem for the view factors, 'G' can be calculated as follows:

$$\sum_{j=1}^{N} F_{i-j} = 1 \tag{9}$$

$$A_n F_{n-I} = A_i F_{i-n} \tag{10}$$

$$G_i = \sum_{j=1}^{N} J_i F_{i-j} \tag{11}$$

Combining Equations 7-8 and Equation 11 to eliminate 'G', it results:

$$q_i/A_i = (E_{bi} – J_i)\, \varepsilon_i /(1-\varepsilon_i) = J_i – \sum_{j=1}^{N} J_i F_{i-j} \tag{12}$$

Thus, writing Equation 12 for each surface, a system of 'N' equations with 'N' unknown variables 'J_i' can be solved, and $q_i(\theta)$ is calculated.

During night, the sky-temperature was calculated by the following equation (Bournet et al., 2006; Swinbank et al., 1963):

$$T_{sky}=0{,}0552*(T_{air})^{1{,}5}\ [°K] \tag{13}$$

Thus the power irradiated towards the sky is:

$$5{,}67*10^{-8}*\ \varepsilon_{PV} *((T_{PV\text{-}panel})^4-(T_{sky})^4) \tag{14}$$

The simulation procedure resulted in a program called Simul-CPV.

Description of the System

The Combi-PV system was evaluated using the plan depicted in Figure 1. Table 1 shows physical and thermal properties of roof and walls layer materials (ground, eventually) as input values. Table 2 reports the dimension of the PV-panel and of the under-mounted system utilized as input parameters for the simulation model. The Combi-PV is a close-system containing a double inner space, connected at the bottom by a hose, separated by a thin layer at the foot of the roof from the upper part (see Fig. 1). The goal of the layer is to collect the drips of condensed water. The lower cavity contains saline water which is evaporated by the heat captured by the roof, namely a PV-panel. Every box wall is well insulated and therefore the PV-panel behaves as a heat exchanger.

The system can be regarded as a heat phase-shifter: during night-time it exploits the temperature difference between the warmed water and the cold roof radiating towards the sky; during day-time, the temperature difference between warm water and the hot roof being irradiated by the sun. In order to exploit the day-time heat stored inside this device, two micro-fans were connected to pipes for circulating hot and warm air respectively inside the two cavities, owing to the counter-bouyancy air movement needed. Figure 2 reports the circulation of air inside the condensation-device during night-time (a) and day-time (b). During night-time the PV-panel irradiates towards the sky and gets colder than the dew-point of the air, thus condensating water vapour. During day-time, the PV-panel gets hot and transfers heat to air which is circulated by a micro-fan; the hot air enriches itself of water vapor blowing in a little space having the saline water at the bottom. This air is laminated on the layer dividing the two main cavities by a plastic foil. The layer is refreshed by some air circulating inside the lower cavity by another micro-fan and plastic foil, thus keeping the layer colder than the dew-point of the air, except for the periods nearby the intersection of the two curves 'water surface temperature' and 'PV-panel inner surface'. Micro-fans can be supplied by the PV-panel itself. During summer-time the PV-panel roof can reach very high temperature values, more than 50°C, due to the high global radiation and the high air temperature (35°C maximum).

RESULTS AND DISCUSSION

Simulated data were processed by Simul-CPV software. Running some iterations, Simul-CPV software calculated the inside air temperature, the temperature of inner surfaces, and the net radiant heat exchanged among surfaces. Figure 3 shows the simulated temperature of the inner surface of the roof, as well as the inside air temperature. As one can see, the water mass behaves as a heat accumulator, shifting the heat phase and letting the system work. The radiant heat exchange q_i (θ) among roof and walls is depicted in Figure 4. It reports that q_i (θ) must not be neglected during night or day-time. The program calculates the net heat power stored inside the Combi-PV device as the result of heat exchange between the water mass and the outside air, through the 'walls'. Referring to 1 square meter of PV-panel, as shown in Figure 5, the transient net heat power values were calculated.

The water evaporated and condensed in the system can be easily calculated splitting the heat power values by the latent heat of water. Figure 6 shows the hourly evapo/condensed water inside the Combi-PV device. Summing up all the hourly amounts, it results that the system is capable to condense 0.6 kg d^{-1} m^{-2} during night-time, and 3.5 kg d^{-1} m^{-2} during day-time, taking into account a 0.85 efficiency of the drip collector.

CONCLUSIONS

The theoretical models of transient heat transfer and radiant exchange were used to develop a simulation model called 'Simul-CPV' which predicts the amount of water evaporated from a saline source and condensed in non-saline water. The model proved to be a very useful tool in order to simulate the thermal behavior of such a system under different environmental conditions and physical properties of the materials. The best configuration could be easily investigated by carying out feasibility studies for different what-if scenarios. Under the working hypothesis it was calculated that night-time air circulation runs spontaneously and, during a clear sky night, it is capable to evapo-condense 0.6 kg d^{-1} m^{-2} of non-saline water. Day-time working requires a more complicated geometry of the device and the use of two micro air-fans, that can be supplied by the PV panel itself, but the evapo-condensation arises up to 3.5 kg d^{-1} m^{-2}, more than 5 times the nightly production. Further studies are needed to validate the model.

NOMENCLATURE

A surface area of enveloping surfaces (m^2).

k thermal conductivity (W/m°C).

$C_j = A(c_{j-1}\rho_{j-1}\frac{\Delta x_{j-1}}{2} + c_j\rho_j\frac{\Delta x_j}{2})$, mean thermal capacity (J/°C).

ρ: density of the material (kg/m^3).

c: specific heat (J/kg°C).

q: radiant heat flux, positive if incoming (W/m^2).

h: convective heat transfer coefficient (W/m^2°C).

T^t_j : temperature at the j layer calculated at time interval "tΔθ"(°C).

Vol_{build}: volume of the structure (m^3).

V: number of air volume, equal to Vol_{build}, replaced during time interval 'Δθ'.

ε_i: emittance of the surface i.

α_i: absorptance of the surface i.

ρ_i: reflectance of the surface i.

J_i: surface radiosity (W/m^2).

G_i: incident radiant heat (W/m^2).

E_{bi}: black body emission power (W/m^2).

q_i (θ): net radiant heat emitted from the surface 'i' (W/m^2).

F_{i-j} : view factors.

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Tables

Table 1. Physical and thermal properties of materials of the roof and the walls.

Variable	Glass	Tedlar	Poly-etylene	Poly-urethane	Water	Air	Ground
Specific heat (J/kg°C)	830	500	2300	900	4186	1005	1850
Density (kg/m^3)	2300	1800	960	40	1000	1.22	1600
Thermal conductivity (W/m°C)	0.80	0.50	0.50	0.04	0.600	0.25*	2.30
Material thickness (m)	0.006	0.004	0.010	0.050	0.500	0.050	0.500

* Equivalent thermal conductivity for air chamber 0.05 m thick.

Table 2. Dimension of the roof and the walls*.

It [m]	1.7	1.5	1.5	0.8
$\alpha_i = \varepsilon_i$ absorptance=emittance of the surface i	0.7	0.6	0.6	0.6

* As depicted in Figure 1.

Figures

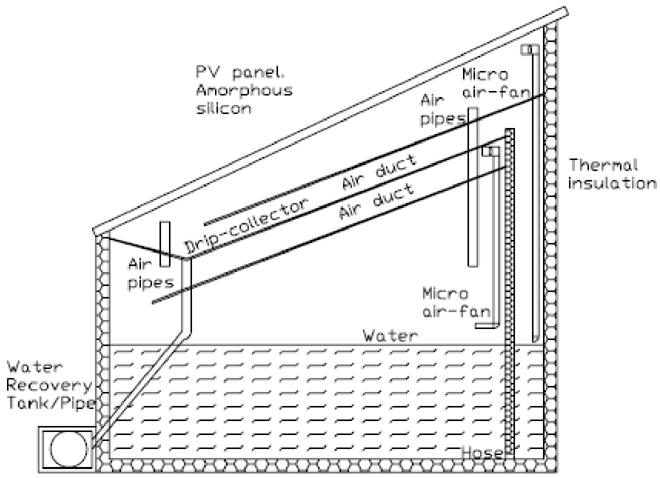
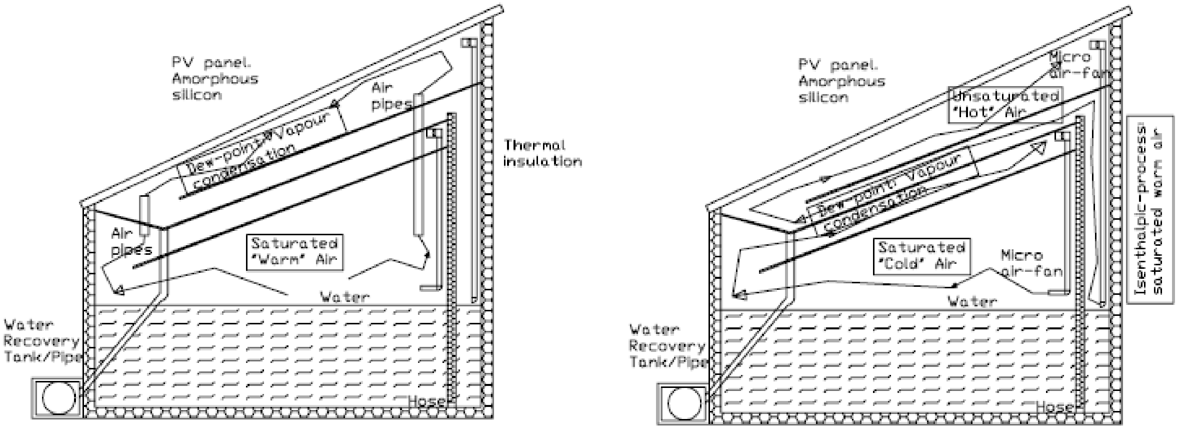


Fig. 1. Plan of the Combi-PV system.



(a) (b)

Fig. 2. Circulation of air inside the condensation-device during night-time (a) and day-time (b).

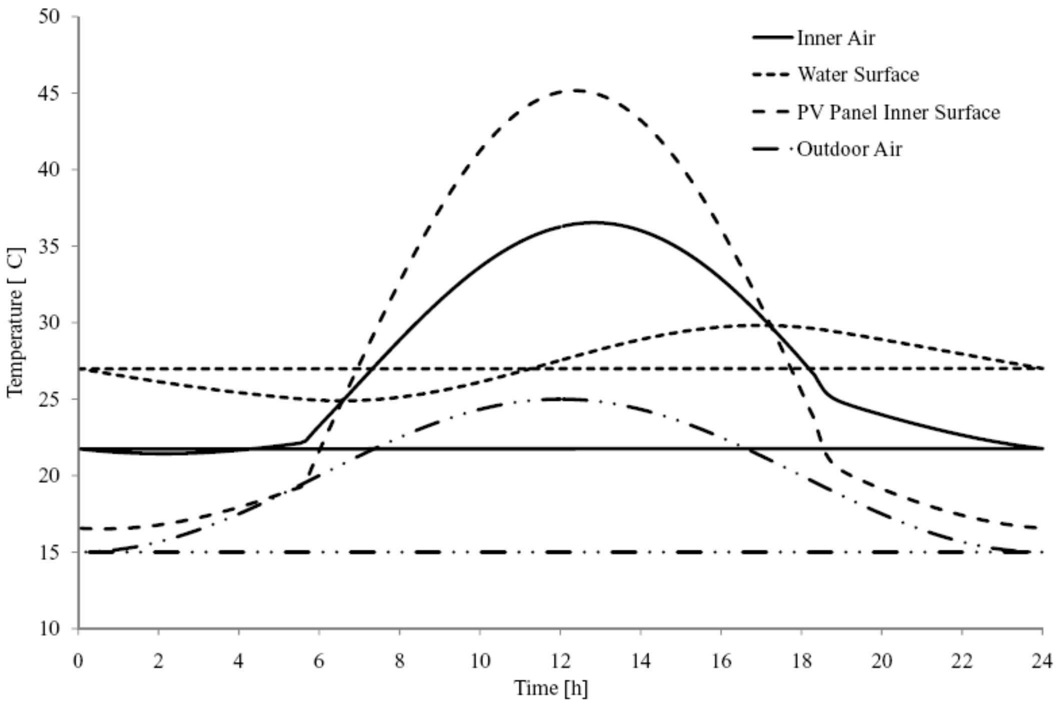


Fig. 3. Simulated temperature of the inner surfaces of the Combi-PV system; Max. Solar Global Radiation = 800 W m^{-2} on a surface perpendicular to sun beams.

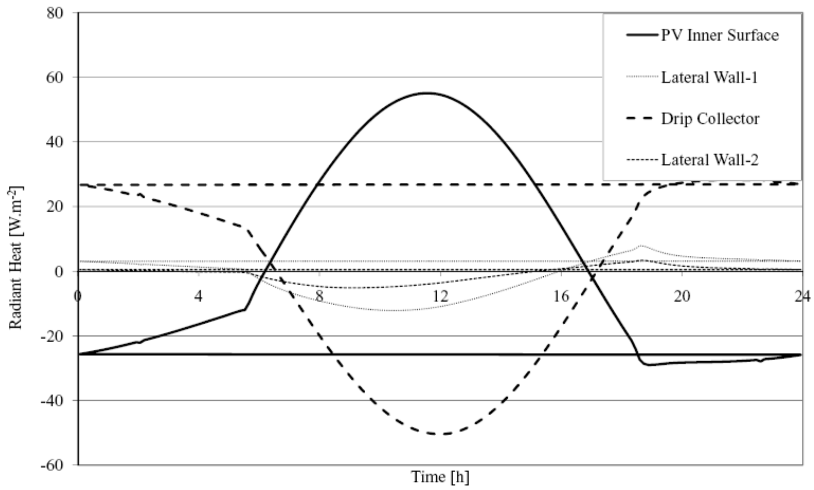


Fig. 4. Calculated radiant heat exchange $q_i(\theta)$ among inner surfaces (positive when outgoing from surface).

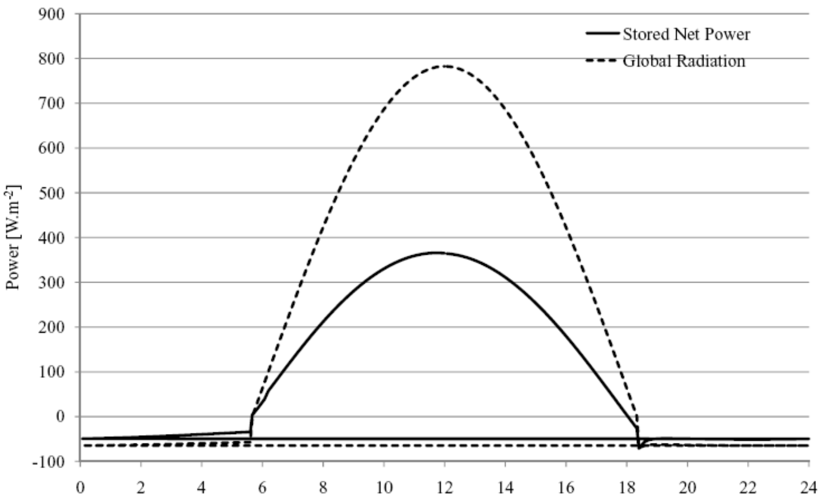


Fig. 5. Calculated net heat power stored inside the system vs. solar global radiation.

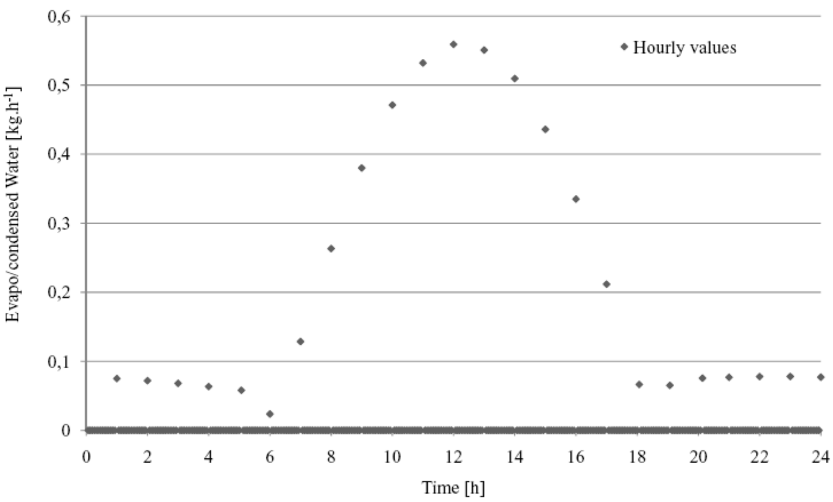


Fig. 6. Calculated evapo/condensed water (hourly amount) inside the Combi-PV device.