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SOLAR WORLD FORUM

*Proceedings of the International Solar
Energy Society Congress*

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Edited by

DAVID O. HALL
King's College
London

JUNE MORTON
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TABLE OF CONTENTS

Volume Four

SOLAR ELECTRICITY GENERATION	2683
SOLAR ENERGY IN DEVELOPING COUNTRIES	3105
SOLAR ENERGY INTERNATIONAL & NATIONAL PROGRAMMES	3151
AUTHOR INDEX	3351
SUBJECT INDEX	3359

Volume One

CONGRESS OPENING ADDRESS	xiii
FARRINGTON DANIELS ADDRESS	xvi
ACTIVE HEATING & COOLING IN BUILDINGS	
I. System Performance—DHW Systems, Control, Collectors and Materials	1
II. System Performance—Space Heating Systems; Systems with Heat Pumps	253
III. Cooling and Air Conditioning; Cooking; Drying; Conversion to Work; Swimming Pools and Solar Ponds	515
IV. Thermal Energy Storage; System Test Methods	687

Volume Two

AGRICULTURAL PROCESSES	945
BIOMASS	1205
IMPLEMENTING SOLAR ENERGY	1363
INDUSTRIAL PROCESSES	1589

Volume Three

PASSIVE HEATING & COOLING IN BUILDINGS	1753
PHOTOCHEMISTRY, THERMOCHEMISTRY & ADVANCED SYSTEMS	2175
SOLAR RESOURCES	2309
SIMULATING & MODELLING	2503

SOLAR ELECTRICITY GENERATION

RAPPORTEUR'S REPORT P.T. Landsberg	2687
KEYNOTE PAPERS	
RESULTS OF THE UNITED STATES RESIDENTIAL PHOTOVOLTAICS PROGRAM P.D. Maycock	2695
SOLAR THERMAL POWER GENERATION. THE EXAMPLE OF THE EUROPEAN 1 MW(e) POWER PLANT 'EURELIOS' J. Gretz	2705
CONTRIBUTED PAPERS	
CONCEPTUAL DESIGN OF A 500 kW/YEAR MASS-PRODUCTION PROCESS OF LOW-COST SILICON SOLAR CELL AND MODULE K. Yamagami, S. Noguchi, K. Kurokawa & T. Horigome	2715
THE ROLE OF RESEARCH AND TECHNOLOGY DEVELOPMENT IN PHOTOVOLTAICS R. Ferber, E. Costogue & R. Forney	2720
LARGE-SCALE PRODUCTION OF CdS/Cu ₂ S SOLAR CELL MODULES B. Schurich, J. Worner & N. Abdelmonem	2726
CURRENT STATUS OF THE INVESTIGATION OF THE EVAPORATED CdS/Cu ₂ S SOLAR CELLS J. Worner, B. Schurich & N. Abdelmonem	2732
LARGE AREA CADMIUM SULPHIDE THIN FILMS PRODUCED BY ELECTROPHOTETIC DEPOSITION T.J. Cumberbatch, I.D. McNally, E.W. Williams, D.J. Gibbons, H. Clow, P.M. Dickinson, R. Hill, N.M. Pearsall, J. Woods, G. Russell & F. Poulin	2738
PHOTOVOLTAIC POWER FOR THE PEOPLE P.R. Wolfe	2743
ON THE SIMULTANEOUS USE OF LAND FOR SOLAR ENERGY CONVERSION AND AGRICULTURE A. Goetzberger & A. Zastrow	2750
A SERIES OF R & D ON SOLAR PHOTOVOLTAIC CONVERSION SYSTEMS OF JAPAN K. Kurokawa, H. Akabane, H. Hosokawa & K. Murakami	2755
PROSPECTS FOR PHOTOVOLTAIC ELECTRICITY GENERATION IN WESTERN AUSTRALIA R.R. Booth & S.G. Saunders	2760
PHOTOVOLTAIC ENERGY SYSTEM FOR RURAL TROPICAL HABITAT C.E. Emetaram, E.D. Tari & R. Wijewardene	2767
PHOTOVOLTAIC POWERED VILLAGE LIGHTING SYSTEMS: A SYSTEM DESIGN EXPERIMENT J. Avery, E. Berman & B. Pyle	2772

PHOTOVOLTAIC EXPERIENCE IN THE TROPICS P.J. McKenzie & W.R. Bottenberg	2777
DESIGN OF LOW COST AND RELIABLE PHOTOVOLTAIC POWERED REFRIGERATOR FOR TROPICAL REGIONS CLIMATE W.R. Anis, R.P. Mertens & R.J. Van Overstraeten	2784
DESIGN OF MINIMUM COST STAND-ALONE FLAT PLATE PHOTOVOLTAIC SYSTEMS W.J. Kaszeta	2789
PHOTOVOLTAIC UTILITY-INTERACTIVE RESIDENCES FOR THE UNITED STATES A.R. Millner, R.W. Matlin, S.J. Strong & R.J. Osten	2798
A JOINT INDIAN HEALTH SERVICE - DEPARTMENT OF ENERGY PHOTOVOLTAIC PROJECT TO PROVIDE ELECTRICITY AND DOMESTIC WATER FOR AMERICAN INDIANS S. Reyes	2803
RESIDENTIAL PHOTOVOLTAIC APPLICATIONS IN THE UNITED STATES S.J. Strong & R.J. Osten	2815
SOPHIA ANTIPOLIS PHOTOVOLTAIC SOLAR HOUSE IN OPERATION D.J. Denis	2821
COMPUTERIZED FAULT ISOLATION OF LARGE SCALE MODULAR PHOTOVOLTAIC ARRAYS C.S. Costanzo, K.E. Hogeland & V.V. Risser	2827
A PRACTICAL 40 WATT SILICON FLAT PLATE PHOTOVOLTAIC MODULE WITH IMMUNITY TO REVERSE BIAS HEATING DAMAGE AND MINIMAL POWER LOSS CAUSED BY SHADOWING OR LOW TEMPERATURE OPERATION I.A. Lesk, N.G. Sakiotis & R.N. Kassner	2833
OVER 20% EFFECTIVE EFFICIENCY PHOTOVOLTAIC FLAT PANEL A. Luque, A. Cuevas, J. Eguren & J. del Alamo	2839
ELECTRICITY GENERATION - PHOTOVOLTAIC STORAGE BATTERIES AND CONTROL A. Dichler	2848
THERMOPHOTOVOLTAICS: AN ALTERNATIVE TO SIMPLE SOLAR CELLS R.C. Neville	2857
NON-TRACKING PHOTOVOLTAIC CONCENTRATORS E. Harting, D.R. Mills & J.E. Giutronich	2866
THE EFFECT OF INHOMOGENEOUS IRRADIATION ON EFFICIENCY AND ECONOMY OF SILICON SOLAR CELLS A. Wagner	2871
CHARACTERIZATION OF A DICHROIS MIRROR USED IN A PHOTOVOLTAIC SYSTEM WITH TWO TYPES OF SOLAR CELLS (BICOLOUR SOPHOCLE) D. Esteve, V.V. Pham, O. Soumaoro, F. Therez & G. Vialaret	2876
CONSTRUCTION AND FIELD TESTS ON A PARABOLIC TROUGH'S CONCENTRATOR'S PHOTOVOLTAIC MODULE M. Guiffrida & M. Liquindoli	2881
CONSTANT FLUX REFLECTORS WITH SPHERICAL ABSORBERS N. Tully	2882
PHOTOVOLTAIC INTERMEDIATE APPLICATION EXPERIMENTS K.L. Biringer	2886
HIGH TEMPERATURE TESTING OF SOLAR CELLS UNDER CONCENTRATED SUNLIGHT CONDITIONS E.K. Stefanakos, W.J. Collis & E.H. Martin	2891

2777	OPTICAL ANALYSIS OF NON-IDEAL PARABOLOIDAL CONCENTRATING REFLECTORS S.V. Shelton	2892	
TAIC POWERED REFRIGERATOR FOR rstraeten	2784	A TOTAL SOLAR PHOTOVOLTAIC ENERGY SYSTEM H.V. Smith	2903
PLATE PHOTOVOLTAIC SYSTEMS	2789	THEORETICAL AND EXPERIMENTAL ANALYSIS OF TWO PASSIVE COOLING SYSTEMS FOR THE SOLAR CELLS WITH CONCENTRATION A. Dumas & F. Draghetti	2908
ES FOR THE UNITED STATES & R.J. Osten	2798	STRUCTURAL DESIGN DEVELOPMENT OF A PHOTOVOLTAIC CONCENTRATING ARRAY USING FRESNEL LENSES S. Broadbent & J.A. Sanders	2913
NT OF ENERGY PHOTOVOLTAIC TIC WATER FOR AMERICAN INDIANS	2803	A MICROCOMPUTER BASED SUN TRACKING SYSTEM A. Traca-de-Almeida, H. Jesus & L. Moura	2920
THE UNITED STATES	2815	RECENT DEVELOPMENTS IN A SOLAR POWER GENERATOR R. Almanza, F. Munoz, A. Valdes, E. Barrera & E. Montes	2926
E IN OPERATION	2821	A SPANISH 'POWER TOWER' SOLAR SYSTE. THE PROJECT CESA-1 A. Munoz Torralbo	2931
ALE MODULAR PHOTOVOLTAIC sser	2827	IMPLICATIONS, REQUIREMENTS AND RESTRICTIONS FOR SOLAR TOWER SYSTEMS WITH HIGH TEMPERATURE BRAYTON CYCLE K.J. Erhardt	2936
HOTOVOLTAIC MODULE WITH AND MINIMAL POWER LOSS CAUSED N r	2833	100 kW SOLAR POWER PLANT IN CORSICA 'SOFRETES/BERTIN' E. Bacconnet, M. Dancette & J. Malherbe	2941
IC FLAT PANEL l Alamo	2839	LARGE PARABOLIC DISH COLLECTORS WITH SMALL STEAM- OR GAS-TURBINE OR STIRLING POWER CONVERSION SYSTEMS K. Bammert & M. Simon	2950
DRAGE BATTERIES AND CONTROL	2848	ADVANTAGES OF LARGE PARABOLIC DISH SYSTEMS FOR POWER GENERATION A.G. Sutsch	2951
SIMPLE SOLAR CELLS	2857	EFFECTS OF A CIRCUMFERENTIALLY NON UNIFORM INCIDENT SOLAR FLUX ON THE WALL TEMPERATURE DISTRIBUTION IN A TUBE FLOW M. Debiane & D. Blay	2960
h	2866	ADVANCED SOLAR RECEIVERS - HIGH TEMPERATURE STEAM LOOP EXPERIMENTS A.H. Campbell & H.L. Teague	2967
ON EFFICIENCY AND ECONOMY OF	2871	COMPARISON OF ANALYSIS AND TEST RESULTS FOR A CENTRAL RECEIVER P.S. Bremner & R.K. McMordie	2972
ED IN A PHOTOVOLTAIC SYSTEM (OPHOCLE)	2876	APPLICATION OF SOLAR CELLS IN SOLAR TOWER PLANTS F.K. Boese, H. Kiupel, U. Leuchs & H.G. Spillekothen	2977
herez & G. Vialaret	2881	STUDY OF LOW POWER ENGINES: THERMODYNAMIC CONVERSION OF SOLAR ENERGY C. d'Amelio, M. Blasi & R. Tuccillo	2983
IC TROUGH'S CONCENTRATOR'S	2881	SOLAR SYSTEMS WITH LIQUID METAL MAGNETOHYDRODYNAMIC ELECTRIC POWER GENERATORS H. Branover, A. El-Boher & A. Yakhot	2993
BSORBERS	2882	TWO YEARS' OPERATION OF THE 30/50 kW _e SOLAR FARM AT GETAFE/SPAIN R. Koehne, M. Kraft & A. Perez Vidal	2999
ERIMENTS	2886	SOLAR ACTIVATED POWER GENERATION UTILISING A MULTI-VANE EXPANDER AS A PRIME MOVER IN AN ORGANIC RANKINE CYCLE M. Hussein, P.W. O'Callaghan & S.D. Probert	3006
NDER CONCENTRATED SUNLIGHT tin	2891		

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 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, ϵ 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 σ_0 is the Stefan-Boltzmann constant, ϵ 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:

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

and the cell temperature is obtained by standard algorithm.

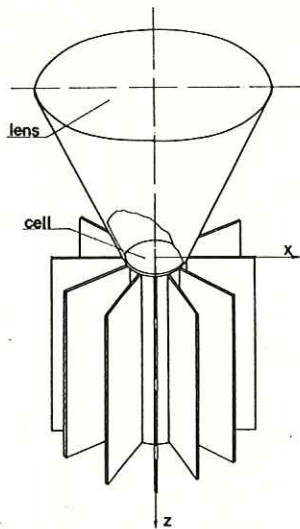


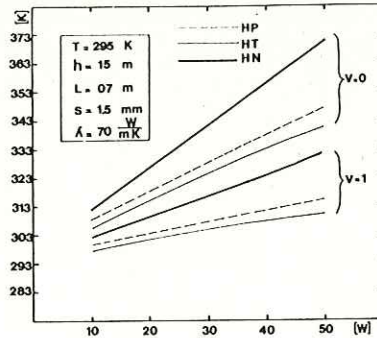
Fig. 1. Design of the device

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 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 were 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 shows² 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.

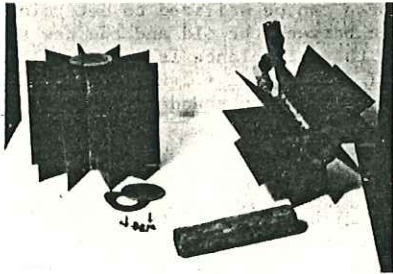


Fig. 6. Experimental devices

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

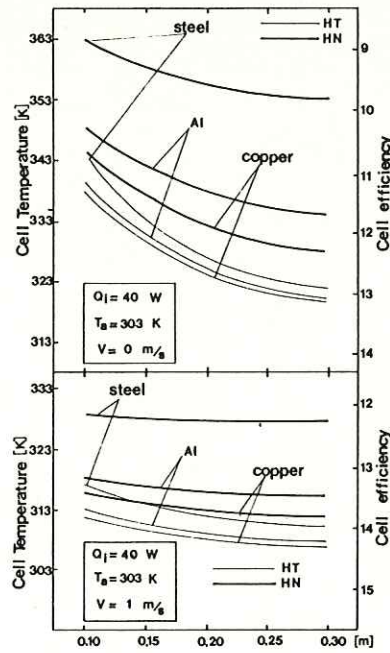


Fig. 2. T_C and ϵ vs. the fins length

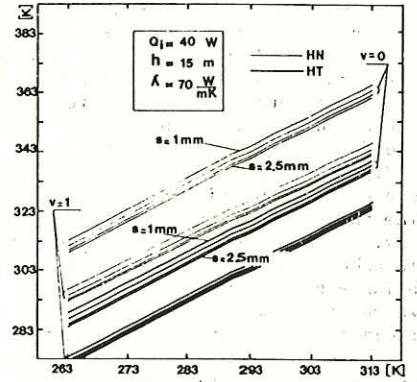


Fig. 3. T_C vs. T_A at different values of thickness of the fins

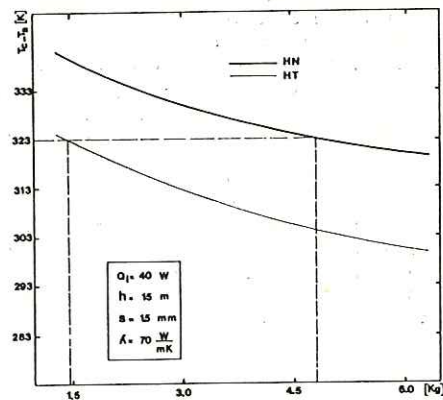


Fig. 4. Influence of the fins number in terms of weight on the difference $(T_C - T_A)^1$

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

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.

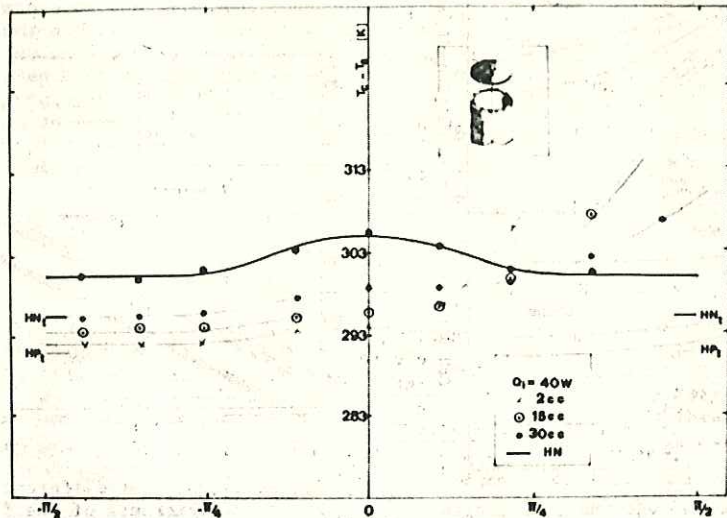


Fig. 7. The difference ($T_C - T_A$) vs. inclination angle

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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