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Theoretical and experimental analysis of two passive cooling systems for the solar cells without concentration / Dumas, Antonio; F., Draghetti. - STAMPA. - 4:(1981), pp. 2908-2912. ( Internattional Solar Energy Society Congress 1981 Brighton (UK) 23-28 August 1981).

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# SOLAR WORLD FORUM

*Proceedings of the International Solar  
Energy Society Congress*

*Brighton, England, 23-28 August 1981*

Edited by

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King's College  
London

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London

Volume Four

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**PERGAMON PRESS**

OXFORD · NEW YORK · TORONTO · SYDNEY · PARIS · FRANKFURT



U.K.	Pergamon Press Ltd., Headington Hill Hall, Oxford OX3 0BW, England
U.S.A.	Pergamon Press Inc., Maxwell House, Fairview Park, Elmsford, New York 10523, U.S.A.
CANADA	Pergamon Press Canada Ltd., Suite 104, 150 Consumers Road, Willowdale, Ontario M2J 1P9, Canada
AUSTRALIA	Pergamon Press (Aust.) Pty. Ltd., P.O. Box 544, Potts Point, N.S.W. 2011, Australia
FRANCE	Pergamon Press SARL, 24 rue des Ecoles, 75240 Paris, Cedex 05, France
FEDERAL REPUBLIC OF GERMANY	Pergamon Press GmbH, 6242 Kronberg-Taunus, Hammerweg 6, Federal Republic of Germany

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First edition 1982

**Library of Congress Cataloging in Publication Data**

Main entry under title:

Solar world forum.

1. Solar energy—Congresses. I. Hall, D. O. (David Oakley) II. Morton, June. III. International Solar Energy Society.

TJ810.S725 1982 621.47 82-323

AACR2

**British Library Cataloguing in Publication Data**

International Solar Energy Society. Conference (1981:Brighton)

Solar world forum

1. Solar energy—Congresses

I. Title II. Hall, David O

III. Morton, June

621.47 TJ810

ISBN 0-08-026730-0

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England

Maxwell House, Fairview Park,  
1523, U.S.A.

Energy Ltd., Suite 104,  
Willowdale, Ontario M2J 1P9, Canada

Energy Pty. Ltd., P.O. Box 544,  
Sydney, Australia

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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, impinged 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  $100 \text{ kw/m}^2$ , 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:

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

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

with  $s, L, \epsilon$  and  $n$  respectively thickness, length, efficiency and number of the fins, and with  $R$  and the radius of the core cross section and the thermal conductivity coefficient of core and fins. The boundary conditions are:

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

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

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 impinging 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

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

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.

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  $h_r = \epsilon\sigma_0(1 - \sin(\phi/2))(T + T_A)(T^2 + T_A^2)$

with  $\sigma_0$  is the Stefan-Boltzmann constant,  $\epsilon$  is the wall emissivity and  $\phi$  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:

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

and the cell temperature is obtained by standard algorithm.

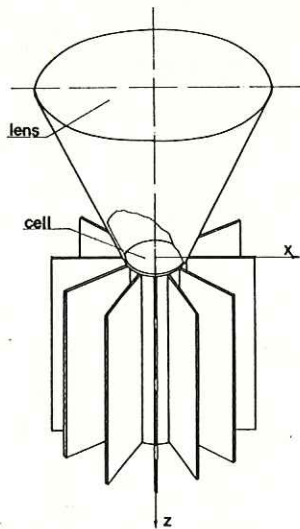


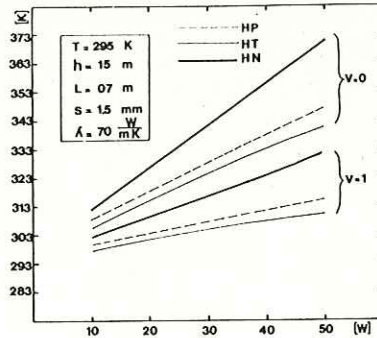
Fig. 1. Design of the device

#### 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 the thermal resistance of the core causes a drop in temperature of the cell and, consequently, an increase of its efficiency up to 30% when the radiative flux is  $100 \text{ kw/m}^2$ ;  
 -- the presence of a light wind ( $v = 1 \text{ m/s}$ ), as in the mountains, 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 together 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<sup>2</sup> the results of several tests of the two passive cooling devices together 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.

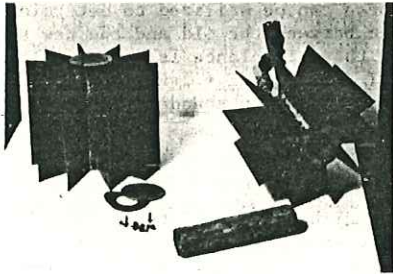


Fig. 6. Experimental devices

<sup>2</sup> The value of  $Q_I$  is wrong:  $Q_I = 12 \text{ W}$

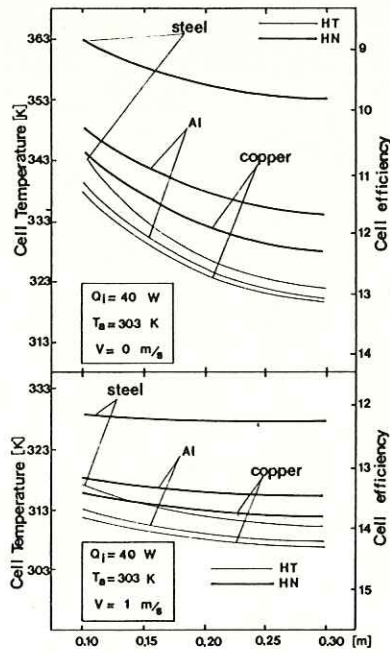


Fig. 2.  $T_C$  and  $\epsilon$  vs. the fins length

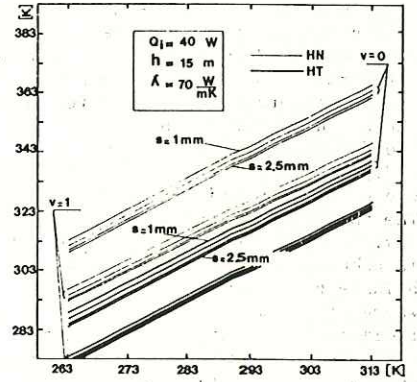


Fig. 3.  $T_C$  vs.  $T_A$  at different values of thickness of the fins

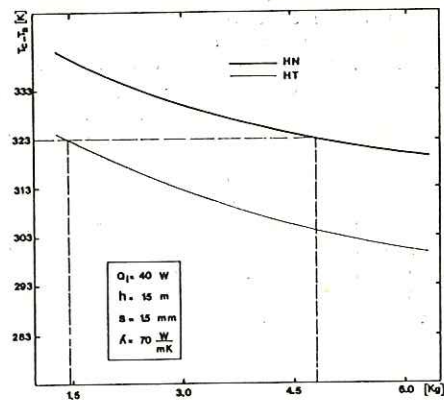


Fig. 4. Influence of the fins number in terms of weight on the difference  $(T_C - T_A)^1$

the slopes indicated as HP.  
 By these results it is possible to affirm 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

<sup>1</sup> 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.

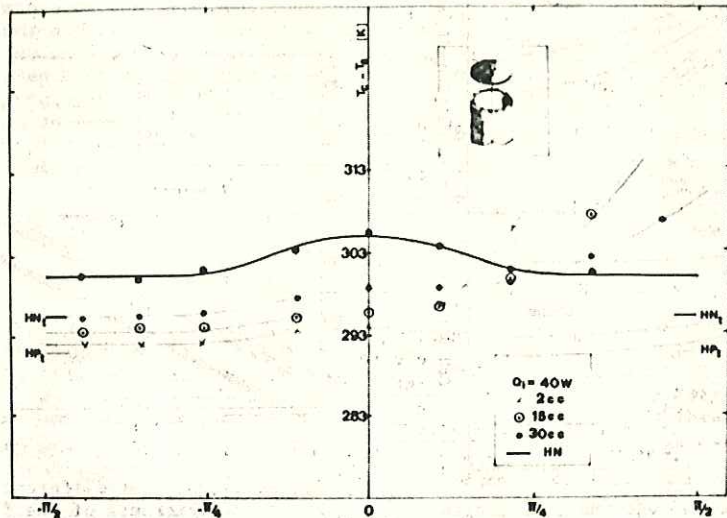


Fig. 7. The difference  $(T_C - T_A)$  vs. inclination angle

#### 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;
- it is possible to utilize concentration ratios greater than 100;
- other designs, suggested by the previous tests could give better performances.

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