DESIGN AND NUMERICAL ANALYSIS OF ENHANCED COOLING TECHNIQUES FOR A HIGH CONCENTRATION PHOTOVOLTAIC (HCPV) SYSTEM

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ABSTRACT: A thermal model has been developed to predict the heat output of a PV cell, in order to examine the most efficient and cost effective cooling system for a 500x concentrating PV cell. The worst case scenario (i.e. zero forced convection from a flat surface) was selected in order to identify and measure the scale of the challenge. Different geometries and materials of heat sinks were then designed and tested for passively cooling purposes of the High Concentration Photovoltaic system. It is shown that passive cooling of a CPV system with CR of 500x is insufficient to maintain the cells below 80°C, especially for high ambient temperatures. Numerical analysis and simulations in MatLAB and COMSOL Multiphysics report on the level of cell's surface temperature and convective heat transfer coefficient that is required for the PV cell to operate below safe temperature limits.
Keywords: Multijunction Solar Cell, Concentrating Photovoltaics, Concentrator Cells, Thermal Modeling, Thermal Performance

1 INTRODUCTION

During the last few decades, an impressive growth in Renewable Energy Sources (RES) Sector has been observed. It is commonly accepted that RES such as solar, wind and biomass energy can provide an alternative to the use of fossil fuels which are significantly contributing to environmental pollution and the threat of global warming.

The aim of this research is to design a cooling device of a 500x Concentrating Photovoltaic System (CPV). This research is part of a larger project, BioCPV, which will develop and integrate the use of CPVs with Biomass and Hydrogen generation to help to reduce the urban and rural energy divide in India.

Concentrating Photovoltaics use mirrors and lenses to focus solar irradiance on a receiver. This allows a reduction in the required PV cells to produce of the same electrical power. CPV's provide an opportunity to use highly efficient (>30%) multijunction cells (MJ cells) which are more expensive than conventional PV cells. The use of MJ cells can only be economically viable if high concentration solar flux, above 300 suns, is produced.

However, while the concept of this technology is straightforward, the practice has proved deceptively difficult. The main technical barrier is the high surface temperature on the cell which leads to reliability issues as well as a reduction in the conversion efficiency of the cell [1]. The PV cell's efficiency drops with increasing temperature cause by the high heat flux from the solar source. Also, if the cell operates with a temperature over a specified by a manufacturer limit (80^{o}C for AZURSPACE GMBH [2]) the cells will exhibit long-term degradation [3]. Therefore, under high concentration there is also a considerably higher heat load that needs to be dissipated.

The first challenge in generating electricity efficiently is effectively cooling the cells to allow peak performance in all conditions. A second challenge, although not part of the current investigation, will be to take advantage of the "waste heat" to improve the overall system efficiency. If this can be achieved it will lead to more economical, efficient and reliable concentrating photovoltaic systems.

Many investigations have been done on the cooling of CPVs; according to Royne et al. [3], who presented an extensive overview on different cooling techniques, passive cooling can be sufficient for single cell geometries for solar flux up to 1000 suns as there is a large area available below the cell for a heat sink. Moreover, for densely packed cells and for concentrations higher than 150 suns, active cooling is necessary. Also, Royne et al. [3] have concluded that the thermal resistance of the cooling system must be less than 10^{-4}Km^2/W for concentration levels above 150 suns.

Araki et al. [4] used an array of Fresnel lens' in a single cell geometry to focus the sunlight on the cells under a concentration of 500x. The cells were placed on a printed epoxy and copper sheet and then onto an aluminum plate to spread and dissipate the heat. The outdoor experiments showed a cell temperature increase with a ΔT of only 18^{o}C between ambient and cell temperature. A good thermal contact between the cell and the heat spreader is therefore, critical to keep the cell's temperature as low as possible. Natarajan et al. [5] modelled a LCPV system (10 suns) with and without the use of passive cooling in order to predict the solar cell temperature distribution under peak solar radiation. It was found that, without any cooling fins, the maximum temperature was 68.2^{o}C while using the fins the maximum temperature was 55.1^{o}C (Tambient=20^{o}C). Natarajan et al. [5] also investigated the effect of the ambient temperature on a passively cooled solar cell (2 fins) and found that for an ambient temperature of 50^{o}C the solar cell temperature was 84.9^{o}C. Cell temperature correlations were proposed for ambient temperature between 20^{o}C and 50^{o}C and solar radiation between 100W/m^2 and 1000W/m^2. It was also agreed that the thickness and thermal conductivity of the back plate of the receiver is important for reducing the temperature of the solar cell.

Chou et al. [6] modelled the thermal management of a HCPV using Finite Element Analysis. Using an aluminum plate as a heat sink, Chou et al. [6] calculated a maximum cell (III-V) temperature of 69.02^{o}C and a maximum aluminum plate temperature of 66.89^{o}C for a thermal resistance of 4.67^{o}C/W, ambient temperature of 34^{o}C, wind velocity of 4.2m/s under concentration ratio of 380x. The numerical model was found to be in very good agreement with the experimental results to an accuracy of 98% (62-68^{o}C for wind speed between 4.2-6.6m/s). Min et al. [7] designed and analysed a thermal model for a single concentrator solar cell. A black coated aluminum plate was used as a heat sink which was 700 times larger than the III-V cells size in order to maintain the temperature below 40°C and achieve a high conversion efficiency; the concentration ratio was 400x. The results were validated experimentally with an accuracy of 1 K.

However, work still has to be undertaken to determine the optimum solution for each case of HCPV system. More analytical and detailed models are required to achieve this target. The purpose of this investigation is to develop a model which will predict the thermal behavior of the CPV configuration. A large aluminum heat sink is also tested, in order to investigate the possibility of a 500x HCPV system to be cooled passively.

2 METHODOLOGY/APPROACH

In this work, a single cell is evaluated using Finite Element Analysis (FEA) in COMSOL Multiphysics [8], a commercially available software. COMSOL Multiphysics offers the possibility to combine different engineering or physical problems in the same model while being able to choose different solvers in order to achieve better accuracy. Another advantage is that it gives the option to import data from MatLAB.

2.1 Problem Identification

As discussed previously, the aim of this research is to design a cooling device of a 500x concentrating photovoltaic system in order to cool down the cell below a certain limit and avoid performance degradation and secondly, to transfer the 'waste' heat to the biomass and hydrogen unit with the most cost effective, reliable and efficiency way.

The first step of this investigation was to predict the minimum heat transfer coefficient that is needed to keep the temperature of the cells below 80°C. Hourly meteorological data from Athens which has a similar climate, in terms of direct solar radiation and ambient temperature, to that of the proposed location of the eventual power plant has been used in this study for a period of one year. Assuming that the cell's electrical efficiency is constant at 36% it was found that a heat transfer coefficient higher than 5.9kW/m^{2}K is needed from the cooling system.

The electrical power flux for each hour of the year (j) is calculated from the equation:

$$q_{elec}(j) = CR\,.\; n_{optical}\cdot q_{solar}(j)\,.\,n_{elec} \qquad (1)$$

where CR is the geometrical concentration ratio of the system (500x), $q_{solar}(j)$ is the hourly direct radiation from the sun W/m^2,

$n_{optical}$ the optical efficiency of the system (~85%) and n_{elec} equals to 36% as mentioned before.

The heat flux absorbed by the receiver equals to:

$$q''_{heat}(j) = (CR\,.\,n_{optical}\cdot q_{solar}(j)) - q_{elec}(j) \qquad (2)$$

where $q''_{heat}(j)$ is the amount of heat flux that has to be anticipated by the candidate cooling system.

The heat transfer coefficient needed for each hour of the year is calculated from equation below and it is shown graphically in figure 1:

$$h(j) = \frac{q_{heat}\,(j)}{T_{Surface} - T_{Ambient}(j)} \qquad (3)$$

where $T_{surface}$=80°C. It is apparent, that very high rates of heat flux occur, especially during the summer period. Consequently, a high heat transfer coefficient from the cooling system is required, in order to keep the system working without the risk of degradation.

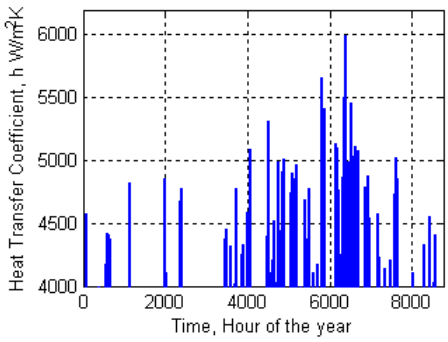

Figure 1: Heat transfer coefficient needed for a maximum cell surface temperature of 80°C.

2.2 Theory and governing equations

The conjugate heat transfer interface was used to model the thermal problem. This interface has the advantage to combine both heat transfer in solids and fluids including laminar or turbulent flow at the same time.

In the case of a passively or actively cooled receiver, the heat is transferred by conduction between the solid layers of the receiver and the steady state equation is given by 4:

$$\nabla.(\mathrm{k}\nabla\mathrm{T}) = 0$$
$$q = -k.A.\frac{\Delta T}{\Delta x} \qquad (4)$$

The solar energy that is transformed to heat must be dissipated from the bottom substrate or cooling system to the environment or to another unit. The heat which is dissipated either by natural or forced convection is described by 5:

$$q = h.A.\Delta T \qquad (5)$$

The heat, which is lost in the environment, due to natural convection happens on every surface that faces the ambient. COMSOL Multiphysics already contains the correlations for each type of surface either that is vertical, horizontal or inclined. Those can be found from Incropera and DeWitt [9].

The heat that is lost to the environment due to radiation is given by 6:

$$q_{rad} = \varepsilon.\sigma.(T_{cell}^4 - T_{amb}^4) \qquad (6)$$

COMSOL software also solves numerically the governing equations as the Navier-Stokes which can describe the momentum, continuity and energy equation. In the case of incompressible Newtonian fluid flow, were the density is almost constant then the Navier-Stokes equation (7) can be simplified into a continuity and momentum equations, (8) and (9) respectively;

$$\rho\frac{\partial u}{\partial t} + \rho(u.\nabla)u = \nabla.\left[-\rho I + \mu(\nabla u + \nabla u)^T) - \frac{2}{3}\mu(\nabla.u)I\right] + F \qquad (7)$$

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho u) = 0 \qquad (8)$$

$$\rho u.\nabla u = -\nabla p + \nabla(\mu(\nabla u + (\nabla u)^T)) \qquad (9)$$

In the case of an active cooling system modelling, sometimes it is desirable to run the model in high flow rates having high Reynolds number, thus turbulent flow, in order to dissipate the heat more effectively. COMSOL gives the possibility to run the RANS (Reynolds Averaged Navier-Stokes) turbulent model type for that case and select between the k-ε, k-ω, Spalart-Allmaras models. However those models are not discussed in the present report but will be applied and investigated in the near future.

The most important parameter when designing a cooling system is the pressure drop. Turbulent flow, gives high pressure drop across a pipe which means high parasitic power consumption from the pump. This is very critical in the design, as it is not desirable to waste high amounts of power from what is produced by the CPV plant. This is also very crucial for the cost to run the application and should be the main criterion when modeling different cooling techniques.

2.3 Assumptions and boundary conditions

A few assumptions have to be done in order to be able to model the 2D physical problem;

1) The direct solar radiation is considered to be $1000 W/m^2$ and uniform across the III-V cell.
2) The geometrical concentration ratio, CR=500x.
3) Optical efficiency was considered to be 85% and the electrical conversion efficiency of the cells 36%.
4) Therefore the heat flux on the cell is equal to $272 kW/m^2$.
5) The ambient temperature is assumed to be 20^oC.
6) Perfect thermal insulation was assumed on the sides of the cell and a no slip boundary condition was considered on the wall.
7) Open boundary at the right and left between the cell and Fresnel lens with no stress.

2.4 2D time dependent model without heat sink

A numerical model was then developed to calculate the surface temperature of the III-V cell for single cell geometry. Firstly, the receiver was modeled without the use of heat sink, in order to calculate the highest/worst temperature on the cell and compare with literature for validation purposes.

Figure 2 shows the schematic diagram of the setup that is simulated while on table I and II, the dimensions and thermophysical properties of each layer are presented. For simplicity, the bypass-diodes, the electrical connections and packing materials are not modeled. The gap between the cell and the Fresnel lens (top layer) which is covered by air is around 26mm.

The model applies an inflow heat flux on the cell, while the bottom and top surface release heat to the environment through natural convection and radiation (surface to ambient). The left and right boundaries of the fluid are set as 'open boundaries'. The volume force of the fluid is F_y=-g(ρ_∞-ρ) [N/m^3].

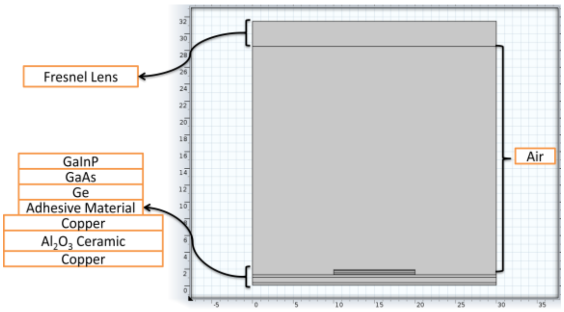

Figure 2: CPV system configuration of the simulation test setup.

Table I: 2D Model dimensions.

Layer	Length [mm]	Thickness [mm]
Fresnel Lens (PMMA)	30	3
GaInP	10	0.1
GaAs	10	0.2
Ge	10	0.2
Adhesive Material	10	0.1
Copper	30	0.3
Al_2O_3 Ceramic	30	0.63

Table II: Materials' thermophysical properties.

Layer	k [W/mK]	Cp [J/kgK]	ρ [kg/m^3]
Fresnel Lens (PMMA)	0.1875	1465	1162
GaInP	73	73	5300
GaAs	65	550	5316
Ge	60	310	5323
Adhesive Material	10	800	4000
Copper	401	385	8933
Al_2O_3 Ceramic	20	880	3700
Aluminum	238	903	2702

2.5 2D time dependent model with heat sink

On the same way as described above, a model was developed using a large aluminum heat sink. The heat sink's dimensions can be seen on table III while the schematic diagram is shown in figure 3. The receiver is attached on the heat sink by an extra layer of adhesive material while the rest layers are in the same order as previously.

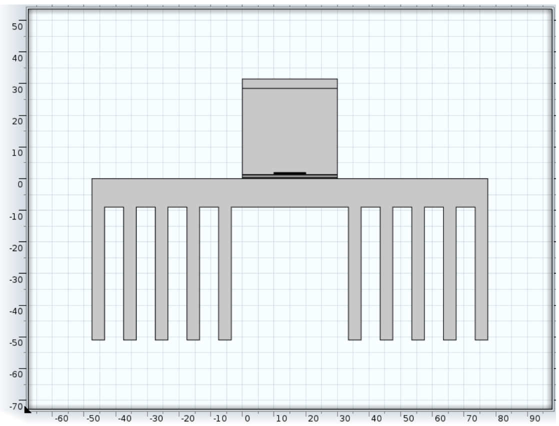

Figure 3: CPV system with heat sink configuration.

Table III: Aluminum heat sink's dimensions.

Length [cm]	Thickness [cm]	Number of Fins	Fins Length [cm]	Fins Thickness [cm]
12.5	0.9	10	4.2	0.4

3 MESHING AND SOLVERS

The simulation ran using the solver algorithm PARDISO (sPARse DIrect SOlver) which is a direct solver. COMSOL Multiphysics has the capability to choose the solver by default, but other solvers such as SPOOLES (SParse Object Oriented Linear Equation Solver) and MUMPS (MUltifrontal Massively Parallel sparse direct Solver) were tested as well and seemed to take much longer time to simulate than the PARDISO. All the solvers above are based on the Gauss reduction method [10]. More details on the PARDISO solver can be found on [11].

The receiver configuration including the heat sink was meshed using the physics controlled mesh sequence as part of COMSOL. A fine mesh setting was chosen for this study with 6,333 elements over an area of 942mm^2 (no cooling) and 9,733 elements on 3750mm^2 (including the heat sink).

4 RESULTS AND DISCUSSION

The simulation for the worst case scenario was conducted for a period of 600 seconds. The steady state for this scenario was achieved after around 370 seconds and the temperature distribution can be seen in figure 4. The cell's temperature is 1374^{o}C while the difference between the cell and the bottom substrate is ΔT=9^{o}C. The results agree with the literature [4, 7, 12, 13], as it has been reported that for a CR of 500x the temperature on the cell can be up to 1400^{o}C.

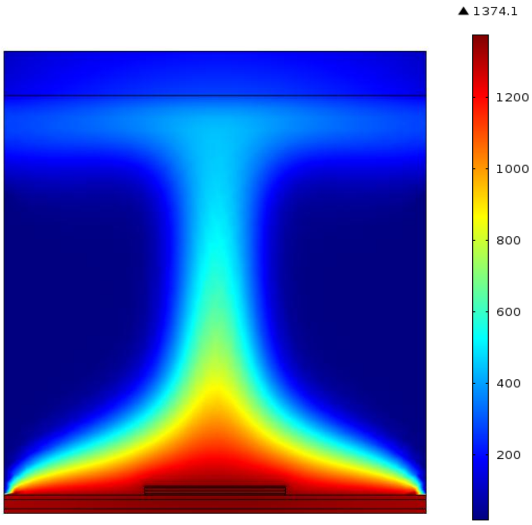

Figure 4: Temperature distribution [^{o}C] across the receiver and lens for t=600s.

In figure 5 the velocity magnitude profile is shown. The no slip boundary condition took over on the Fresnel lens' and substrate's walls and the parabolic flow profile is created. A maximum velocity of 0.5776m/s mainly in the center of the fluid is observed.

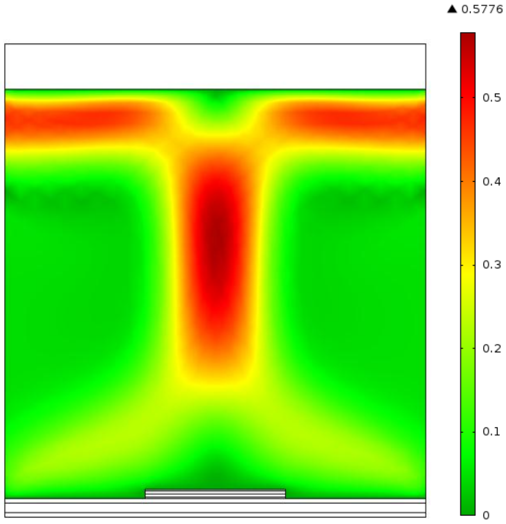

Figure 5: Velocity magnitude [m/s] in air gap for t=600s.

Figures 6 to 8 illustrate the results of the temperature and velocity magnitude profile of the attached on an aluminum heat sink receiver. In this case, the steady state was achieved after approximately 6400 seconds. A cell's surface temperature of 176.36^{o}C was observed. The temperature difference between the cell's surface and the heat sink is ΔT 15^{o}C (on the center of the heat sink base) to 37^{o}C (middle fins) while the left and right edges show a ΔT of 56^{o}C. Also, a maximum temperature difference of 4.5^{o}C from the center of the cell to the edges is observed.

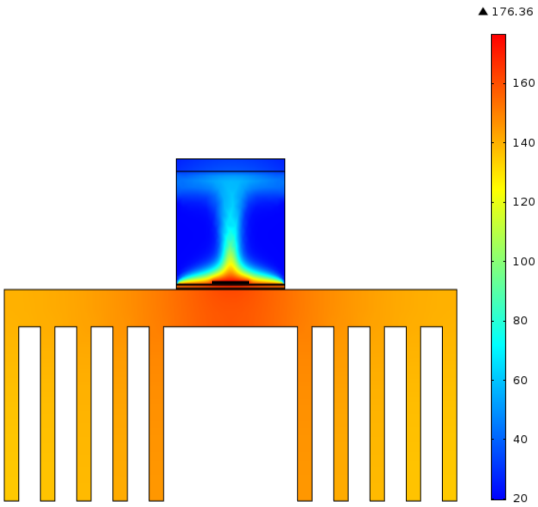

Figure 6: Temperature distribution [^{o}C].

The option of using a copper heat sink instead of aluminum was simulated as well with a maximum temperature of 167.27^{o}C. Both solutions proved that passive cooling is not enough to dissipate effectively the high heat fluxes on the cell even using a very large size of heat sink. In particular for the present project's considerations, if the Indian high ambient temperatures during summer are considered, then the option of passive cooling should be rejected.

In figure 9 the behavior of the cell with a convective heat transfer coefficient of 5.9kW/m^{2}K on the bottom substrate layer was tested for validation purposes of the section 2.1. Good agreement is observed while the maximum cell temperature is 55.45^{o}C and always lower than 80°C, even when the ambient temperature reaches the 40°C.

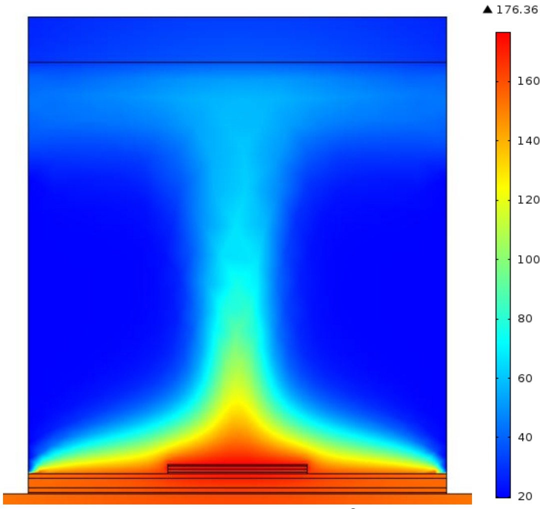

Figure 7: Temperature distribution [°C] on the receiver (zoomed image from figure 6).

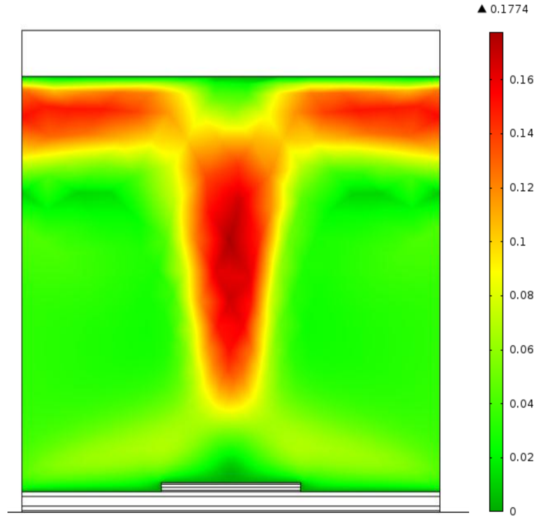

Figure 8: Velocity magnitude [m/s] in air gap.

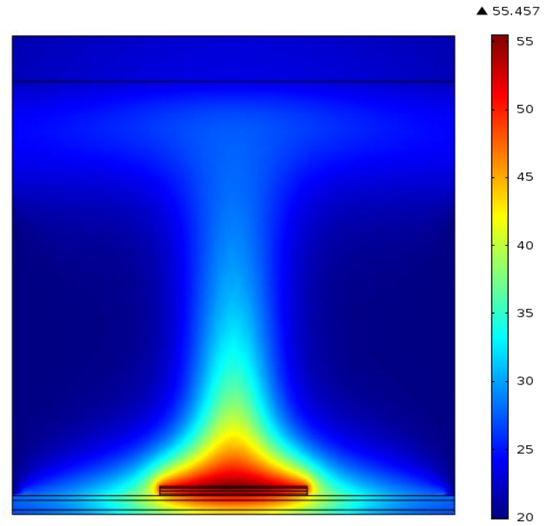

Figure 9: Temperature distribution across the receiver after 600 seconds for h=5.9kW/m^{2}K.

5 CONCLUSIONS AND FUTURE WORK

A detailed numerical analysis using FE methods was used to predict the thermal behavior of the receiver under different scenarios. While the results are in good agreement with literature, it is very important that they are validated experimentally. In order to evaluate many different designs an indoor test facility will be needed.

Passive cooling cannot dissipate enough heat from the cell even when a very large heat sink (compared to the cell's size) is used. Also, considering that the current project's design will be installed in India, it can be confidently said that passive cooling would be catastrophic for the installation.

According to Norton et al. [14], concentrating photovoltaic systems that use multijunction cells can suffer from suboptimal performance due to changes in the solar spectrum. Future work should include the solar spectrum dependence of MJ cells. The electrical and thermal performances of MJ cells are highly dependent on the solar spectrum changes as the electrical efficiency of 36% (which was assumed as constant in the present work) is never reached in the field.

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