Heat and Technology
Vol. 32, n. 1-2, 2014

ENERGY CHARACTERIZATION OF AN UNGLAZED PV/T PANEL FOR THE PRODUCTION OF ELECTRICITY, HEATING AND COOLING.

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SUMMARY

In this work the possibility of using unglazed PV/T solar systems for the daytime production of electricity and heat, and cooling energy during night, has been considered. The diurnal refrigeration increases the electric efficiency of the photovoltaic cells and it makes thermal energy available; while at night the panel exploits the radiative cooling of the sky to produce a cooled water flow rate at a lower temperature than the outside air.

By means of a physical model of a modified PV/T collector implemented in a computer code, the bi-directional distribution of temperature in the absorber plate has been considered. The collection system has been sized in order to favour the production of electrical power. An investigation on the PV/T panel in order to evaluate the variability of the of heat loss coefficient in function of the absorber temperature has been developed. Subsequently, the electric, thermal and cooling efficiencies in function of the main characteristic parameters of the panel have been determined.

INTRODUCTION

The European Directive 2010/31/EC on building energy performance provides significant indications to reduce the energy consumption in buildings, and Italian Legislative Decree no. 28/2011, transposing the RES Directive 2009/28/EC, sets the framework for the exploitation of renewable energy in order to reduce energy consumption and to achieve the Directive objectives by 2020. The constraints to reduce the energy requirements concerns both new buildings and buildings undergoing renovation interventions.

In the present work the possibility of using unglazed PV/T panels for the production of electricity, heating and cooling energy has been considered. During daylight the panel refrigeration increases the electric efficiency of the photovoltaic cells and it makes thermal energy available for DHW production or for heating applications. During nocturnal hours, the panel exploits the infrared radiation to the sky for the production of a cooled water flow-rate at a temperature lower than that of the outdoor air. The lack of a glass cover makes these panels much more dispersant compared to traditional solar thermal collectors, and this helps the production of electricity but reduces the production of thermal energy [1]. The elimination of the glass cover in a PV/T panel, whose main task is to electrically isolate the photovoltaic cells and to give mechanical consistency to the whole device, in the traditional panel is not feasible because the previous two tasks would remain unresolved, but the idea of using the same collection surface to produce electricity, heat and cooling remains valid.

Some authors have widely experimentally investigated the thermal performance achievable by a PV/T panel, also for cooling applications, and tested it with the climatic conditions of Madrid [2]. The results obtained have allowed for the validation of a calculation model that may be used to predict the energy performance of the panel under different climatic conditions. In Madrid the prototype operation produced 51 kWh/m² per year of cooling energy and 205 kWh/m² per year of electricity. The same prototype, simulated by the developed computer code in the hot and humid climate of Shanghai, has shown a produced cooling energy of 55 kWh/m² per year and 142 kWh/m² per year of electricity.

The surfaces used in electric generation by photovoltaic panels are significantly greater than those required for the production of DHW, thus the heating and cooling energy produced by investigated PV/T panel may be used by suitable storage systems for winter heating and summer cooling . The optimal use of PV/T devices requires their continuous cooling during the production of electrical energy [3].

Theoretical evaluation employed by the Florschuetz model [4] for PV/T panels with a glass cover have shown that thermal efficiencies are lower by 7÷12% compared to a simple solar thermal collector, while the comparisons between a PV/T panel and a non-traditional photovoltaic panel have shown that the performances are variable related to the inlet temperature of the cooling flow rate [5].

The PV/T panels exploiting night radiative exchange appear attractive also in combination with the operation of heat pumps, as the electric energy produced may be used to drive the compressor while the heating and cooling energy stored in accumulation systems can be used directly as sources or as integrative energies [6] [7], with a significant growth of heat pump performance coefficients.

MATHEMATICAL MODEL

The physical model of thermal energy delivered in the investigated PV/T is shown schematically by the equivalent electric circuit of Fig. 1. The equation of instantaneous heat balance of the photovoltaic cell, in steady state, with reference to the unit of area, is expressed as:

$$\tau\alpha \cdot IAM \cdot G_T = \tau\alpha \cdot IAM \cdot G_T\, \eta_{pv} + \frac{T_{pv}-T_{ae}}{R_1} + \frac{T_{pv}-T_{sk}}{R_2} + \frac{T_{pv}-T_{ab}}{R_3} \quad (1)$$

with $\tau\alpha$ product of the transmission coefficient of the glass cover (for unglazed panels $\tau=1$) and the cell absorption coefficient, IAM is a parameter to modify the optical properties in function of the incident angle of solar radiation, given by the ratio $\tau\alpha/\tau\alpha_n$ and evaluable by the procedure described in [8], G_T the global solar radiation incident on the collection field, sum of beam, diffuse and reflected components, η_{PV} is the electric efficiency of the photovoltaic cells at the temperature T_{pv}, T_{ab} the absorber plate temperature, T_{ae} and T_{sk} respectively temperatures of the outdoor air and of the sky, R_1 e R_2 convective and radiative resistances on the panel front side, and finally R_3 conductive resistance of the adhesive layer between the photovoltaic cells and the absorber plate.

The equation of the thermal balance of the absorber plate assumes the form:

$$q_{fin} = \frac{T_{pv}-T_{ab}}{R_3} - \frac{T_{ab}-T_{ae}}{R_B} \quad (2)$$

with q_{fin} is the thermal power delivered by conduction through the absorber plate, R_B is the thermal resistance of the rear side between the panel and the outside environment at the temperature T_{ae}.

The determination of the thermal power transferred by the absorber plate to the cooling flow rate requires the evaluation of the temperature field on the same plate. This problem is solved by the same procedure used in traditional plane solar thermal collectors [8]. In the specific case, if D is the channel diameter, W the distance between the axis of two consecutive pipes, the temperature field may be expressed by the following relationship [9]:

$$T_{ab}(x) = \frac{b}{a} + \left(T_{ae} - \frac{b}{a}\right)\frac{\cosh(mx)}{\cosh\left(m\frac{(w-D)}{2}\right)} \quad (3)$$

where:

$$m = \left(\frac{\frac{1}{R_3} + \frac{1}{R_B} - \frac{E}{R_3}}{t \cdot k}\right)^{\frac{1}{2}} \quad (4)$$

$$E = \frac{1}{R_3\left(\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3}\right)} \quad (5)$$

$$\frac{b}{a} = \frac{\tau\alpha \cdot IAM \cdot G_T\left(1-\eta_{pv}\right) + \frac{T_{ae}}{R_1} + \frac{T_{sk}}{R_2} + \frac{T_{ae}}{R_B E}}{\frac{1}{E}\left(\frac{1}{R_3} + \frac{1}{R_B} - \frac{E}{R_3}\right)} \quad (6)$$

with k plate thermal conductivity and t its thickness.

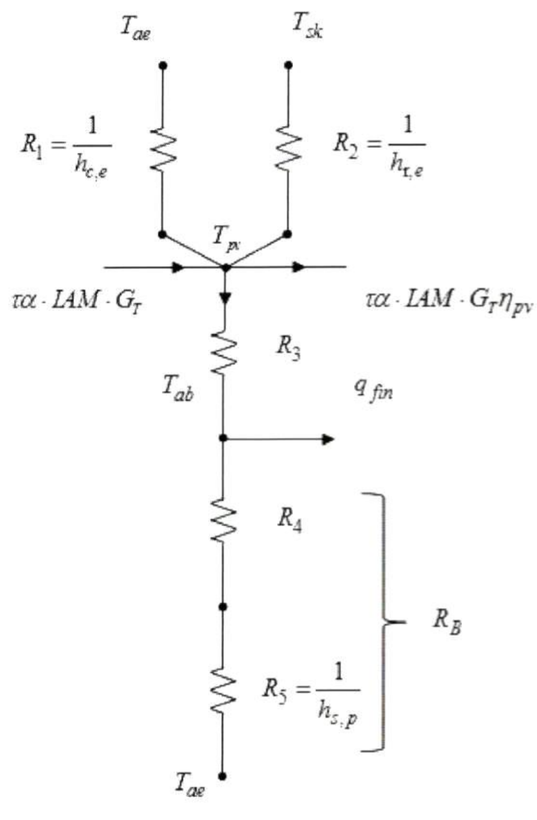

Fig. 1 – Equivalent electric circuit of the investigated PV/T panel.

The temperature profile in x direction allows the calculation of the thermal power that a single plate element, with surface area equal to 1·(W-D)/2, delivers to the channel wall:

$$q_f = -k \cdot t \left.\frac{dT_{ab}}{dx}\right|_{x=\frac{w-D}{2}} \quad (7)$$

$$q_f = -k \cdot t \cdot m\left(\frac{b}{a} - T_{ae}\right)\tanh\left(m\frac{(w-D)}{2}\right) \tag{8}$$

For the calculation of the useful thermal power (W/m) delivered by the plate to the fluid flow rate , the direct thermal exchange mechanisms of the pipe element have to be considered. With reference to the equivalent electric circuit of fig. 2, the useful thermal power is given by the relation:

$$q_u = 2\,q_{fin} + \frac{D\left(T_{pv} - T_B\right)}{R_3} - \frac{D\left(T_B - T_{ae}\right)}{R_B} \tag{9}$$

$$q_u = \frac{T_B - T_f}{R_6 + R_7} \tag{10}$$

where T_B is the plate temperature in correspondence of the channel, T_f is the fluid temperature, R_6 is the thermal resistance between the plate and the inner pipe wall and R_7 the convective resistance between wall channel and fluid.

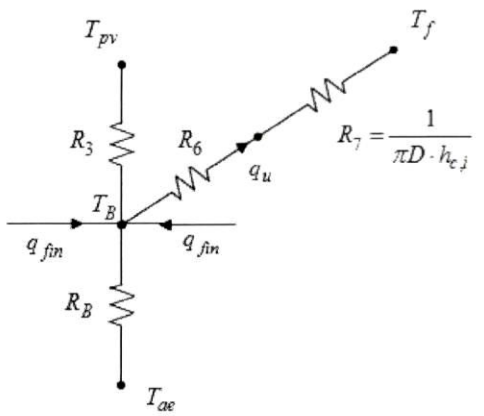

Fig. 2 – Equivalent electric circuit of the plate element and the channel located below.

Considering equations (1), (8), (9) and (10), the useful thermal power q_u assumes the form:

$$q_u = \frac{K}{\vartheta}T_f + \frac{\varepsilon}{\vartheta} \tag{11}$$

with:

$$K = -D \cdot E\left(\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_B E}\right) - 2k \cdot t \cdot m \cdot \tanh\left(m\frac{(w-D)}{2}\right) \tag{12}$$

$$\vartheta = 1 + D \cdot E\left(R_6 + R_7\right)\left(\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_B E}\right) + 2k \cdot t \cdot m \cdot \tanh\left(m\frac{(w-D)}{2}\right)\left(R_6 + R_7\right) \tag{13}$$

$$\begin{aligned}\varepsilon = {} & D \cdot E\left[\tau\alpha \cdot IAM \cdot G_T\left(1 - \eta_{pv}\right) + \frac{T_{ae}}{R_1} + \frac{T_{sk}}{R_2} + \frac{T_{ae}}{R_B E}\right] + \\ & +2k \cdot t \cdot m \cdot \tanh\left(m\frac{(w-D)}{2}\right)\frac{\left[\tau\alpha \cdot IAM \cdot G_T\left(1 - \eta_{pv}\right) + \frac{T_{ae}}{R_1} + \frac{T_{sk}}{R_2} + \frac{T_{ae}}{R_B E}\right]}{\frac{1}{E}\left(\frac{1}{R_3} + \frac{1}{R_B} - \frac{E}{R_3}\right)}\end{aligned} \tag{14}$$

If L is the channel length and n their number, the global useful thermal power Q_u, is equal to:

$$Q_u = q_u \cdot L \cdot n \tag{15}$$

For the calculation of the fluid temperature variation law in the channels, the equation of fluid flow rate heating has to be integrated:

$$\dot{m} \cdot c_p \cdot dT_f = n \cdot q_u \cdot dy$$

with q_u given by Eq. (11). The obtained final expression is:

$$T_f(y) = \left(T_{f,i} + \frac{\varepsilon}{K}\right)\exp\left(\frac{n \cdot K}{\dot{m} \cdot c_p\,\vartheta}y\right) - \frac{\varepsilon}{K} \tag{16}$$

Imposing $y=L$ in the prior relationship, the outlet flow rate temperature can be obtained:

$$T_{f,u} = \left(T_{f,i} + \frac{\varepsilon}{K}\right)\exp\left(\frac{n \cdot K}{\dot{m} \cdot c_p\,\vartheta}L\right) - \frac{\varepsilon}{K} \tag{17}$$

and moreover Eq. (17) allows the evaluation of the mean fluid temperature in the channels:

$$\begin{aligned}\overline{T}_f &= \frac{1}{L}\int_0^L T_f(y)\,dy \\ \overline{T}_f &= \frac{\left(T_{f,i} + \frac{\varepsilon}{K}\right)}{\frac{n \cdot K \cdot E}{\dot{m} \cdot c_p \vartheta}}\left[\exp\left(\frac{n \cdot K \cdot E}{\dot{m} \cdot c_p \vartheta}\right) - 1\right] - \frac{\varepsilon}{K}\end{aligned} \tag{18}$$

The mean plate temperature is evaluable by the relation:

$$\overline{T}_{ab} = \frac{D \cdot \overline{T}_B + (w - D)\overline{T}_{fin}}{w} \tag{19}$$

Finally, the mean cells temperature $\overline{T}_{pv}$ can be obtained by Eq. (1) imposing $T_{ab} = \overline{T}_{ab}$

The mathematical model is solved by an iterative procedure starting from an initial value of the mean cells temperature and from the following value verification by Eq. (1). The knowledge of the mean cells temperature $\overline{T}_{pv}$, of the absorber plate temperature $\overline{T}_{ab}$, of the fluid $\overline{T}_f$, of electric power P_e, of the thermal power $Q_{u,t}$, provided in diurnal hours, and the cooling power $Q_{u,f}$ provided in the nocturnal hours, allows for the energy characterization of the investigated PV/T panel. The reference equations are:

- Electric power and electric efficiency:

$$P_{el} = \tau\alpha \cdot IAM \cdot G_T A \cdot \overline{\eta}_{pv} \tag{20}$$

$$\bar{\eta}_{pv} = \eta_R\left[1-\beta\left(\bar{T}_{pv}-T_R\right)\right] \quad (21)$$

where the dependence from incident solar radiation has been considered negligible;

- Thermal power and thermal efficiency:

$$Q_{u,t} = A\left[\frac{1/U_a}{R_3+1/U_a}\tau\alpha\cdot IAM\cdot G_T\left(1-\bar{\eta}_{pv}\right)-U_L\left(\bar{T}_{ab}-T_{ae}\right)\right] \quad (22)$$

$$\eta_{u,t} = \frac{1/U_a}{R_3+1/U_a}\tau\alpha\cdot IAM\cdot\left(1-\bar{\eta}_{pv}\right)-\frac{U_L\left(\bar{T}_{ab}-T_{ae}\right)}{G_T} \quad (23)$$

where U_L is the collector overall loss coefficient, defined by the sum of front heat loss coefficient U_a and the rear heat loss coefficient U_p, both defined with reference to the temperature difference between the absorber plate and the outdoor temperature.

In Eq.(22) the available thermal power at the absorber has been obtained by modifying the available thermal power by the parameter $1/U_a/(R_3+1/U_a)$ after the photovoltaic conversion, where R_3 is the thermal resistance between the cells and the absorber plate.

If the thermal removal factor $F_{R,t}$ is considered:

$$F_{R,t} = \frac{\dot{m}c_p\left(T_{f,u}-T_{f,i}\right)}{A\left[\frac{1/U_a}{R_3+1/U_a}\tau\alpha\cdot IAM\cdot G_T\left(1-\bar{\eta}_{pv}\right)-U_L\left(T_{f,i}-T_{ae}\right)\right]} \quad (24)$$

where $T_{f,u}$ is given by Eq. (17), the thermal power $Q_{u,t}$ can be evaluated by the relation:

$$Q_{u,t} = A\cdot F_{R,t}\left[\frac{1/U_a}{R_3+1/U_a}\tau\alpha\cdot IAM\cdot G_T\left(1-\bar{\eta}_{pv}\right)-U_L\left(T_{f,i}-T_{ae}\right)\right] \quad (25)$$

and the thermal efficiency as:

$$\eta_{u,t} = F_{R,t}\left[\frac{1/U_a}{R_3+1/U_a}\tau\alpha\cdot IAM\left(1-\bar{\eta}_{pv}\right)-\frac{U_L\left(T_{f,i}-T_{ae}\right)}{G_T}\right] \quad (26)$$

- Cooling power and cooling efficiency:

The resolution of the equivalent electric circuit of Fig. 1 in absence of solar radiation, considering that the photovoltaic cell loses thermal power towards sky, and receives thermal power from the front side by convection and on the rear side by liminar exchange using an opportune global thermal exchange coefficient, and moreover it obtains thermal power by the cooled flow rate flowing in the channels, leads to the relation:

$$Q_{u,f} = A\cdot E\left[\varepsilon_{pv}\,\sigma\left(\bar{T}_{ab}^4-T_{sk}^4\right)-U_L^*\left(T_{ae}-\bar{T}_{ab}\right)\right] \quad (27)$$

where $U^*_L=(1/R_1+1/R_B)$ does not consider the radiative loss of the front side collector, ε_{pv} is the cell long-wave emissivity coefficient and E is a parameter given by Eq. (5). The latter parameter modifies the thermal power lost from the absorber plate by radiative exchange and the thermal power received by convention from outdoor air, because the absorber temperature is greater than the cell temperature.

The cooling efficiency has been evaluated with reference to the maximum thermal power delivered to the sky by radiative exchange:

$$\eta_{u,f} = E\left[\varepsilon_{pv}-\frac{U_L^*\left(T_{ae}-\bar{T}_{ab}\right)}{\sigma\left(\bar{T}_{ab}^4-T_{sk}^4\right)}\right] \quad (28)$$

The calculation model may be used to optimize the geometrical, thermal and hydraulic parameters of the absorber plate cooling device, to improve the electric efficiency, to characterize the PV/T collector by electric, thermal and cooling efficiencies and to estimate the daily, monthly and yearly achievable energies.

CHARACTERISTIC CURVES OF THE INVESTIGATED PV/T PANEL

The evaluations have been carried out supposing the use of two different technologies of photovoltaic cells, the first represented by polycrystalline silicon with nominal efficiency η_R=12%, and the second by amorphous silicon cells with nominal efficiency η_R=6%. Considering the influence of incident solar radiation as negligible, the characteristic curves are reported in function of the cells temperature only, and are shown in fig.3.

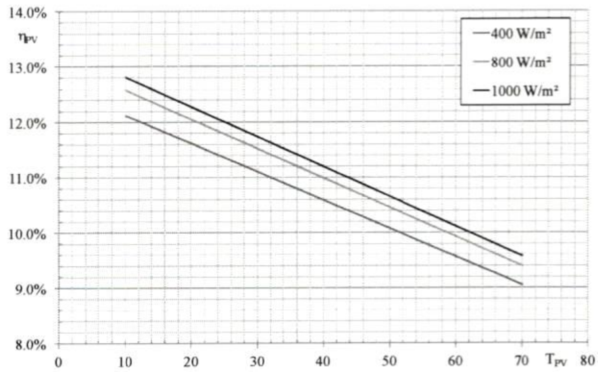

Fig. 3 – Electric efficiency trend of a panel using polycrystalline technology in function of the cell temperature and for three values of incident solar radiation .

For thermal evaluations, a PV/T collector with dimension 2×1 m that uses 12 parallel pipes with an internal diameter of 12.7 mm and located on its rear side with a pitch of a 16.6 cm, has been considered. The cooling/cooled flow rate is 0.245 kg/s to assure a turbulent hydraulic flow. This configuration provides a mean yearly electric efficiency of 12.35%, using water flow rate with inlet temperature of about 42 °C; with reference to the PV panel without a cooling device, the augment of electric efficiency is equal to 2 percentage points. These results were obtained by the measured data of solar radiation, long-wave radiant exchange and outside air temperature collected by the weather station located at the University of Calabria (southern Italy, Lat. 39.3°N).

The collector characterization has been carried out by investigation on the variability of the collector global loss coefficient U_L, considering the collector heat loss coefficient on the rear side U_p constant and equal to 1.25 W/m²K. In fig. 4 the trends of the collector front heat loss coefficient U_a in function of the absorber temperature and outdoor air temperature, are reported. The U_a coefficients have been calculated supposing a convective heat transfer coefficient constant and equal to 14 W/m²K, considering three different conditions of electric generation: no electric production, electric production by amorphous silicon cells with nominal efficiency η_{pv}=6% and electric production by polycrystalline silicon cells with nominal efficiency η_{pv}=12%. The effect of the electric efficiency variability on the panel front heat loss coefficient is negligible, while the dependence on outdoor air temperature is relevant with important growth when the absorber plate temperature is close to the outdoor air temperature. The maximum value of the absorber plate temperature is reached in the absence of the cooling flow rate. For an absorber plate temperature varying between 30°C and 60°C, the panel front heat loss coefficient is about 20 W/m²K when the outside air temperature is set to 0°C, and the absorber temperature varies between 28 W/m²K and 23 W/m²K setting outdoor air temperature to 15 °C. If the absorber plate temperature is40 °C and the outdoor air temperature is set to 30°C, the panel front heat loss coefficient is about 39 W/m²K.

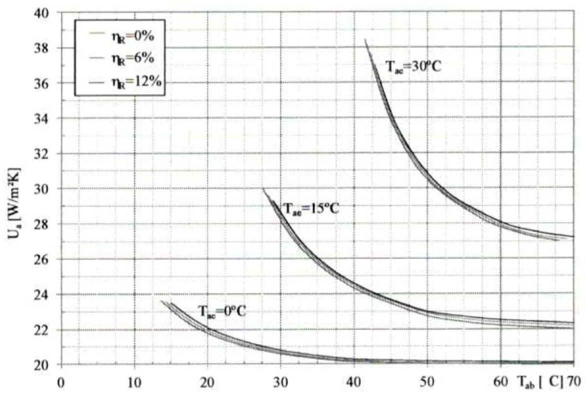

Fig. 4 – Trend of the investigated PV/T front heat loss coefficient in function of the absorber plate temperature and for three different values of the electric nominal efficiency η_R.

In Fig. 5 the trends of the PV/T thermal efficiency evaluated by Eq. (26) in function of the loss parameter $U_L \cdot (T_{f,i}-T_{ae})/G_T$, and for three values of outdoor air temperature and for three values of electric nominal efficiency, are reported. The reduction of the thermal efficiency with an increase in electric efficiency may be assumed constant varying the loss parameter and outdoor air temperature, and it is equal about to 7 percentage points considering the collector operation without electric generation and a PV/T panel exploiting cells with polycrystalline technology.

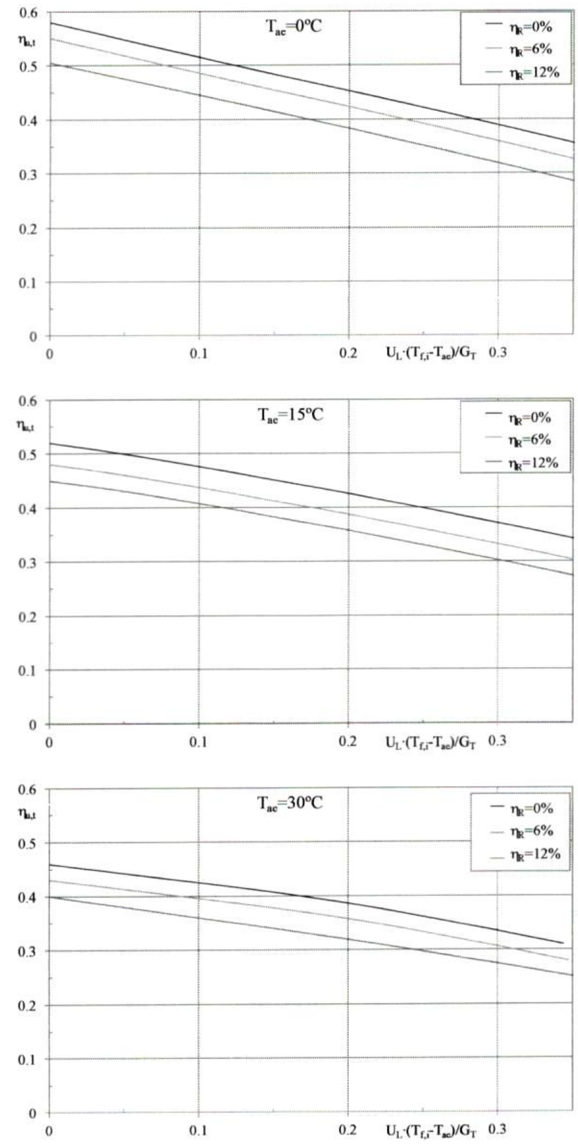

Fig. 5 – Trend of the thermal efficiency in function of the loss parameter $U_L \cdot (T_{f,i}-T_{ae})/G_T$ and for three values of electric nominal efficiency and for three values of outdoor air temperature.

For the investigated PV/T collector the thermal efficiency decreases with the outdoor air temperature growth, since the panel front heat loss coefficient is also affected by an increase.

Finally, the same PV/T collector has been considered as a device which is able to exploit the night radiative exchange with the sky to produce cold water flow rate. In Fig. 6 the cooling efficiency trend , calculated by Eq. (28) in function of the loss parameter $U_L^* \cdot (T_{ae}-\overline{T}_{ab})/\sigma \cdot (\overline{T}_{ab}^4-T_{sk}^4)$, is reported. The U_L^* value has been set to 15.25 W/m²K, the front panel convective heat loss coefficient set equal to 14 W/m²K and the rear panel global heat loss coefficient set to 1.25 W/m²K .

Fig. 6 highlights that the cooling efficiency is almost a linear function of the thermal power loss fraction.

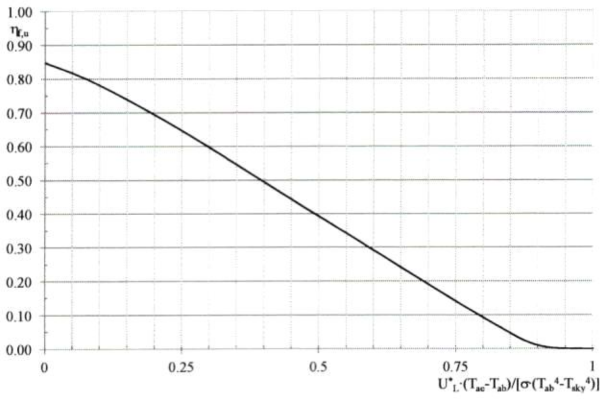

Fig. 6 –Cooling efficiency trend in function of the loss parameter

CONCLUSIONS

A thermal characterization of unglazed PV/T panels to produce electric, thermal and cooling energies has been carried out. By a physical model developed in a computer code, the main thermal, geometrical and hydraulic parameters to describe the investigated PV/T operation have been optimized.

The studied collector has been characterized by the curves of its front heat loss coefficient, of the electric efficiency, thermal and cooling efficiencies.

The study has highlighted that the front panel heat loss coefficient is strongly influenced by the outdoor air temperature. Imposing an absorber plate temperature equal to 50 °C, the augment of the outdoor air temperature from 15°C to 30°C provides a heat loss coefficient growth of about 30%. This result shows that the front panel heat loss coefficient is not constant during the daily solar energy collection.

At night, the temperature difference between the absorber plate and outdoor air is smaller, with more significant consequent variations of the front heat loss coefficient. Moreover, the augment of the front panel heat loss coefficient provides lower values of thermal efficiency.

The trend of the cooling efficiency may be assumed almost linear in function of the lost thermal power fraction. Interesting values of the cooling efficiency can be obtained if the value of the loss parameter is lower than 0.5. Energy evaluations carried out by climatic data collected in the University of Calabria weather station have shown that the mean yearly cooling efficiency is about 25% .

REFERENCES

1. H.A. Zondag, Flat-plate PV-Thermal collectors and systems: A review, *Renewable and Sustainable Energy Reviews*, vol. 12, pp. 891-959, 2008.
2. U. Eicker, A. Dalibard, Photovoltaic–thermal collectors for night radiative cooling of buildings, *Solar Energy*, Vol. 85, pp. 1322-1335, 2011
3. H. Saitoh, Y. Hamada, H. Kubota, M. Nakamura, K. Ochifuji, S. Yokoyama, K. Nagano, Field experiments and analyses on a hybrid solar collector, *Applied Thermal Engineering*, Vol. 23, pp. 2089-2105, 2003
4. Florschuetz L.W., Extension of the Hottel–Whillier model to the analysis of combined photovoltaic/thermal flat plate collectors. *Solar Energy* 22, 361–366 (1979)
5. M. Cucumo, D. Cucumo, A. De Rosa, V. Ferraro, D. Kaliakatsos, V. Marinelli, Analisi Teorica di un Collettore Solare Cogenerativo PV/T a Liquido, Atti del 64° Congresso nazionale ATI, L'Aquila, 8-11 Settembre 2009
6. A. Georgiev, Testing solar collectors as an energy source for a heat pump, *Renewable Energy*, Vol. 33, pp. 832-838, 2008
7. G. Fang, H. Hu, X. Liu, Experimental investigation on the photovoltaic–thermal solar heat pump air-conditioning system on water-heating mode, *Experimental Thermal and Fluid Science*, Vol. 34, pp. 736-743, 2010
8. G. Oliveti, N. Arcuri, R. Bruno, M. De Simone, L. Marletta, G. Evola, Tecnologie solari negli edifici esistenti, EPC editions, 2013.
9. AA.VV., TESS libraries for TRNSYS vs. 17, Thermal Energy System Specialist, Madison, Wisconsin, USA, www.tess-inc.com/trnsys