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Modeling a passive cooling system for photovoltaic cells under concentration

Allison K Gray
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MODELING A PASSIVE COOLING SYSTEM
FOR PHOTOVOLTAIC CELLS UNDER
CONCENTRATION

By

Allison K. Gray

Bachelor of Science in Engineering
University of Nevada, Las Vegas
2005

A thesis submitted in partial fulfillment
of the requirements for the

**Master of Science Degree in Mechanical Engineering
Howard R. Hughes College of Engineering
Department of Mechanical Engineering**

**Graduate College
University of Nevada, Las Vegas
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is approved in partial fulfillment of the requirements for the degree of

Master of Science in Mechanical Engineering

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ABSTRACT

Modeling a Passive Cooling System for Photovoltaic Cells under Concentration

By

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Dr. Robert F. Boehm, Examination Committee Chair
Professor of Mechanical Engineering
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An analysis of the Amonix high concentration photovoltaic (HCPV) passive cooling system was performed in order to obtain a better understanding of the present design and identify poor areas of dissipation. Cooling of photovoltaic cells under high intensity solar irradiance is a major concern when designing concentrating photovoltaic systems. Solar cell temperatures increase if the waste heat is not removed causing the cell voltage to decrease lowering the power generated. The concentrator geometry was studied so a model of the HCPV passive cooling system could be generated to analyze the system numerically. In addition to this experiments were conducted in the field and then used to compare with the numerical results. There was less than a 5% difference between both studies indicating that the numerical results were correct. This study provides beneficial information regarding areas of poor circulation with detailed descriptions about the temperature distributions and air velocity.

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LIST OF SYMBOLS

g	Gravity
Gr	Grashof number
h	Heat transfer coefficient
I_{pp}	Peak power current
I_{sc}	Short circuit current
k	Thermal conductivity
L	Characteristic length
P	Cell power output
Pr	Prandtl number
q	Heat transfer via radiation, convection and conduction
q_{conv}	Convective heat transfer
q_{rad}	Radiation heat transfer
Ra	Rayleigh number
T_s	Surface temperature
T_∞	Ambient temperature
T_{cell}	Cell surface temperature
V_{oc}	Open circuit voltage
V_{pp}	Peak power voltage
α	Thermal diffusivity

β	Thermal expansion coefficient
ε	Surface emissivity
η	Cell efficiency
σ	Stefan-Blotzmann constant
ν	Kinematic viscosity

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CHAPTER 1

INTRODUCTION

Cooling of photovoltaic cells is one of the main concerns when designing concentrating photovoltaic systems [1]. Cells may experience both efficiency loss and degradation due to high temperatures. Design considerations for the cooling systems include low and uniform cell temperatures, system reliability, sufficient capacity for dealing with ‘worst case scenarios’, minimal power consumption by the system, and low costs. Concentration of sunlight onto photovoltaic cells results in the replacement of expensive photovoltaic material with less expensive concentrating mirrors or lenses. This is one method of lowering the cost of solar-to-electrical systems. Because of the reduction in solar absorber material, higher efficiency PV cells can be used. Even with the higher efficiency cells, only a fraction of the incoming sunlight striking the cell is converted into electrical energy. As shown in Table 1, the typical efficiency value for concentrator cells is approximately 21% [2]. The remainder of the absorbed energy is converted into thermal energy in the cell and can cause the junction temperature to rise unless the heat is efficiently dissipated to the environment. The rise in junction temperature will also lower the system efficiency.

Table 1 describes some of the physical characteristics of some concentrator cells. This includes the cell thickness in microns with the fill factor parameters under concentration. The incident power in suns or concentration ratio is given for the

efficiency described in this table. As part of determining cell performance, many researchers use the fill factor as well which is the $(V_{pp} * I_{pp}) / (V_{oc} * I_{sc})$. This gives the performance ratio of the cell.

Table 1. Concentrator solar cell characteristic described by Sinton [2].

Cell Characteristics (cell area = 0.15 cm ²)				
Cell Thickness (microns)	130	233	130	233
Incident Power (suns $\pm 1\%$) (1 sun=0.1Watt.cm ²)	150	149	350	325
V_{oc} ($\pm 5mV$)	810	810	825	820
I_{sc} ($\pm 0.5\%$ A/cm ²)	5.17	4.98	12.0	9.97
V_{pp} ($\pm 8mV$)	704	701	701	690
I_{pp} ($\pm 1\%$ Acm ²)	4.83	4.67	11.0	9.15
Fill Factor ($\pm 1\%$)	0.81	0.81	0.78	0.77
Cell mount temperature ($\pm 10C$)	24	22	23	23
Efficiency (± 0.5) (Calibrated Calorimetrically)	22.40%	21.70%	21.60%	19.10%

Purpose of Study

There are five major design considerations for cooling of photovoltaic cells under concentrated sun irradiance: cell temperature, temperature uniformity, reliability, usability of thermal energy, pumping power [1] and cost. Most of these design considerations are considered in this study of the passive cooling system. It is important that the passive cooling system remove waste heat as effectively as possible to insure lower cell temperatures and the removal of waste heat from the back of the cell area. The design should be reliable and simple so that it can easily be duplicated both in the factory and field while keeping the pumping power low to keep parasitic losses minimal. The current passive cooling system on the Amonix high concentrating photovoltaic (HCPV)

design is being studied so that a better understanding can be developed for the design and search for potential improvement areas.

Research Questions

The air flow around the heat sinks is under investigation for this study. Multiple passive cooling parameters of the current design that need to be investigated are listed below:

What is the worst case scenario for operation in the field?

- Temperature
- Wind Speed

What is the air flow around the heat sink at different tracking angle positions?

- Azimuth
- Elevation
- What are the chamber temperatures?
- What is the air flow inside the chamber?
- What is the rate at which heat is being removed from the cell area?

In the current HCPV design there have not been any system failures during operation under the worst case conditions; however, cell performance is the lowest. The worst case operating conditions are defined as the worst weather conditions that the system will encounter in the field. The worst case ambient air temperature condition in the field is approximately 40°C or 313K. The passive cooling system is based on natural convection making a 0 m/s wind speed the worst case wind condition because the heat sink will be dissipating the least amount of heat.

The HCPV system must track the sun through the day. This causes the heat sinks to be in different azimuth and elevation angles throughout the day depending on the sun's position. These angles are very important parameters that affect the passive cooling system's performance, and need to be studied over their normal operating range. The wind angle-of-attack on the passive cooling system can have a large effect in the heat sink heat dissipation rate. However under the wind speed worst case condition there is no angle of attack on the heat sink making the azimuth angle negligible. This will simplify the study so that only different elevation angles need to be modeled and simulated.

Figure 1. Sketch of the chamber geometry with heat sinks mounted on the bottom and air surrounding the system.

Having a better understanding of the air flow in and around the chambers, an enclosed volume with a lens on top and a single receiver plate on the bottom, is important for gaining insight about areas of poor air flow. The temperature and velocity

distributions were examined inside and outside of the chamber. The heat sink temperature distributions were also studied to determine the rate at which heat is being removed.

Significance of Study

This project has the potential of allowing Amonix to increase the performance of their module by improving the passive cooling system, which would decrease the cell temperatures in the field, improve temperature uniformity, cell efficiency, and increase cell reliability. PV cell performance is dependant on cell temperature as shown in Figure 2, which illustrates that cell decrease in efficiency when temperatures are increased up to 2% for a 20°C increase in some cases.

Figure 2. Comparison of different cell efficiencies at different cell temperatures. [1]

Having a uniform temperature distribution is important to the performance of a PV cell, because the highest temperature on the cell can decrease the open circuit voltage of that area and restrict the open circuit voltage of the entire cell. This will cause the open circuit voltage of an entire string, a set of PV cells connected in series to decrease, causing a lower peak power point, directly reducing the energy that is generated [9]. Solar cells have shown to degrade under high temperatures. This degradation is more evident with concentrator cells because of the high intensity onto a small area [1].

Because of this phenomenon, there is an interest in an improving in the passive cooling design. Operating at lower temperatures can increase Amonix's solar cell power and energy performance in the field, leading to an increase in overall system performance by lowering the cost per watt.

Amonix HCPV Module Description

The Amonix system relies on refractive optics to concentrate the sun's irradiance onto a solar cell. A square Fresnel lens made of circular facets is used to focus the sunlight onto a cell. The angle of each facet varies as a function of that facet's distance from the center of the lens so that all of the rays will converge to a focal point [Figure 3]. The Amonix concentrating back junction silicon cell is located at this focus point and converts the sunlight to electrical power. The Fraunhofer Institute rated the Amonix cell at a peak efficiency of 27.60% at a concentration ratio of 122 suns [10]. The cells were tested up to 405 suns with efficiencies equal to or greater than 25.00%, as shown in Figure 4.

Figure 3. Single cell concentrator.

Figure 4. Amonix cell efficiency up to 405 suns [10].

Figure 5. MegaModule® schematic

A chamber consists of twenty-four Fresnel lenses manufactured as a single piece or parquet, and cells mounted on a receiver plate at locations corresponding to the center of each lens. The cells are aligned in an array of 4 x 6 and assembled together in the shape of a box, called a chamber. Forty-eight chambers are assembled in a 4 x 12 (11 feet x 45 feet) rectangular shape with a steel C-channel structure used between lenses and receiver plates as the side structures that hold them at the right position. This structure along with the lenses and plates shown in Figure 5, is referred to as a MegaModule®. Each MegaModule® is designed to produce 5 kW of AC power at a direct normal irradiance

(DNI) of 850 W/m^2 and 20°C cell temperature (IEEE standard). Some physical characteristics and operating parameters for the system are given in Table 2. The system is rated at 25 kW ac with a total collector size of 2420 ft^2 (56 m^2). The collector size is considered to be the entire array area not the total cell area. The geometric concentration ratio is 250:1 and is defined as the relationship between the Fresnel lens and cell assuming that all of the energy entering the lens is focused onto the cell. The efficiency is defined as

Table 2. Physical characteristics of the MegaModule[®].

Rated Power Output @ 850 W/m^2 , 25°C	25 kW ac
MegaModules [®] size (ft)	44 x 11 x 2.5
Collector Size (ft)	55 x 44 x 2.5
Aperture Lens Area (ft^2/m^2)	1960/182
Total Face Area (ft^2/m^2)	2508/233
Number Of Cells	5,760
Concentration Ratio (suns)	250:1
Operating Voltage (ac)	277
Max. Wind Speed (mph)	90

CHAPTER 2

LITERATURE REVIEW

Concentrating PV systems are considered a promising approach to reducing solar energy conversion costs where Fresnel lenses or mirrors are used in exchange for costly PV material. Generally concentrating technologies have two major approaches, point focus and linear. Point focus concentrators can focus the sunlight onto a single area or multiple small areas. Both designs require two-axis tracking to keep the light focused onto the point. Linear concentrators focus the sunlight on to a line or linear shape. The same methods used in the point focus are used to concentrate the light. Two of the main differences are concentration shape and the tracking system. Linear concentrators usually need to track in only one axis.

All of the concentrator designs require a heat rejection system. This can be done two ways, with an active system or passive one. Active cooling systems are closed and use a working fluid to remove the waste heat. Typically a pump is used to circulate the cooling fluid and remove the waste heat. Passive cooling designs dissipate the waste heat by natural convection. This cooling method usually uses heat sinks with out the use of a pump or fan to increase the dissipation rate. The passive cooling concentrator systems will be discussed here.

Point Focus Concentrator Designs

Andreev [8] tested a point focus thermal photovoltaic (TPV) design that used a passive cooling system with glass lenses to magnify the sun's light, as shown in Figure 6. The TPV schematic in Figure 6 illustrates the general components of a concentrator including the passive cooling system location relative to the cell.

Figure 6. TPV schematic design [8].

Edenburn [7] discussed point focus passive cooling design as shown in Figure 7. This design was published by Edenburn in the early 80's where he had three major design parameters: (1) base heat exchanger, (2) heat exchanger thickness, and (3) ratio of total

heat exchange area to base area or heat exchange ratio. The cell temperature was calculated by determining the temperature difference across the bond, and solving the thermal conduction equations to find the temperature profile in the heat exchanger's base. This profile was dependant upon the heat transfer rate or convection coefficient, which is affected by the ambient temperature and wind speed. The annual power output for this design was then estimated using the typical meteorological year (TMY) data for Albuquerque.

Figure 7. Point focus concentrator design with heat sinks on the top and bottom of the receiver plate.[7]

Edenburn estimated the array energy generation and array costs for different variations of the point focus design shown in Figure 7 where the cost was divided by the energy and the cost was selected based on minimum energy cost for each lens size. To

find the optimized heat exchanger design (Table 3) Edenburn studied different lens areas at different concentration ratios with different cells and base areas. The data in Table 3 was used with Table 4 to determine the optimum heat exchanger cost compared to the performance and annual energy generated. His study concluded that passive cooling was not effective for use with large aperture area concentrators because high heat flux levels required thick and expensive heat exchangers to maintain high performance. His research also implied that a heat exchanger will change little for more expensive arrays but the difference can be significant for the less expensive ones. Edenburn's heat exchanger design [12][10] was optimized for minimum energy cost, which has been a high priority to the concentrating PV industry. If the waste heat is not being dissipated properly, a risk of cell failure can become an issue and increase maintenance costs in the long run. Edenburn mentions this concern about cell failures during high ambient temperatures, high insolation, and wind speeds around 0 m/s force the heat sinks to rely only on radiation and free convection. His resolution to this issue was to defocus the array or take it "Off-Sun," until the weather conditions were in a safe region for operation.

Table 3. Optimum heat exchanger parameters for point focus Fresnel lens array by Edenburn [10].

Lens Area (m ²)	Concentration Ratio	Cell Diameter (m)	Base Area (m ²)	Base Thickness (m)	Heat Exchanger Area Ratio
0.0058	50	0.0122	0.0025	0.0009	2.5
	92	0.009	0.0025	0.0012	2.5
	170	0.0066	0.0025	0.0012	2.5
0.0233	050	0.0244	0.008	0.0018	3.0
	92	0.018	0.008	0.0024	3.0
	170	0.0132	0.008	0.0026	3.0
0.093	50	0.0487	0.024	0.0036	3.5
	92	0.0359	0.024	0.0050	3.5
	170	0.0269	0.024	0.0060	3.5

Table 4. Optimum heat-exchanger cost and performance for point focus Fresnel lens array by Edenburn [10].

(Array Cost Without Heat Exchanger = \$138/m ²)					
Lens Area (m ²)	Concentration Ratio	Annual Energy (kWh/m ²)	Annual Energy Cells at 28°C (kWh/m ²)	Heat Exchanger Cost (\$/m ²)	Power Cost (\$/W)
0.0058	50	303	314	6.93	1.25
	92	305	314	7.31	1.24
	170	302	314	7.60	1.26
0.0233	50	301	314	8.60	1.27
	92	301	314	9.63	1.28
	170	300	314	9.93	1.29
0.093	50	291	314	9.31	1.32
	92	290	314	11.25	1.34
	170	290	314	12.58	1.36

Edenburn [7] studied not using a finned heat exchanger design and using only a aluminum boxed shaped housing to dissipate waste heat, which would reduce costs and cause a 3% reduction in energy generation for his particular cell and concentration ratio design. The aluminum housing would act as a housing and heat exchanger for the cell so that the finned heat sinks would not be needed. For still air conditions, radiation is the dominant mode of heat transfer making a flat heat exchanger more beneficial compared to a finned design. Edenburn [7] implemented this concept but painted the aluminum surface to change the infrared emittance from 0.1 to a higher value. While using a painted aluminum box, Edenburn believed that the cells would no longer reach degrading temperatures and defocusing would not be needed. However annual winds should be considered to approximate the amount of potential generation time during still air conditions. He later determined that minimal, or no wind speed conditions, were not a large portion of the annual conditions in potential installation sites, resulting in the need for heat sinks in the passive cooling design.

Figure 8. Point focus concentrator design that used a flat plate heat exchanger to dissipate waste heat. [10].

Table 5. Edenburn's results for the heat sinks temperature behind the cell at different concentration ratios.

Concentration Ratio	Substrate Temperature Above Ambient (K)
170	73
380	75
1000	78
1500	81
3400	85
13700	91

Edenburn studied concentration ratios up to 170 and determined that out of the two ways to increase the ratio: (1) keep the cell size constant and increase the lens size and (2) keep lens size constant and decrease cell size. His studies showed that as a lens size is increases the concentrated sunlight increases while maintaining the same cell size a passive cooling system is not efficient enough in removing the waste heat. This will require the use of an active cooling system. If the opposite is done and the cell size is reduced, this problem does not occur and active cooling is not needed. Table 5 shows Edenburn's results for heat sink temperatures behind cells at different concentration ratios with meteorological conditions at an irradiance of 800 W/m^2 and wind speeds approximately of 3 m/s with a lens area of 0.093m^2 .

The Martin Marietta system is a point focus concentrator design that over 575 kW installed by 1982 [13]. This was the largest amount of point focusing concentrators installed in the field at that time. Figure 10 is a schematic of the Martin Marietta point focus concentrating PV system that used a passive cooling system to dissipate the heat. Edenburn [12] participated in the development and implemented his research results previously discussed in to the design. This design used the optimum concentration ratio for the cell and lens size based on cell efficiency determined in his previous study [7]. This used heat sinks to cool the cells as shown in Figure 9. Although a study by Edenburn implies that under still air conditions, radiation is the dominant method of removing waste heat and heat sinks may not be needed, this condition is not a predominant situation in the field, thus not applied to the design.

Figure 9. Martin Marietta heat sink design [12].

Figure 10. Point focus concentrating PV system that used passive cooling [12].

Yamaguchi [13] worked on the point focus concentrator that used the housing to dissipate the waste heat generated by the solar cells. This method is similar to what

Edenburn had discussed [7] where a flat plate can be more effective in removing waste heat under still air conditions. An image of Yamaguchi's design is shown in Figure 11.

Figure 11. Point focus concentrator design that uses dome shaped Fresnel lenses [13].

The multijunction cell concentrator shown in Figure 11 has a concentration ratio of 400:1 and is a 7000 cm² module. This is fabricated with 36 pieces of randomly-selected receivers connected in series and the same number of the newly developed dome-shape Fresnel lenses. Only the module walls dissipated the waste heat. Efficiencies up to 27% have been reached.

Linear Focusing Concentrator Designs

J.P. David designed a linear focusing concentrator [15] with the intentions of the system being completely autonomous and not cumbersome, viable in the worst weather conditions, easy to install and maintain and decrease photovoltaic peak kW costs. In 1980 when David was testing this concentrator, the price for silicon cells were \$5-\$10/W_{peak} and the U.S. Department of Energy (DOE) estimated price to drop to \$2.80/W_{peak} by 1982 [15]. Even with this large decrease in silicon costs it was still quite expensive compared to other energy resources, however concentrator designs may be a promising method to further reduce this.

Figure 12. Schematic view of linear concentrator discussed in reference [15] with its converter and segmented mirror design.

This design began operation in June of 1979 and operated for over 18 months. This CPV design had 30 glass mirrors to concentrate the sun's light to approximately 20 suns. The light is focused on to a linear array of PV cells as shown in Figure 12. The passive cooling system that David implemented into his design consists of many small aluminum tubes, each having a length of 10 cm, an outer diameter of 1 cm, an inner diameter of 0.8 cm. A Duralumin plate acted as the interconnection between the cooling system and the cell. The fins were mounted on one side to of the plate with the electronic insulator, copper plate, and cell on the other (Figure 13).

Figure 13. Schematic of the PV target area and passive cooling system [15].

Figure 14. Sketch of the system that David [15] discussed that has been installed at the University of Marseille.

Edenburn investigated linear passive cooling systems [7] as well and point focus. For the linear system the cells are cooled by conducting heat through the substrate bond and conduction paths as shown in Figure 15. Three design parameters were considered in optimizing linear concentrator arrays:

- Height
- Thickness

- Total heat exchange are to base area ratio

These parameters were used to optimize the design. Table 6 compares the three different concentration ratios, heat exchanger dimensions, costs, annual energy, and power costs together to help determine the best heat exchanger design for the linear array. The concentration ratios shown in this table were 20, 30, and 40. The most expensive design was \$1.60/W with a concentration ratio of 20:1. However increasing the concentration ratio to 40 would decrease costs by \$0.02/W and annual energy by 2kWh/m². This table implies that a linear concentrator with a concentration ratio of 30:1 would have the most benefit for low cost per watt (\$/W) and high annual energy considering the given parameters.

Figure 15. Linear concentrator design. [7]

Table 6. Comparison of different passively cooled parabolic linear concentrators [7].

Concentration Ratio	20	30	40
Cell Width (m)	0.0458	0.0305	0.0229
Mast Base Height (m)	0.035	0.035	0.035
Mast Height (m)	0.25	0.25	0.25
Area Ratio	7.50	7.0	6.0
Heat Exchanger Cost (\$/m ²)	23.3	21.3	19.5
Annual Energy (kWh/m ²)	264	264	262
Power Cost (\$/W)	1.60	1.58	1.58

Table 7 compares Edenburn's linear and point focus array systems. Three concentration ratios for the linear design were compared to the single point focus. This comparison showed that, for Edenburn's design, the linear array was much more expensive and the annual energy was lower. The point focus concentrator heat exchanger was significantly less than all of the passive designs and had a 13% potential increase in annual energy. The loss in energy generation is due to higher cell temperatures because of the large temperature differences across conduction paths between cells and the heat sink surfaces.

Table 7. Comparison of the linear and point focus system [7].

	Heat Exchanger Cost (\$/m ²)	Annual Energy (kWh/m ²)	Power Cost (\$/Watt)
Passive 20X Trough	23.3	264	1.60
Passive 30X Trough	21.3	264	1.58
Passive 40X Trough	19.5	262	1.58
Passive 0.0233 m ² 92X Lens	9.6	301	1.28

CHAPTER 3

METHODOLOGY

Numerical Modeling Software

In order to model the temperature distribution and air flow in and around the heat sinks and chamber a computational fluid dynamic (CFD) modeling software had to be used. Fluent is a CFD software, developed by ANSYS, that was used for this analysis. This software uses finite forward difference method to solve its models [24]. All of the programs that were created for this study ran on a PC Dell Dimension computer with a 3.20GHz processor and 3.75 gigabytes of memory.

The convergence criteria was 0.001 for the continuity equation, 0.001 in the x-velocity direction, 0.001 in the y-velocity, 0.000001 for the energy equation, 0.001 for thermal conductivity, 0.001 for epsilon in the turbulence model, and 0.000001 for the radiation model. With these given convergence criterias each model ran for approximately 1 week and converged at approximately 28,000 iterations.

Gambit is the pre-processing software that ANSYS developed for Fluent. This is used to create models and mesh systems that were imported into Fluent. Mesh systems determine what points will be analyzed for the parameters that are under investigation. Post processing is done through Techplot, a post processing software for CFD and numerical simulation software. Tecplot is developed by Techplot and can be used with Fluent to create plots and charts for converged solutions.

Numerical Approach

The numerical model was developed based on the current HCPV design. However since it is difficult to model complicated systems, some assumptions were made. These assumptions are discussed and made based on the actual design. All of the parameters given are comparable to real operating conditions that the passive cooling system experiences in the field.

Energy enters the system at approximately 950 W/m^2 via sunlight at the Fresnel lens. With the lens area being 49 in^2 (0.03161 m^2) the total amount of energy entering the lens possible is 30.03 W . The lens transmission is estimated to be 90% providing a total potential 27.03 W of energy transmitted and concentrated to the cell area. Not all of the energy is concentrated onto the cell and it is estimated that approximately 3% is focused onto other areas decreasing the estimated energy to 26.22 W . The cell is estimated to have a 20% efficiency of sunlight to electricity conversion creating 5.24 W of electricity and 20.97 W of thermal energy. The thermal energy is dissipated through the cell to the receiver plate and heat sink is being investigated here.

The energy equation was determined first in order to get a understanding of the passive cooling system. Sunlight enters the systems at the solar cell; from here the energy is turned into electricity and thermal energy. The thermal energy is conducted through the solar cell and to the receiver plate. The receiver plate dissipates the thermal energy by conduction to the heat sinks. The receiver plate and heat sink surfaces dissipate heat to the surrounding air by both conduction and radiation.

Figure 16. Sketch of the heat transfer in the passive cooling system.

Heat loss through radiation is given in equation (1). The emissivity of the Amonix cell was assumed to be 1. This is not the real emissivity value of the cell, however for this analysis it was assumed to be so. Solar cells absorb most the energy that hits the surface but it is very hard to determine how much is being absorbed and reflected. The Stefan-

Boltzman constant, σ , is multiplied by the emissivity, ϵ , and the difference of the temperatures to the fourth power to calculate the heat loss through radiation. T_s is the surface temperature of the body that is radiating heat and T_∞ is the ambient temperature.

$$q_{rad} = A\epsilon\sigma(T_s^4 - T_\infty^4) \quad (1)$$

$$q_{conv} = Ah \cdot (T_s - T_\infty) \quad (2)$$

$$h = \frac{q}{A(T_s - T_\infty)} \quad (3)$$

$$I - q_{cond} - P = 0 \quad (4)$$

$$q_{cond} = -Ak(T_{heat\ sink} - T_{cell}) \quad (5)$$

$$q_{cond} - q_{conv} - q_{rad} = 0 \quad (6)$$

Equation (2) can be used to calculate the thermal energy removed by the cooling system by convection. Typically the values for heat transfer coefficient, h , in gases for free convection situations range between 2-25 W/(m²K) [16]. For this study an approximate value of 2.02 W/m²K was found for equation (4) using equation (3) and correlated with free convection values discussed by Incropera [16]. Equation (4) is the energy balance equation for the cell side of the system [1] and includes the sun energy reaching the cell, I , the amount of electricity generated, P , and thermal energy that is conducted through the cell to the receiver plate. Conduction through the heat sinks can be

calculated for by using equation (5). At the heat sink surface the energy is being transfer to the atmosphere by convection and radiation and is calculated for using equation (6).

Figure 17. Chamber schematic.

The passive cooling system models were developed based on the physical system. Since the system has been evaluated numerically using Fluent, there are some differences between the model and the actual system that were made in order to facilitate the analysis. In the physical system there are 48 chambers in a 4 X 12 array (Figure 5) that make up a single module. However, since all of the chambers are the same, the analytical model was simplified to a single chamber (Figure 17). It should be noted that this model assumes that the chamber thermal behavior is the same throughout the module, which is not necessarily true. Even though the geometries are the same, the temperature distribution will vary because of the air flow up the back of the module. In addition to the vertical temperature variation between chambers there are also some noticeable

differences along the horizontal direction. The exterior chambers with a side to the ambient air tend to have lower inside air temperatures compared to the interior chambers. A chamber has 24 cells in a 4 X 6 array, so it was simplified to a 6 cell model so that it could be analyzed as a two dimensional model (Figure 18).

The chamber model includes the heat sinks, receiver plate, chamber walls and Fresnel lenses. Ambient air has been modeled around the outside of the single chamber. The worst case scenario was assumed for this study (Table 8). In southern Nevada, a high ambient temperature (40°C) and no wind (speeds less than 1m/s) conditions are considered the worst case situation. With high ambient temperatures the temperature differential decreases thus decreasing the rate of heat being removed from the system. If air is flowing across the heat sink surfaces, the rate of heat removal from the system is increased, so without winds the performance is minimized. The cell is considered to be an isothermal heat source. This is a reasonable assumption because on a cloudless day, the solar irradiance varies very little during the mid part of the day and this is the time of most interest. By considering the irradiance-to-electricity conversion constant, the analysis is simplified by assuming the cell is in a steady state situation. The P-1 radiation model was used to simulate radiation effects in the system [24] and estimates the radiation emitted from one surface to another.

Table 8. Boundary Conditions

Ambient air is 313K
Isothermal heat source at the cell (353K)
Coupled conditions on all surfaces
No wind
Steady State

Figure 18. 2D Chamber drawing of the region analyzed.

Table 9. Material Properties

Material	Density (kg/m ³)	Specific Heat Capacity (J/kgK)	Thermal Conductivity (W/mK)	Dynamic Viscosity (kg/(m-s))
Air	1.1	1006.43	0.024	1.79 x 10 ⁻⁵
Steel	8030	502.48	16.27	
Acrylic	1149	1461.19	0.190	
Aluminum	2719	871	202.4	

Materials modeled in the chamber were the same as the actual system. The heat sink and receiver plate are brushed 2024 aluminum, the chamber structure or sides are galvanized steel, and the Fresnel lenses are acrylic (Table 9). Air was assumed to be an ideal gas. The standard k - ε turbulence model was used to simulate air flow. This semi-empirical model is based on transport equations for turbulent kinetic energy and its dissipation rate. For this model it is assumed that the flow is fully turbulent and the

effects of molecular viscosity are negligible [24]. The Rayleigh Number (7) is used for convection studies and was used to determine if the standard k-ε turbulence model was appropriate for this analysis. A Rayleigh number of $\sim 5 \times 10^9$ was calculated to determine that the flow is turbulent. The P1 radiation model is a series expansion that was used to simulate radiation between surfaces and to the surrounding area.

$$Ra = Gr \cdot Pr \quad (7)$$

$$Gr = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} = \frac{\text{bouyancy_forces}}{\text{viscous_forces}} \quad (8)$$

$$Pr = \frac{\nu}{\alpha} = \frac{\text{viscous_diffusion_rate}}{\text{thermal_diffusion_rate}} \quad (9)$$

Figure 19. Rayleigh Number at different characteristic lengths.

The mesh model (Figure 20) that was analyzed was created to scale based on the chamber drawing (Figure 18). The preprocessor software, Gambit was used to build this model. Over 250,000 nodes are in this mesh system with the grid size ranging from 0.05 to 0.002. The chamber is located in the center of the mesh system.

Figure 20. Mesh system.

Figure 21. Mesh around the chamber.

The mesh is very dense in and around the chamber relative to the whole air mesh system (Figure 20). Around the areas of highest concern, the mesh was made very dense (Figure 22). The ambient air at the system limits is over a meter away from the area under investigation, the nodes in that region are farther apart. The energy being dissipated by the heat sink is the main objective in this study causing the average mesh sizes to be 0.003 and the densest area in the model.

Figure 22. Mesh around the heat sink, cell area, and inside a portion of the chamber.

Experimental Method

Temperature data was collected in the field using thermocouples and hot wire anemometers. Three experiments were conducted in the field. Ten different chambers on two different modules were monitored for two days with thermocouples. This included two bottom and top chambers on each of the modules. One middle chamber was instrumented in each of the modules as well. All of the chambers that were monitored are labeled in Figure 23.

Type K thermocouples were used for collecting the temperature measurements in the areas shown in Figure 23. The thermocouples were 36 inches long with glass braided insulation and stripped lead terminations. A Campbell Scientific CR21X data logger was used to collect and store the data once every 30 seconds. Three temperature measurements were taken behind the solar cell on the heat sink and thermal tape was used to keep the sensor in contact with the surface (Figure 25).

Figure 23. Module schematic of where temperatures were measured.

Temperatures and wind speeds inside and around the chamber were taken with a combined hot wire anemometer and temperature sensor (Figure 24). This sensor records data 9mm above the desired surface with an uncertainty of $\pm 2\%$ for winds (0-50m/s) and ± 1 degree temperature (30 to +150 degrees Celsius) [26]. Figure 25 is an image of part of the hot wire anemometer experiment and shows a sensor mounted on the heat sink. This sensor is C4. There are two other sensors mounted on the receiver plate as well, C1 and C2, but can not be seen because the heat sink is in front of them. Data were collected at eight different locations and compared to the DNI, ambient temperature, and wind speed to search for correlations. The eight different measuring points are shown in Figure 26.

Figure 24. Image of the hot wire anemometer that was used for collecting temperature and wind speed data.

Figure 25. Image of part of a receiver plate and heat sink with a hot wire anemometer mounted on the end of the heat sink.

Figure 26. Locations on the chamber that were measured with the hot wire anemometer and temperature sensors.

CHAPTER 4

RESULTS

The current Amonix passive cooling system was analyzed numerically and experimentally and the results were compared with the goal of obtaining differences less than 10%. Once calculated the numerical model could be used to obtain more information on the air flow around the heat sinks and determine areas of poor circulation. The air flow details inside the chambers were calculated as well to determine if an increase in convection could result in an increase in performance. The results shown in this study may provide assistance in improving the passive cooling design in the short and long term and lead to an increase in energy generation per unit area.

Numerical Results

Because the Amonix HCPV unit is a 2-axis tracking system, the elevation angle changes throughout the day as it tracks the sun. In order to use one mesh system to simulate many elevation angles, the gravity vector angle was changed in the CFD software. Four different elevation angles were studied; 22.5°, 45°, and 67.5°, and 90° (vertical). Temperature distributions and velocity vector plots have been calculated at each of these angles. Figure 27 to Figure 41 show the results that were numerically obtained with Fluent. Each figure includes different views of the temperature distribution and the associated velocity profile for four elevation angles.

Figure 27. Temperature distribution of the chamber at a 90° elevation angle.

Figure 28. Velocity vector plot of the chamber at a 90° elevation angle.

Each of the temperature distribution figures, that were produced as output from Fluent was used to create a temperature plot, similar to what is shown in Figure 30. The profiles of the inside and outside of the chamber at different cross-sectional locations were compared to the other profiles and experimental data.

The air velocity vector information shown in Figure 28 is a visual illustration of how the air is moving in the system at this angle. In all of the velocity vector plots shown the air is moving at a rate ranging from 2.20×10^{-5} to 3.81×10^{-1} m/s. with the peak speeds to the left and right of the chamber. The average air velocity inside the chamber is approximately 1.14×10^{-1} and 2.29×10^{-2} around the heat sinks. Out of the four angles studied here, the one shown in Figure 28 is the only position that is not reached while generating energy. This is the most wind resistant position, also called “wind stow.” Although this orientation is not reached while generating energy, it can occur any time the wind speeds exceed the design operating conditions. While the Amonix HCPV system is tracking the sun, this angle is almost reached during operation in the summer when the sun’s elevation is high. This elevation angle is important to this study in two ways; first, for understanding the passive cooling systems as it approaches this angle, and for understanding the dissipating heat shortly after generation has stopped.

The temperature data shown in Figure 27 are important to illustrating the temperature distribution, but another method of extracting values was needed for direct comparisons. In all of the temperature data plots, the results at the line $x=6.70$ (Figure 29) have the most fluctuation because it is located between the receiver plate and the tip of the heat sink. This line includes the temperature of the heat sink and the air very close to it. The

line was very important in determining the free convective heat transfer coefficients as part of the simulation verification process.

Figure 29. The temperature values for the points in these lines were compared at the different elevation angles.

The temperature distributions for the heat sinks and receiver plate at the 90° degree elevation angle are plotted in Figure 31. It is important to know the difference in temperature of the heat sinks to the receiver plate because this gives a visual performance estimate for the passive cooling system. Small temperature differences may indicate there is a poor passive cooling design and if this is the case it is important to understand why. Since the chamber tracks the sun, it is important to account for reduction in heat dissipation due to position and determine the overall performance by analyzing all angles.

Figure 30. Plot of temperature distribution in and near the outside of the chamber at a 90° elevation angle.

Figure 31. Plot of the receiver plate and heat sink temperature distribution of the chamber at a 90° elevation angle.

Figure 32. Temperature distribution plot of the chamber at a 67.5° elevation angle.

Figure 33. Velocity vector plot of the chamber at a 67.5° elevation angle.

Figure 34. Plot of temperature distribution in and near the outside of the chamber at a 67.5° elevation angle.

Figure 35. Plot of the receiver plate and heat sink temperature distribution of the chamber at a 67.5° elevation angle.

Figure 36. Temperature distribution plot of the chamber at 45° elevation angle.

Figure 37. Velocity vector plot of the chamber at 45° elevation angle.

Figure 38. Plot of temperature distribution in and near the outside of the chamber at a 45° elevation angle.

Figure 39. Plot of the receiver plate and heat sink temperature distribution of the chamber at a 45° elevation angle.

Figure 40. Temperature distribution plot of the chamber at a 22.5° degree angle.

Figure 41. Velocity vector plot of the chamber at a 22.5° degree angle.

Figure 42. Plot of temperature distribution in and near the outside of the chamber at a 22.5° elevation angle.

Figure 43. Plot of the receiver plate and heat sink temperature distribution of the chamber at a 22.5° elevation angle.

Figure 44. Air temperature behind the heat sinks at $x=-6.50$ at four different elevation angles.

Figure 44 to Figure 50 are plots of the temperature lines for each angle position. They were plotted together to illustrate the estimated change in temperature as elevation angles vary over the course of a day. Figure 44 and Figure 45 show that the change in temperature behind the chamber increases as the elevation angle decreases. This

indicates that as the chamber elevation increases, less heat is being dissipated out of the passive cooling system. This is expected because as the chamber reaches a horizontal position more heat is being dissipated upwards into the chamber rather than away from the chamber via the heat sinks.

Figure 45. Air temperature behind the heat sinks at $x=-6.60$ at four different elevation angles.

The temperature values at $x=-6.70$ are plotted in Figure 46. This figure illustrates the temperature distribution in a critical area around the heat sinks. The peaks are temperature values of the heat sinks and other values are the air temperatures. At $Y=0$ the air temperature is approximately 313K or the ambient temperature, and increases as it reaches the heat sinks at 0.39. As the heated air surrounding the heat sinks rise along the chamber it increases, beginning at 325K at 0.39 up to 340K at 1.7.

Figure 46. Air and heat sink temperatures behind the chamber at $x=-6.70$ at four different elevation angles.

Figure 47 is a chart of the temperature distribution at $x=-6.75$ in and around the chamber. At the 90° degree elevation angle a peak temperature is reached at the center ($x=1.18$) of the chamber where there is a stagnant area between two vortices. Air is unable to circulate in this small area causing the temperature to increase. Peak

temperatures are reached at the top of the chamber (~1.60m) for the 22.5°, 45°, and 67.5°, degree elevation angles

Figure 47. Air chamber temperatures in front of the cells at x=-6.75 at four different elevation angles.

The peak temperature in the 90° degree elevation angle is as visible in Figure 48 as in Figure 47. The temperature peaks of the other three elevation angles are in the same place inside the upper portion of the chamber as shown in Figure 47 as well. At 22.5° degree elevation the highest chamber temperature is reached around the top of the chamber at approximately 338.5K.

Figure 48. Air chamber temperatures at x=-6.80 at four different elevation angles.

Figure 49. Air chamber temperatures at $x=-6.90$ at four different elevation angles.

Figure 50. Air chamber temperatures at x=-7.00 at four different elevation angles.

Table 10. Average heat sink temperatures

Elevation Angle	Average Heat Sink Temperature (K)	Minimum Heat Sink Temperature (K)
22.5	348.30	345.41
45	349.06	346.47
67.5	349.08	346.44
90	347.99	345.40

The average heat sink temperature is lower at 90° and 22.5° degrees in elevation than at 45° and 67.5° degrees. At a steep angle similar to 22.5° degrees more heat is being dissipated straight up the chamber causing an increase in the temperature difference between cell and heat sink minimum temperature. A similar effect occurs when the chamber is close to a horizontal position like 90°. This indicates that the conduction and convection is increasing for these elevation angles showing that less heat is being transferred through the heat sink and more to the inside of the chamber. There is an approximate 2% increase in temperature difference between the 22.5° and 90° degree position versus the elevation angles of 45° to 67.5°.

Experimental Results

Figure 53 to Figure 58 show temperature data that was collected from the five different top chambers (T1-T8) on a system in Arizona at a Arizona Public Service solar plant, on July 24, 2002. This was collected over an entire day so that the temperature over that time period could be evaluated. Ambient temperature trends in the American southwest typically show higher temperatures in the evening compared to the morning, which also appear to have an effect on the temperature distribution for the chambers. Figure 53 shows a sudden increase in temperature at approximately 13:40, and could have been caused by an increase in solar insolation (Figure 51), or turning the PV strings off line. If the strings on the system were disconnected from the grid the amount of energy could have been generated would be turned into waste heat.

The temperature probe used to measure ambient temperature at the Arizona Public Service weather station was not measuring the data correctly. The measured data is

approximately 5-7 degrees less than the actual temperature. A corrected ambient temperature profile has been plotted in Figure 51. This corrected data was verified with the ambient temperature data collected with the thermocouple measurements used in for monitoring the chambers and is shown in Figure 56.

Figure 51. The measured ambient temperature, corrected ambient temperature and Direct Normal Irradiance (DNI) for July 24, 2002.

Figure 54 is a plot of two corner chambers that face each other on two modules next to each other. Some possible reasons for the difference in temperatures may be due

to open air wind speeds and direction on the system. As the wind moves across the back side of the system the air temperature increases as it removes thermal energy. If the temperature of T3 is higher than T2, the air moved across chamber T2 first.

Figure 52. Change in elevation angle during the day on July 24, 2002.

Figure 55 shows how chamber temperatures increase from one chamber to another going in the upward direction of the Module. T2 is higher than T5 during most of the day. Temperatures were also monitored on the heat sinks behind the cell and show how high temperature can be at times (Figure 56). T7 lost contact with the heat sink surface in the afternoon and began reading ambient temperature. One other thermocouple, T8, remained in contact with its surface as shown in Figure 56.

Figure 53. Plot of the inner column chamber temperatures; T1 and T4.

Figure 54. Plot of two outer chamber temperatures; T2 and T3.

Figure 55. Two outside module chamber temperatures; T2 and T5.

Figure 56. Two receiver plate temperatures taken behind the cell; T7 and T8.

Figure 57. Chamber air temperatures for two top chambers on the top of one module.

Figure 58. Two different chamber air temperatures at the top of a module.

Figure 57 and Figure 58 illustrate effects similar to Figure 54. T1 was 2-6 degrees higher than T2 for most of the day. In addition to the wind direction having an effect on temperature, the chamber placement on the module can cause variations as well. The outside module chambers have a side surface exposed to the ambient temperature, therefore there is more area to dissipate heat to the environment.

Figure 59. The ambient temperature, corrected ambient temperature, and DNI for July 25, 2002.

The temperature was monitored in the lower chambers of two different modules on July 25, 2002. The ambient temperature and solar insolation for this day is plotted in Figure 59. The weather data plotted in this figure is from the same weather station at the Arizona Public Service solar plant that is shown in Figure 51. The ambient temperature

data for this day is approximately 5-7 K lower than the actual temperature and the corrected data is shown in this plot.

Figure 60. Change in elevation angle during the day on July 25, 2006.

All of the lower chamber measurements are noted with an, “a” at the end of their label. Figure 61 is a plot of a two different module lower inner chambers, T1a and T4a. T1a and T4a data were within a few degrees of each other for most of the day. Similar results were found for T2a and T3a which were two different lower Module chambers as well. The temperature data collected for both T2a and T3a are shown in Figure 62. Figure

61 and Figure 62 plot may indicate the direction of the air flow across the chambers. T4a data is higher than T1a and T2a in the late afternoon indicating that the air had moved across the module with those chambers before reaching T4a.

Figure 61. Comparison of two chamber temperatures; T1a and T4a.

Chamber temperatures, T1a and T2a, for two lower chambers on a different module were plotted together to compare temperature variation over the day. These two chamber temperatures were compared to T3a and T4a chambers to study differences in different chambers based on position on a module. T3a and T4a were plotted together to compare two chambers next to each other. T1a and T2a appear to follow along with the assumption that the air flowed across T1a before reaching T2a because T2a is hotter in the late afternoon. However the data plotted in Figure 65 suggests that the air is flowing in the opposite direction because T3a is higher than T4a during the same time period.

This may be due to the angle of attack of the air on the T3a thermocouple causing a reduction in air flow at this point.

Figure 62. Comparison of two chamber temperatures; T2a and T3a.

Figure 63. Comparison of two lower chamber temperatures T1a and T2a.

Figure 64. Comparison of two lower chamber temperatures T3a and T4a.

Figure 65. Comparison of two lower chamber temperatures T3a and T5a.

T3a and T5a were plotted and compared to illustrate the difference in chamber temperature along the module in the upwards direction. The two chambers had a seven to eight degree difference in temperature over the day as shown in Figure 65. The chamber with thermocouple T5a was located two chambers above T3a.

Thermocouple T7a was located behind the receiver plate behind a cell. At approximately 13:00 the thermocouple lost contact with the receiver plate and began measuring ambient temperature, causing sudden decrease. This data was also plotted with T4a so that the temperature difference between the chamber and heat sink temperature could be compared. Only a short period of time is valuable for this comparison and shows how the heat sink temperature is higher than the chamber, which is expected. Situations where the chamber temperatures are higher than the heat sink indicate poor surface contact and heat dissipation.

Figure 66. Plot of T7a temperature data.

Figure 67. Ambient temperature and DNI for July 19, 2007.

The temperature was monitored with the hot wire anemometer sensors in one lower chamber on July 19, 2007 for eight hours. Each sensor is monitors both air temperature, C_n , and velocity, V_n . Figure 68 and 69 shows the temperature relative to chamber elevation angle. Even though the passive cooling system is dissipating more heat after solar noon because of its position, the temperatures do not decrease because the ambient temperature is increasing. A correlation between the wind speed and temperature was

sought (Figure 70), however nothing was discovered. Although no correlation was found in this study, this does not imply that there is none.

Figure 68. Temperature data for the sensors mounted inside the chamber compared to the ambient temperature and chamber elevation angle.

Figure 69. Temperature data for the sensors mounted outside the chamber compared to the ambient temperature and chamber elevation angle.

Figure 72 is a comparison plot of the temperatures compared to the DNI including the ambient temperature during the day. As the day approaches solar noon the DNI reaches the peak values but the ambient and chamber temperatures do not reach a peak until the late afternoon (approximately 15:00).

Figure 70. Temperature data from sensors mounted outside of the chamber compared to the wind speed.

Figure 71. Temperature data from sensors mounted inside of the chamber compared to the wind speed.

Figure 72. Cell temperature data compared to the DNI and ambient temperature.

The wind speeds measured in and around the chamber are compared to the weather station anemometer data in Figure 73. None of the measured data around the chamber is

as high as the weather station data but the sensors mounted on the outside of the chamber (V1, V2, V4 and V7) have a closer correlation than the inside temperatures. The measurements taken inside the chamber are low and do not differentiate more than 0.5m/s, which is expected for normal operation.

Figure 73. Measured wind speed data comparison.

CHAPTER 5

FINDINGS OF THE STUDY

Comparing the numerical and experimental results was an important goal for this analysis in order to verify the analytical model results. Heat sink temperature measurements were taken during meteorological conditions that were similar to those simulated in the numerical model. Once the analytical model was verified, it could be used to investigate ways of improving the removal of waste heat.

Figure 74. Numerical and experimental data of the chamber temperatures.

The data for each of the chambers were compared to the results obtained from the numerical analysis. The data displayed in these comparison plots show the Fluent results and experimental data at the same elevation angle. The chamber temperatures are not the same throughout, which is shown in the numerical results and the comparison plots (Figure 74-Figure 77). The experimental results do not vary as much as the numerical ones because the measurements were taken from a single location throughout the day. However, the average experimental chamber air temperature results are within 6% of the numerical ones.

Figure 75. Numerical and experimental data of the chamber temperatures.

Experimental chamber air measurements were close to the numerical ones. However the cell/receiver plate temperatures were not as close. The cell/receiver plate measurements varied from 0.20%-11.50% lower than the numerical ones as shown in Table 11 and Figure 76 and Figure 77. Part of the deviation occurred because the thermocouples that were used to measure these parameters lost contact with the surface. When good contact was maintained with the heat sink surface variations were within 3%. When contact was lost the temperature difference increased to approximately 10% as shown in Table 11.

Figure 76. Numerical and experimental data of the cell/receiver plate temperatures.

The percent deviation values presented in Table 11 are calculated from the experimental data presented in Figure 74 through Figure 77 with the numerical data obtained from Fluent. The values were calculated using equation (10).

$$\frac{\text{experimental} - \text{theoretical}}{\text{experimental}} \times 100 \quad (10)$$

Table 11. Difference in temperature between the numerical and experimental data with the thermocouple experiment.

Elevation Angle (degrees)	Chamber			Cell/ Receiver Plate		
	average	maximum	minimum	Average	maximum	minimum
22.5	-4.4%	-2.3%	-6.7%	-12.7%	-11.2%	-14.0%
45	1.3%	4.3%	-2.2%	-3.0%	0.2%	-7.2%
67.5	-1.0%	3.0%	-2.6%	-5.1%	-3.1%	-7.2%
67.5	-1.3%	2.7%	-2.7%	-8.5%	-6.1%	-10.9%
45	-1.5%	1.8%	-3.1%	-8.1%	-0.2%	-12.1%
22.5	-3.0%	-0.6%	-5.1%	-11.7%	-10.0%	-13.0%

Some potential reasons these variations between the model and field data are listed:

- Wind speeds-not always zero

The numerical model was simulated for a worst case scenario and assumed the wind speed was zero. If the wind was present at the time the measurement was taken, then the heat sink temperature could have decreased.

- Surface contact not complete

Complete surface contact between the cell package, receiver plate and heat sink is extremely important to have good heat dissipation. If there is not complete contact, it is possible that heat flow could be decreased by these small gaps and be

unable to dissipate as well to the heat sink. This would decrease the performance of the cells and the heat sink temperature.

- Steady State conditions not always attained

Variations in the flux on the cell can lower the cell temperature thus lowering the cell and receiver plate temperatures. It is common for the sun's irradiance to vary, thus decreasing the power output and the cell and receiver plate temperatures. The model in this study considered the cell temperature to be constant which does not take this effect in to consideration.

Figure 77. Numerical and experimental data of the cell/receiver plate temperatures.

- Tracking error

Variations in tracking can cause variations in the flux on the cell to change. The Amonix HCPV system requires dual axis tracking to generate energy. High concentrating systems require high tracking accuracy, thus making it possible to cause flux variations on the cells, which could cause lower cell/ receiver plate temperatures.

The hot wire anemometer data collected in the field had a 0-11% variation compared to the numerical data (Figure 78 through Figure 85). In the morning when the ambient temperature had not reached 313K, the difference between field data and analytical results were the largest at 11% but as the day progressed this decreased. In the morning, the differences between the results is 2.35% to 10.80% with location C5 as the closest value to the numerical results. These variations were calculated by using equation (10). However as the ambient temperature increases, the variation reduces to an average 2.30% at 67.5° degrees and 1.80% at 22.5° degrees in the afternoon.

Table 12. Difference in temperature between the numerical and experimental data with the hot wire anemometer experiment.

Elevation Angle (degrees)	C1	C2	C3	C4	C5	C6	C7	C8
45	-10.8%	-9.9%	-4.1%	-5.7%	-2.5%	-4.3%	-4.3%	-4.3%
67.5	-10.1%	-8.6%	-1.9%	-4.2%	-2.1%	-3.5%	-2.4%	-3.6%
67.5	-6.8%	-6.5%	0.3%	-2.3%	-0.8%	-2.2%	0.1%	-1.3%
45	-4.4%	-5.1%	1.0%	-1.7%	0.9%	-1.4%	-0.8%	-0.9%
Average	-8.0%	-7.5%	-1.2%	-3.5%	-1.1%	-2.9%	-1.9%	-2.5%

Sensor C1 and C2 had the largest deviation from the Fluent results. These sensors were located on the chamber being on the heat sink and exposed to the climate conditions which might explain the large difference. Figure 73 compares all of the hot wire anemometer data to the open air wind speed data, and shows that V1 and V2 had correlating values which may be part of the reason why there is a discrepancy.

Figure 78. Numerical and experimental data at location C1.

Figure 78 shows that as the day carried on, the wind speeds measurements for V1 decreased and became more stable while C1 temperatures increased and approached the Fluent values. This shows that even very light winds can have a direct affect upon the

temperatures. The location of the C1 sensor was outside of the chamber, which means it was directly exposed to the wind conditions. V1 and the wind data shown in Figure 78 show a direct correlation.

Figure 79. Numerical and experimental data at location C2.

C2 is located outside of the chamber similar to C1 causing similar responses to the wind as shown in Figure 79. Although C2 is located outside of the chamber like C1, its location is above the heat sink as shown in Figure 26. From approximately 12:30 to

16:00, V2 does not fluctuate with the wind as much as during the morning which may be caused by the winds angle of attack. If the wind is contacting the chamber at the right angle, it could cause the heat sink to act as a barrier and reducing the wind effects and the variation between results.

Figure 80. Numerical and experimental data at location C3.

Sensor C3 is located above the top cell and inside the chamber. The results shown in Figure 80 had an approximate average of 1.2% difference compared to the numerical results. The largest variation occurred in the morning before the ambient temperature

reached 313K with a discrepancy of 4.1% but in the afternoon this reduced to approximately 1.0%. The wind current data inside the chamber (V3) is fairly steady compared to the open air wind speed. It is expected to see little variation in the interior air velocity data because the chamber is almost completely sealed from the outside conditions.

Figure 81. Numerical and experimental data for location C4.

Sensor C4 is located at the tip of the heat sink at the top of the chamber. The average difference between the numerical and experimental data is 3.5%. The variation was

initially 5.7% and decreased to 1.7% at the end of the day. The experimental ambient temperature approached the numerical model's at 313K this variation reduced. The wind speed measurements, V4, had a correlation with the open air wind speeds. In the morning this is more prevalent than the afternoon which may be due to the direction of the wind and its angle of attach on the back of the chamber.

Figure 82. Numerical and experimental data for the location C5.

Having a sensor on the tip of the heat sink was important for analyzing the heat dissipation rate. Although the variation between heat sink results ranged from 1.7% to 5.7%, these results gave a lot of insight to the passive cooling system performance. The air velocity data collected for V4 had a similar response to the open air wind speed as V1

and V2. In addition to having similar responses to the wind speeds the temperature data correlates to the ambient temperature as shown in Figure 81. This similarity is expected considering the location being on the bottom heat sink and on the tip. Heat rises up the chamber and since this is located at the bottom, no heat is rising out of the heat sinks and to the sensor. Thus the effect on the temperature of this sensor due to the passive cooling system and chamber is minimized causing the weather conditions to be the main influence.

Figure 83. Numerical and experimental data for location C6.

Sensor C5 V5 was located inside of the chamber on the top surface. Figure 82 is a plot of the temperature and air velocity data collected from the sensor, ambient temperature, wind speed, and Fluent temperature data at that point. This data had less than a 3% difference in temperature values when compared to the numerical model ones. The largest difference between results was in the morning when the ambient temperature had not reached 313K and there was a 2.4% difference. The chamber air velocity is steady throughout most of the day similar to the V3 data. The similarity between V3 and V5 air velocity data is because of their location inside the chamber.

Figure 84. Numerical and experimental data for location C7.

Sensor C6, C7, and C8 data are plotted in Figure 83 through Figure 85. Sensor C6 and C8 were located inside the chamber and showed similar responses to the wind that V3 and V5 did. V7 was located on the outside of the chamber on the end of a heat sink causing it respond to wind conditions. Although V7 was more responsive to the wind than V6 and V8 all three of them had a less than 5% variation from the numerical results.

Figure 85. Numerical and experimental data for location C8.

Thermal radiation is energy emitted by matter that is at a finite temperature [16]. Part of the heat sink is facing the receiver plate causing radiation exchange heat transfer back to the receiver plate decreasing the energy dissipated to the surroundings. Estimating the amount of energy radiated from the heat sink was needed to determine if a significant

amount of waste heat is being radiated back to the system. Equation (11) is used to calculate how much energy was being reflected to the receiver plate. The view factors had to be calculated using information provided in Table 13 to compute equation (11). The view factors were calculated for the heat sink with the view factor reciprocity relation [16] to determine the fraction of radiation leaving the surface that is intercepted by another.

Figure 86. Radiation schematic of the passive cooling system.

Table 13. View factors for 2 dimensional geometries [25].

	$F_{1-2} = \frac{1}{2B} \left[\sqrt{(B+C)^2 + 4} - \sqrt{(C-B)^2 + 4} \right]$	$B = 0.011 \text{ m} \quad (11)$
		$C = 0.14 \text{ m}$
		$F_{1-2} = 0.91670$

		$\alpha = 75.5 \text{ degrees} \quad (12)$
		$A = 0.0338$
		$F_{2-1} = 0.056$

$$q_i = \sum_{j=1}^N A_i F_{ij} \sigma (T_i^4 - T_j^4) \quad (13)$$

The perimeter of the heat sink used to dissipate heat via radiation is approximately 0.19 meters and out of this total distance 0.05m (25%) is facing the receiver plate. The heat sinks can radiate up to approximately 5% of the total energy being dissipated by the heat sinks to its surroundings by radiation as shown in Table 14. Even though 25% percent of the heat sink surface is facing the receiver plate only about 5% out of the total amount of energy radiated at each elevation angle is reflected back into the receiver plate.

Table 14. Energy radiated out of the heat sink to ambient and back to the receiver plate.

Elevation Angle	Energy back into System (%)
22.5	3.91%
45	4.78%
67.5	4.80%
90	4.67%

CHAPTER 6

SUMMARY

The results between the experimental data and the numerical model results were compared and analyzed. The chamber air results had an average difference of 3%. This was a fairly close correlation between results, but the cell and plate results were not as accurate, having a variation of less than 7%. The data from when the thermocouples used for measuring the cell and receiver plate lost contact with the surface have not been considered in this variation. All of the temperature values solved for numerically and obtained experimentally correlate with each other. This information offers a understanding for the performance of the passive cooling system and indicate potential areas of improvement.

These results show satisfactory approximation for the air flow inside the chamber. Amonix has estimated that with a four degree decrease in temperature there is a potential increase of 1% annual energy generated. The cell temperature will not be lower than the surrounding air temperature or chamber air temperature. The chamber air temperature is over ten degrees Celsius above the ambient temperature throughout the day. With the air flow information future work could be done on improving this and decreasing the cell temperature. This future work could include ventilation studies so that the chamber temperatures may have less difference throughout the day.

The air flow around the heat sinks is important to understanding the passive cooling systems performance. In the velocity vector plots for each elevation angle, vortexes were developed approximately the same places. Vortexes developed at all of the elevation angles studied between the receiver plate and heat sink as shown in Figure 87. The recirculation of air is a poor heat dissipation area and is a potential area for improvement.

Figure 87. Plot of computed vortices around passive cooling system.

With the data provided in this study opportunities to improve the passive cooling design have been discovered. Although the heat sinks was the main area under

investigation for this study the chamber showed to have an important role in the heat dissipation rate as well. The chamber temperatures and air flow information indicate a potential for improvement.

Recommendations

This study showed some areas for improvement that could be implemented into the passive cooling system design that would improve the performance and have a minimal effect on the assembly process.

Figure 88. Image of heat sinks mounted in the back of a Amonix HCPV system in the horizontal direction.

The velocity vector results that are shown in Figure 28, Figure 33, Figure 37, and Figure 41 have vortices in between the receiver plate and the heat sink. One way to reduce these vortices would be to change the direction of the heat sink from horizontal (Figure 88) to vertical (Figure 89). Then the heat is dissipated out of the heat sinks and receiver plate allowing the surrounding air to heat up and rise with less impedance. Mounting the heat sinks in a vertical direction allows hot air to slide up the back rather than reticulating between the end of the heat sink and the receiver plate.

Figure 89. Image of the heat sinks mounted on the Amonix HCPV system in the vertical direction.

Another way to reduce these vortexes would be to increase the perforation in the sides of the heat sink. The current design is has small holes approximately one eighth of an inch in diameter and approximately six inches apart along the sides. By increasing the amount of holes and or the size of the current ones may decrease the vortexes building up and increase air flow. This potential modification has not been studied yet for optimum hole dimensions, distance apart or a cost analysis and would need to be before any change is made.

Another possible way to increase the amount of waste heat dissipated would be to change the sides of the heat sink so that the amount of energy radiated to the receiver plate is reduced. The sides of the heat sink were straightened so that the 75.5 degree angle is changed to 90 degrees and the bend at the tip is removed.

Current Heat Sink Design

Proposed Heat Sink Design

Figure 90. Heat Sink Design Comparison

The proposed heat sink design reduces the fraction of radiation leaving heat sink surface to the receiver plate to 0.07133 potentially radiating 78.4% less energy to the receiver plate. This heat sink design has the same base dimensions as the current one. For

the proposed design the same amount of material would be used because the length is the same except there is no extra bending on the sides.

Table 15. View Factor for the new 2 dimensional heat sink design [25]

	$F_{1-2} = \frac{1}{2} \left(1 + H - \sqrt{1 + H^2} \right)$	$H = 0.046$ (14)
$H = \frac{h}{w}$	$F_{2-1} = 0.07133$	

This proposed design has not been investigated and is based on the concept of reducing energy radiated back to the receiver plate and steps in manufacturing. With less bending in the proposed heat sink design there is a potential reduction in costs and at large and small quantities. However this design has not been studied to see if it is a benefit to the overall design and this should be done before any conversion is made. Although the proposed design does radiate less heat to the receiver plate there is an increase in energy radiated from one side of the heat sink the other which might cause an overall decrease in total thermal energy dissipated.

The suggestions described in this study would not have a large impact on Amonix's current manufacturing process. The first suggestion of changing the heat sink direction has already been implemented into the design and is being tested at the UNLV-CER and Nevada Power Clark Generation Station. Increasing the perforation and modifying the angle have not been implemented and must be investigated before any changes are made. Although not all of these suggestions are implemented they are simple and may cost

effective improvements that would increase the design. Amonix can use these results as a reference for future design studies for increasing performance.

REFERENCES

- [1] Royne, A., Dey, C., Mills, D., "Cooling of Photovoltaic Cells under Concentrated Illumination." *Solar Energy Materials & Solar Cells*. 2005 (451-483).
- [2] R. A. Sinton, Kwark, P. Gruenbaum, R. M. Swanson, "Silicon Point Contact Concentrator Solar Cells." Conference record, 18th IEEE PVSC, 1985 (61-65).
- [3] L. W. Flowschuetz, "On Heat Rejection from Terrestrial Solar Cell Arrays with Sunlight Concentration." 11th IEEE PVSC, New York, 1975 (318-326)
- [4] G. Sala. Ch. 8: Cooling of Solar cells, in: *Optics for Photovoltaic Concentration*, Adam Hilger, Bristol, 1989 (239-267).
- [5] M. J. O'Leary and L. D. Clements. "Thermal-Electric Performance Analysis for Actively Cooled Concentrating Photovoltaic System." *Solar Energy* 25, 1980 (401-406).
- [6] D. J. Mbewe, H. C. Card, D. C. Card, "A Model of Silicon Solar Cells for Concentrator Photovoltaic and Photovoltaic Thermal System Design", *Solar Energy*. V. 35 1985 (247-258).
- [7] M. W. Edenburn. "Active and Passive Cooling for Concentrating Photovoltaic Arrays." 14th IEEE PVSC, 1980 (771-776).
- [8] V. M. Andreev, V. A. Grilikhes, V. P. Khvostikov, O. A. Khvostikova, V. D. Rumyantsev, N. A. Sadchikov, M. Z. Shvarts, "Concentrator PV Modules and Solar Cells for TPV Systems." *Solar Energy Materials & Solar Cells*. 84, 2004: 3-17.
- [9] R. K. Mathur, D. R. Mehrotra, S. Mittal, and S. R. Dhariwal. "Thermal Non-Uniformities in Concentrator Solar Cells." *Solar Cells*. 11. 1984: 175-188.
- [10] A. Slade, V. Garboushian. "27.60% efficient Silicon Concentrator Solar Cell for Mass Production." 15th International Photovoltaic Science and Engineering Conference. October 2005, Shanghai, China.
- [11] B. D. Shafer, M. W. Edenburn, M. Garner, H. Togami. "Low- Cost, High Performance, Point Focus Concentrator Array Design." IEEE. CH1508. 1980.
- [12] F. K. Deaver, M. W. Edenburn. "Passive Cooling Measurements for a Concentrating Photovoltaic Array." *Journal of Solar Energy*. V. 105, November. 1983.

- [13] E. Boes, M. Edenburn. "Photovoltaic Concentrator Technology in the USA." 4th Photovoltaic Solar Energy Conference. May 1982. Stresa, Italy.
- [14] Masafumi Yamaguchi, Tatsuya Takamoto, and Kenji Araki. "Super High-Efficiency Multi-Junction and Concentrator Solar Cells." *Solar Energy Materials & Solar Cells*. 90 2006: 3068–3077.
- [15] J. P. David, F. Floret, J. Guerin, J. C. Paiva, and L. Aiache. "Autonomous Photovoltaic Converter with Linear Focusing Concentrator." *Solar Cells*. 4. 1981: 61-70.
- [16] F. P. Incopera, D. P. DeWitt. Introduction to Heat Transfer. 4th edition. John Wiley & Sons. 2002.
- [17] T. K. Mallick, P. C. Eames. "Design and Fabrication of Low Concentrating Second Generation." *Solar Energy Materials & Solar Cells*. 91. 2007: 597-608.
- [18] G. Sala, Cells and Optics for Photovoltaic Concentration, Adam Hilger, Bristol, 1989, (239-267).
- [19] V. L. Dalal, A. R. Moore, "Design Considerations for High-Intensity Solar Cells, *J. Applied Physics*. v48 1977 (1244-1251).
- [20] SunPower, Application notes for HED312 Silicon Concentrator Solar Cells, 2002.
- [21] W. E. Horne, Solar Energy System, patent US5269851, USA, 1993.
- [22] K. Stone, A. Gray, V. Garboushian, K. Markarian, R. Boehm, R. Hurt, M. Hale, H. Hayden, T. Fletcher. "Operation and Performance of the Amonix High Concentration Photovoltaic System at the University of Nevada, Las Vegas during the Second Year of Operation", ASME Solar Energy Conference, July 8-13, 2006, Denver Colorado.
- [23] R. Byron. R. Bird, Warren Stewart, Edwin Lightfoot. Transport Phenomena. 2nd Ed. New York: John Wiley & Sons, 2002.
- [24] FLUENT 6.2 User's Guide. Fluent Inc. 2005: 733-745.
- [25] R. Siegel, J. Howell. Thermal Radiation Heat Transfer. 4th Ed. Great Britain: Taylor & Francis, 2002.
- [26] "Candle Stick Sensor." Advanced Thermal Solutions Inc. Retrieved on September 21, 2007 from http://www.qats.com/productdata/Candlestick_sensor.pdf.

- [27] Akbarzadadeh, T. Wadowski. "Heat Pipe-Based Cooling Systems for Photovoltaic Cells under Concentrated Solar Radiation." *Applied Thermal Engineering*. Vol. 16. No. 1. 1996: 81-87.
- [28] Luque, G. Sala, J. C. Arboiro. "Electric and Thermal Model for Non-Uniformly Illuminated Concentration Cells." *Solar Energy Materials & Solar Cells*. 51. 1998:269-390.
- [29] D. L. Evans, L. W. Florschuetz. "Terrestrial Concentrating Photovoltaic Power System Studies." *Solar Energy*. Vol. 20. 1978:37-43.
- [30] D. L. Evans, L. W. Florschuetz. "Cost Studies on Terrestrial Photovoltaic Power Systems with Sunlight Concentration." *Solar Energy*. Vol. 19. 1977:255-262.
- [31] T. K. Mallick, P. C. Eames, B. Norton. "Using Air Flow to Alleviate Temperature Elevation in Solar Cells within Asymmetric Compound Parabolic Concentrators." *Solar Energy*. Vol. 81. 2007:173-184.
- [32] K. Araki, H. Uozumi, M. Yamaguchi. "A Simple Passive Cooling Structure and its Heat Analysis for 500X Concentration PV Module." *IEEE*. 2002:1568-1571.
- [33] D. Pachon, I. Anton, G. Sala. "Rating and Modeling of Concentrator Systems." *IEEE*. 2002:1600-1603.
- [34] Slade, R. Gordon, D. Dutra, V. Garboushian. "Long Term Reliability of Mass-Produced High-Efficiency Silicon Point-Contact Solar Cells under 250X Concentration." *IEEE*. 2002:1015-1018.
- [35] C. Kirchsteiger. "A New Approach to Quantitative Assessment of Reliability of Passive Systems." *Safety Science*. Vol. 43. 2005:771-777.

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A. Gray, K. Stone, R. Gordon, D. Dutra, R. Hurt, and R. Boehm. "Field Testing and Performance of an Amonix Multijunction Cell Module at the University of Nevada Las Vegas." ASME Energy Sustainability 2007. June 27-20, 2007, Long Beach, California.

A. Gray, M. J. Hale, A. Sahm, M. Newmarker, R. Hurt, R. Boehm, A. Andreas, P. Gotseff, and T. Stoffel. "Southern Nevada Renewable Resource Assessment." ASME Energy Sustainability 2007. June 27-20, 2007, Long Beach, California.

A. Gray, K. Stone, G. Wood, R. Boehm, and K. Johnson. "Assisting in Matching Nevada Power Company's Electrical Load Profiles with High Concentration PV Systems." ASME Energy Sustainability 2007. June 27-20, 2007, Long Beach, California.

K. Stone, A. Gray, V. Garboushian, K. Markarian, R. Boehm, R. Hurt, M. Hale, H. Hayden, T. Fletcher, "Operation and Performance of the Amonix High Concentration Photovoltaic System at the University of Nevada, Las Vegas During the Second Year of Operation", ASME Solar Energy Conference, July 8-13, 2006, Denver Colorado.

K. Stone, A. Gray, V. Garboushian, K. Markarian, R. Boehm, R. Hurt, G. Wood, H. Hayden, T. Fletcher, "Installation and Pperation of the Amonix High Concentration PV System at Nevada Power Company in Las Vegas, Nevada", IEEE 4th World Conference on Photovoltaic Energy Conversion, May 7-12, 2006, Waikoloa, Hawaii

V. Garboushian, K. Stone, R. Boehm, R. Hurt, A. Gray, H. Hayden, "Analysis of Five Years of Field Performance of the Amonix High Concentration PV System", Power Gen Renewable Conference, Las Vegas, April 11, 2006.

A. Sahm, A. Gray, R. Boehm, K. Stone, "Cleanliness Maintenance for an Amonix Lens System", ISEC2005-76036, 2005 International Solar Energy Conference, August 6-12, 2005, Orlando, Florida.

K. Stone, V. Garboushian, R. Boehm, R. Hurt, A. Gray, T. Fletcher, "Installation, Operation, and Performance of the Amonix High Concentration Photovoltaic System at the University of Nevada, Las Vegas", ISEC2005-7609, 2005 International Solar Energy Conference, August 6-12 2005, Orlando, Florida.

K. Stone, R. Hurt, T. Fletcher, V. Garboushian, A. Gray, H. Hayden, and R. Boehm, "Evaluation of the Amonix High Concentration Photovoltaic System after Five Years of Field Operation", 20th European Photovoltaic Solar Energy Conference, 6 - 10 June 2005, Barcelona, Spain.

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