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SOLAR ELECTRICITY GENERATION

THEORETICAL AND EXPERIMENTAL ANALYSIS OF TWO PASSIVE COOLING SYSTEMS FOR THE SOLAR CELLS WITH CONCENTRATION

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ABSTRACT

The thermal behaviour of a finned massive core used like passive cooling device has been, theoretically and experimentally, compared with the one of a finned heat pipe. The results show that the almost total reduction of the thermal resistance of the core causes a drop of the solar cell temperature and, consequently, an increase of its efficiency up to 30%, depending from geometrical and thermophysical parameters.

KEYWORDS

Heat pipe;passive cooling system; solar cell cooling.

INTRODUCTION

It is well known that the photovoltaic cells, impingued by radiative flux, produce electrical and thermal energy. The coupling of passive cooling systems to the cells can be sufficient, if the concentrated radiative flux is less than 100 kw/m^2, to dissipate the developed heat and to maintain the working temperature of the cell at values lower than 360 K. A simple passive cooling device is a metallic full core, finned on the side-wall, with a constant cross section of suitable shape, whereas the top is the face of the solar cell. The cooling efficiency of this system is reduced by the thermal resistance of the core, the higher the less is the cross section surface. The study of this effect, reported in this paper, has been performed, in former stage, theoretically, by two mathematical models of the device above described, different between themselves only by the bounded or infinite conductance of the core and indicated, henceforth, respectively HN and HT. In a latter stage, two mild steel works like passive cooling device have been built, having the former a massive core and the latter a heat pipe, working the face of the cell as evaporator and the rest of the surface as condensator.

MATHEMATICAL MODELS

With reference to the Fig. 1 we assume the following hypotheses:
-- the cross section of the core is equal to the cell surface;
-- no thermal gradient across the lens and between cell and its face;

-- no conductive heat flux between truncated cone and cell;
-- temperatures of the external ambient air T_A, of the truncated cone surface T_S, of the enclosure air T_E and of the cell T_C are uniform, furthermore $T_S = T_E$;
-- same convective coefficients for core and fins;
-- the only axial gradient can be different from zero in the core, (when it is zero, the HT model is described).

The heat balance between two cross sections distant dz gives, in steady-state conditions,:

$$d^2T/dz^2 = (T - T_A)2U/(\lambda R) \qquad (1)$$

where $U = (h_r + h)(2\pi R - ns + 2n\varepsilon L)/(2\pi R)$

with s,L,ε and n respectively thickness,length, efficiency and number of the fins, and with R and λ the radius of the core cross section and the thermal conductivity coefficient of core and fins.

The boundary conditions are:

$$dT/dz = Q_D(T) \qquad z = 0 \qquad (2)$$

$$dT/dz = 0 \qquad z = H \qquad (3)$$

where H is the height of the fins.

The thermal power $Q_D(T)$ dissipated by the cooling device is dependent from the cell temperature and is unknown, but at z = 0 we have the global heat balance given by $Q_D(T) = Q_I - Q_W(T) - Q_E$ (4)

where the terms at right-hand are respectively the radiative power impinguing the cell, the electrical power obtained and the thermal power transferred to the enclosure. This last is equal, in steady conditions, to the thermal power dissipated by the enclosure, so that we have

$$U_C(T_C - T_S)S_C = U_S(T_S - T_A)S_S \qquad (5)$$

where S is a exchange surface, U is an overall heat-transfer coefficient, and we can calculate Q_E if the cell temperature is known.

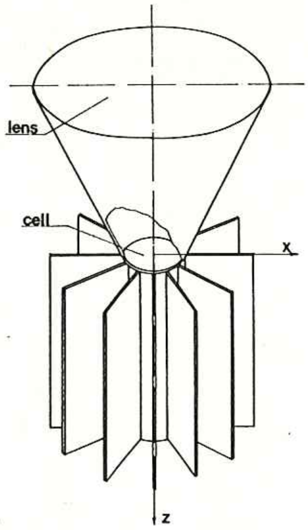

Fig. 1. Design of the device

The relationship, used for the coefficients of free convection between external ambient air and fins is taken from Van Den Pool and Tierney (1977), whereas for the forced convection the relationship given from Giulianini, Cocchi and Vaccari (1966) is used. The radiative exchange coefficient h_r is given by $h_r = e\sigma_o(1 - \sin(\phi/2))(T + T_A)(T^2 + T_A^2)$

with σ_o is the Stefan-Boltzmann constant, e is the wall emissivity and ϕ is the dihedron between the fins.

The equation (1) with the boundary conditions (3,4) has been solved by a standard finite difference method. A first guess of the temperature's values permit to calculate h, h_r and Q_D so that these can be utilized to determine new values of temperature until the max difference between the old and the new solutions is less than .001.

In the case of HT model, the thermal balance is very simple and gives:

$$Q_I = Q_D + Q_E \; ; \qquad Q_D = H(T_C - T_A)(2n\varepsilon L + 2\pi R - ns)(h_r + h)$$

and the cell temperature is obtained by standard algorithm.

Numerical Results

It is immediate that the thermal behaviour of two models is dependent by numerous parameters, but, here, for the sake of brevity the Figures from 2 to 4 show the thermal behaviour of the system and the cell efficiency compared with only some parameters.

Unfortunately, the HT model describes only an ideal heat pipe since the heat flux, from the evaporator to the condensator, passes through several thermal resistances connected in series. A semi-empirical evaluation of their sum, that is to say the global thermal resistance, gives $R_g = 0.18$ K/w. The Figure 5 show this effect, with

effects on the cell temperature;
-- a length of the fins less than 0.1 m is recommended for both models;
-- the ratio height/diameter for the core must be less than 5 in HN model;
-- the almost total reduction of thr thermal resistance of the core causes a drop in temperature of the cell and, consequently, an incr-ase of its efficiency up to 30% when the radiative flux is 100 kw/m^2;
-- the presence of a light wind (v = 1 m/s), as in the moutains, permit a solar cell system with a concentration ratio higher than 100, using heat pipe as core;
-- the weight of the device is an important parameter to design a solar photovoltaic plant

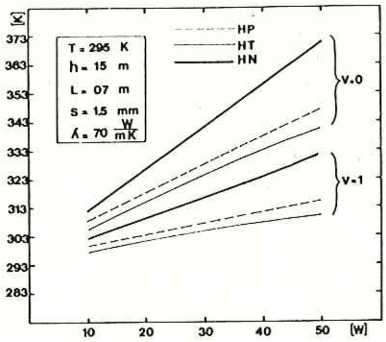

Fig. 5. T_C vs. Q_I

EXPERIMENTAL CHECKING

Utilizing the previous results, two cooling devices have been built, as shown in Fig. 6, the former having a full massive core, the latter a heat pipe as a core, whereas the fins were welded in both cases. The cylindrical tube of the heat pipe is a mild steel work, whereas the wick structure is constituted by four layers of stainless steel net of about 80 meshes/cm and held toghether by spot weldings. Bidistilled water, as working fluid, was, in a working time of five minutes, the cause of a rapid generation of corrosion with working end. In place of the water, acetone was used. The photovoltaic cells was nicely supplied from the Laboratory L.A.M.E.L.- C.N.R. of Bologna. The radiative flux was obtained by a tungsten lamp of 500 W and collimated by an optical system at low efficiency, so that it was possible to obtain a value of 30 as a maximum concentration ratio.
The Figure 7 show[2] the results of several tests of the two passive cooling devices toghether with the corresponding theoretical results, indicated as HN_t and HP_t, these last only for vertical position of the devices. It is immediate that the central belly of the slopes depending by bad working of the fins in horizontal position of the devices. It is worthy to note the pleasant agreement among the experimental data and the theoretical results, remembering that no welding thermal resistance has been accounted in the mathematical models.

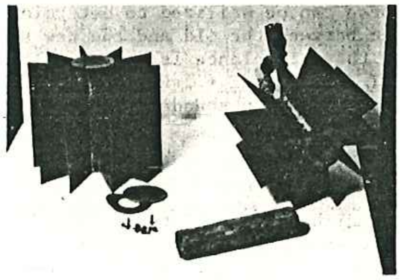

Fig. 6. Experimental devices

2 The value of Q_I is wrong: Q_I = 12 W

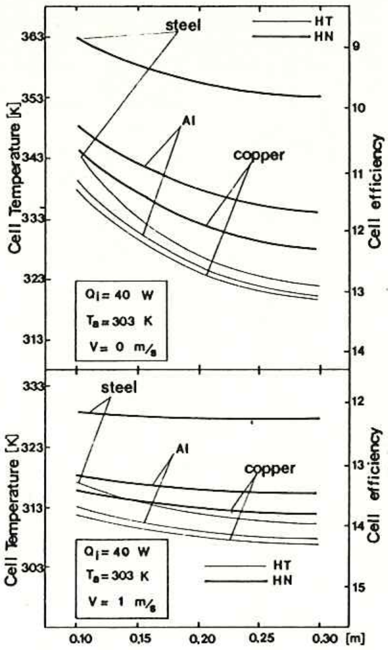

Fig. 2. T_C and ε vs. the fins length

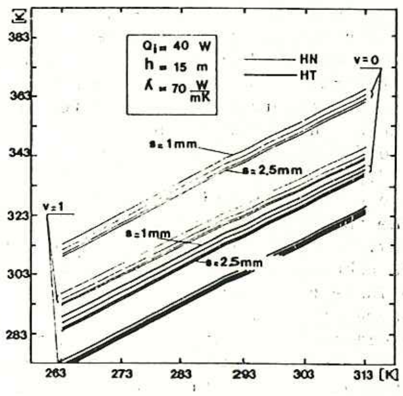

Fig. 3. T_C vs. T_A at different values of thickness of the fins

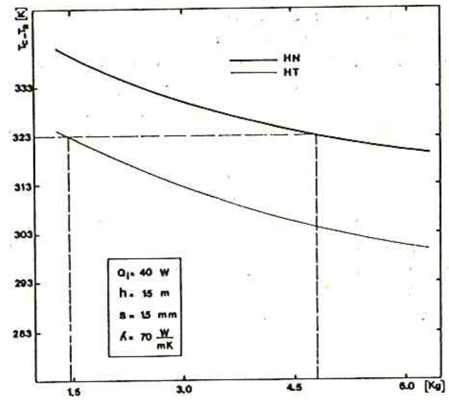

Fig. 4. Influence of the fins number in terms of weight on the difference $(T_C - T_A)$[1]

the slopes indicated as HP.

By these results it is possible to adfirm that

--the thermal power dissipated by the enclosure can be disregarded;

--the thickening of the fins causes a high rise in weight of the device with scarce

[1] The temperature scale is wrong: $(T_C - T_A)_{true} = (T_C - T_A)_{wrong} - 273$

Since the acetone cannot, in a wick structure, rise up, for capillarity,more than 6 cm, increasing the inclination of the device, the heat pipe does not work more. In fact, the greater is the contents of acetone the greater is the inclination angle of working stop. On the contrary, when the heat pipe is assisted by the gravity, in our case the evaporator turned upside-down, better results are obtained reducing the contents of the working fluid.

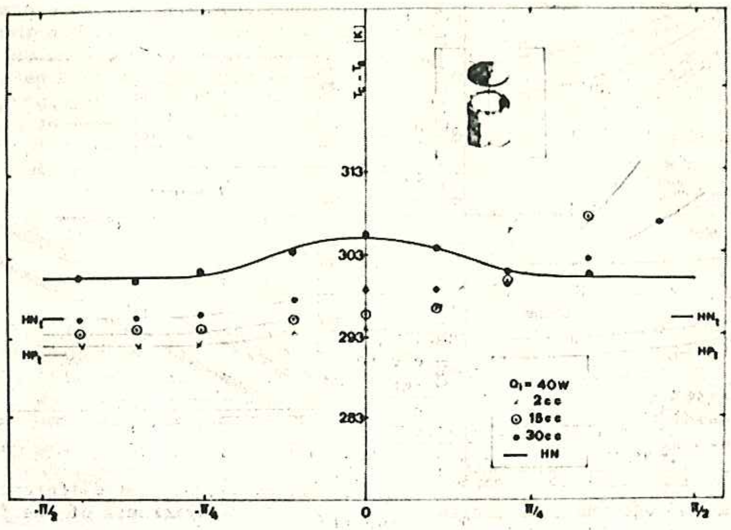

Fig. 7. The difference ($T_C - T_A$) vs. iclination angle

CONCLUSIONS

All results point out that the use of a heat pipe is a powerful tool in a passive cooling system and in particular it is to note:

-- all structure is very light;

-- itis possible to utilize concentration ratios greater than 100;

-- other designes,suggested by the previous tests could give better performances.

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