EFFICIENT HEAT TRANSFER METHODS IN A HYBRID SOLAR THERMAL POWER SYSTEM FOR THE FSPOT-X PROJECT

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ABSTRACT

The FSPOT-X Project, focused on maximizing exergy generated from AM1.5 sunlight, targets an overall system efficiency of >35%. The objective hybrid power system will deliver grid-ready AC power while simultaneously providing thermal energy storage for dispatchable electrical power generation in post sunset conditions. The challenging system-level requirements flow-down critical temperature differential and thermal transport requirements to multiple system components and their interfaces. By integrating and demonstrating multiple technologies, the FSPOT-X hybrid power system seeks to efficiently convert photons to electrons maximizing heat transfer efficiency across system element interfaces. These include: I1) capturing all incident sunlight from the solar concentrator in a receiver cavity to maximize energy generation from the CPV cells, I2) extracting PV thermalization heat from the receiver and into the reflux chamber, I3) moving heat from the reflux chamber through the thermal transfer interface, I4) using the thermal transfer interface to shift heat into the TAPC’s hot heat exchanger, I5) storing excess unused heat in phase change material, and I6) disposal of waste heat at the system level. For each of these thermal interfaces, effective and efficient technical means are being used and applied in order to maximize overall system efficiency for delivery of a next generation cost-effective and market-ready solar power system.

1. INTRODUCTION

Cost-effective methods for transferring absorbed heat converted from incident concentrated AM1.5 sunlight are critical to system level efficiency for the hybrid solar thermal power system being developed under the Full Spectrum Power for Optical/Thermal Exergy (FSPOT-X) Project. Led by Northrop Grumman Aerospace Systems and teamed with the Jet Propulsion Laboratory, the FSPOT-X effort is focused on maximizing exergy for the power system under development. This power system targets an overall efficiency of >35% with high system exergy while simultaneously providing thermal energy storage (TES) for dispatchable electrical power generation in post sunset conditions which significantly enhances the hybrid system power output profile. These challenging system-level requirements flow-down critical temperature differential and thermal transport requirements to multiple system components and their interfaces.

1.1 The FSPOT-X Hybrid Solar Thermal Power System

FSPOT-X’s hybrid solar thermal power system integrates concentrated solar photovoltaic (CPV) cells together with a thermoacoustic power converter (TAPC) [1-3] and thermal storage (TS) to produce electricity from sunlight directed at the power system by a solar concentrator as shown in Figure 2. By combining CPV, TAPC and TS, the FSPOT-X power system efficiently converts photons to electrons through effective heat transfer methods.
The FSPOT-X Project objectives include demonstration of the integrated hybrid solar thermal power system measuring end-to-end efficiency and quantifying exergy performance. This demonstration will focus on the design and test of an integrated TAPC/PV/TS solar power plant module. We are proceeding through testing and collection of information on: a) TAPC efficiency operation, b) high temperature PV cell life performance, c) integrated thermal storage and d) system operating temperatures and environments. We are assessing near-term and long-term markets for solar thermal power generation modules and execute the project’s Technology-to-Market Plan to commercialize the objective system.

The hybrid power system achieves its target high efficiency objectives by converting all PV thermalization losses to useable heat at a useful temperature. This heat is in turn converted to electricity in a bottoming cycle by the TAPC – or optionally could be used for additional dispatchable heat. By integrating the PV, TS and TAPC into a single component illuminated by a concentrating solar reflector, the number of system components and overall system performance is directly driven by maximizing heat transfer across critical interfaces.

1.2 Major System Thermal Interfaces
As highlighted by the enumerated red circles in Figure 1, there are multiple major thermal interfaces in the FSPOT-X demonstration power system architecture. These include: I1) capturing all incident sunlight from the solar concentrator in a receiver cavity to maximize energy generation from the CPV cells, I2) extracting PV thermalization heat from the receiver and into the reflux chamber, I3) moving heat from the reflux chamber through the thermal transfer interface, I4) using the thermal transfer interface to shift heat into the TAPC’s hot heat exchanger, I5) storing excess unused heat in phase change material, and I6) disposal of waste heat at the system level. The critical requirement is to preserve system exergy by transporting high thermal fluxes across low temperature differentials at these key interfaces. For each of these thermal interfaces, cost effective and efficient technical means are to be used and applied in order to maximize overall system efficiency for delivery of a next generation cost-effective and market-ready solar power system.

2. DEMONSTRATION ARCHITECTURE CHALLENGES
For the FSPOT-X Project, a number of challenges flow from the approach taken to demonstrate the hybrid power system. The most critical ones involve:

a. Adapting and Integrating Existing Commercial Hardware: Rather than design and build critical system elements for optimal overall system performance, FSPOT-X is using commercially available hardware for much of the system demonstration driven by project budget and execution schedules. These elements include the pre-existing solar concentrator, CPV cells designed and fabricated for CSP (concentrated solar power) use and a commercially built pressure wave generator. Because of this reliance on available components, the FSPOT-X team must adapt the demonstration system implementation to the design limitations of these existing components.

b. CPV Cells Limit System Operating Temperatures (and System Lifetime): Given that FSPOT-X is using commercial off-the-shelf CPV cells, the thermal limitations of these cells directly constrains overall system operating temperatures and thus the maximum efficiency of the system demonstration. While multi-junction CPV cells are designed for multi-sun concentrated use (generally 100 to 1000 suns), their operating temperatures are
usually up to 110° [4] – not the 300 to 500° C or even higher temperature ranges for most effective heat engine operations. Moreover, CPV cells lose power generation efficiency at higher temperatures by 0.04% per each °C of increase thus directly limiting the amount of electrical power that can be generated via CPV. In addition, with the lack of test data for higher temperature life time operation of commercial CPV cells, assessing and predicting power system lifetime further complicates cost-effective development of an objective commercial power system.

c. Predicting TAPC Performance in Terrestrial Use: Operational design and development of TAPC hardware has been based on its use in space environments, where the dominant heat transfer regimes are through conduction and radiation and where the primary thermal energy source for TAPC operation was to be from heat released by radioisotope decay at temperatures above 800° C. In the context of applying TAPC technology for terrestrial use by FSPOT-X, TAPC power generation at much lower temperatures, impacted by convective heat losses in addition to thermal conduction and radiative effects, and system control under diurnal/seasonal/yearly cycles must now be taken into account.

d. Selecting Optimum CPV/TAPC Interface Operating Temperatures: In the system configuration shown in Figure 1 the CPV system serves as topping cycle over TAPC bottoming cycle in converting the absorbed solar energy. The multi-junction (MJ) CPV operates quite effectively at room temperatures and temperatures below 80 C, but as mentioned above, the MJ CPV sub-system loses efficiency by 0.04% for each 1 °C increase in its operating temperature. The TAPC on the other hand experiences significant efficiency increases as its hot-side temperature increases simply due to Carnot effects. In the hybrid CPV/TAPC system shown in Figure 1, an optimum CPV/TAPC interface temperature exists, at which one would operate the CPV and ultimately dissipate thermal energy to the TAPC, which optimizes either system efficiency or system power. This optimum temperature condition depends on whether one is interested in maximizing system efficiency or maximizing system power output. Since we are interested in maximizing system efficiency, then to a close approximation the optimum operating temperature for the MJ CPV and TAPC hot side would be dictated by the following relationship:

\[
\frac{\partial \eta_s}{\partial T} + \frac{\partial \eta_{TAPC}}{\partial T} = \eta_s \cdot \frac{\partial \eta_{TAPC}}{\partial T} + \eta_{TAPC} \cdot \frac{\partial \eta_s}{\partial T} \quad [2]
\]

These are two fundamentally different relationships, which would yield different values for this optimum interface operating temperature. However, they are both critical in determining the system energy flows at these two optimum conditions and in determining how far from optimum conditions other CPV/TAPC interface temperature selections and designs may be. In our current work, several factors, including CPV cell survival and reliability, have led to selecting our CPV/TAPC interface operating temperature to be 350°C.

3. MANAGING SYSTEM-LEVEL HEAT TRANSFER INTERFACES

It should be noted that the development of the FSPOT-X power system involves two principal efforts – a) design, build and test of a demonstration system integrating existing hardware and b) definition and specification of an objective architecture for cost effective, grid-ready power production. Given the challenges of documenting both efforts simultaneously, this paper primarily focuses on identifying system-level critical heat transfer interfaces for the demonstration activity with an eye toward how to develop and implement these interfaces for the objective architecture.

In this context, we focus on heat transfer across interfaces of primary system elements such as between the solar concentrator and the CPV cells (Interface I1 from Figure 1). Heat transfer characteristics within a specific device (e.g. within the TAPC or internal to the reflux chamber) are modeled as needed (device physics) and measured where appropriate but not directly addressed in this paper. We rather focus our attention on interface heat flux requirements derived from the inherent design constraints of our existing components and the required energy flows emanating from our system performance goals.

4. BASELINED DEMONSTRATION HEAT TRANSFER APPROACHES

As highlighted in Section 1.2, 6 major system-level thermal interfaces have been identified and the baseline heat transfer approach for each interface is specified below.

4.1 I1: Concentrator-Receiver-PV Interface

<table>
<thead>
<tr>
<th>ID</th>
<th>Thermal Interface</th>
<th>Demo Baseline</th>
<th>T_{inf}</th>
<th>ΔT_{inf}</th>
<th>IF Risks</th>
</tr>
</thead>
<tbody>
<tr>
<td>I1</td>
<td>Concentrator-Receiver-PV</td>
<td>reflective / radiative absorb</td>
<td>370-350°C</td>
<td>0°C</td>
<td>optical alignments, solar flux &gt; heat transfer</td>
</tr>
</tbody>
</table>

Figure 3. I1: Concentrator-Receiver-PV Interface

At the I1 Concentrator-Receiver-PV Interface, solar energy input into the demonstration power system occurs. When
concentrated sunlight from the Solar Concentrator is focused on the aperture to the Receiver, sufficient illumination flux is needed to generate power from the CPV cells of the PV Module. The receiver wall design must also transfer cell thermalization losses and bypass IR energy into the walls of the Receiver, which is ultimately transported to the TAPC as well as any excess energy being stored in Thermal Storage.

The primary concern at this heat transfer interface is a) providing sufficient solar flux necessary for electrical power generation and b) holding the PV Module to the maximum temperature limit quantified from elevated CPV temperature testing. Our system goal is to operate the MJ CPV cells at 350-370°C at this interface.

In using an existing solar concentrator, the FSPOT-X team will size the receiver based on power generation capability of the overall hybrid solar thermal power demonstration system, not the full capability of the solar concentrator. This is because the 12-mirror paneled 54 m² Solar Concentrator that provides sufficient area and pointing capability for 1000 suns at its receiver, which provides concentrated reflected total solar illumination greater than electrical generation capability of the demonstration system.

Our system level goal in this demonstration work is to collect and absorb about 26 kWt of solar energy, which is then distributed amongst the CPV cells, the TAPC hot side junction, and the TS. Initial system thermal-power models indicate, after accounting for parasitic thermal losses (approximately 10% as discussed below), the CPV cells will “directly see” about 14 kWt of this energy in its active wavelength range and produce about 7 kWe. The remainder of the solar IR bypass energy and cell thermalization energy will provide about 16.4 kWt to the TAPC, which could provide an additional ~4.1 kWt. These overall high-level system energy flows generally dictate our sub-system energy flows discussed below.

For this reason, total flux from the Solar Concentrator will likely be limited by the maximum temperature limit of the PV module and heat flux requirements at the reflux boiler design discussed below. Design of the existing concentrator allows for aiming each of its 12 mirror panels and/or covering them individually.

**4.2 I2: Receiver-PV-Reflux Chamber Interface**

![Figure 4. I2: Receiver-PV-Reflux Chamber Interface](image)

At the I2: Receiver-PV-Reflux Chamber Interface, PV thermalization energy as well as unused absorbed Receiver collected energy as heat. Thermalization of the solar cell comes from the solar thermal heat as well as the losses due to the absorption of the high-energy photons.

Given that thermalization heat is drawn from the PV Module via conduction, the rate of conductive heat transfer into the Reflux Chamber will be critical to limiting and maintaining the operating temperature of the PV Module. Our preliminary work shows that this PV-Reflux Chamber interface must support a heat transport flux of 20-50 W/cm² and possibly higher as solar concentrations increase to the 1000X range. Our design is allocating 15-16°C temperature differential at the Reflux Chamber-working fluid interface and about 3°C temperature differential between the PV surface and Reflux Chamber wall. In addition, the radiative absorption performance of the Receiver affects the total heat transfer into the Reflux Chamber.

**4.3 I3: Reflux Chamber-Thermal Transfer Interface**

![Figure 5. I3: Reflux Chamber-Thermal Transfer Interface](image)

At the I3 Reflux Chamber-Thermal Transfer Interface, heat is transferred from the Reflux Chamber into the Thermal Transfer Interface. Within the baseline demonstration system design, this interface is a field thermal interface joint to allow for attachment between the Reflux Chamber and the remainder of the hybrid power system. This allows for the Thermal Transfer Interface to be directly assembled together into the hot heat exchanger of the TAPC. For this reason, this thermal interface will provide continuous convective flow across the interface but may offer unintended thermal leak paths if not properly closed and insulated.

**4.4 I4: Thermal Transfer-TAPC Interface**
Figure 6. I4: Thermal Transfer-TAPC Interface

At the I4 Thermal Transfer-TAPC Interface, heat is transferred into the hot heat exchanger (HHex) of the TAPC’s heat engine. On one side of this interface, the working fluid of the Reflux Chamber/Thermal Transfer Interface will flow across the I4 interface and condense to transfer heat into the interface. On the other side of the interface, the working fluid of TAPC will be heated and used to generate power.

Critically important is the working fluid’s heat transfer coefficient of the Reflux Chamber/Thermal Transfer Interface and the surface area of the I4 interface available for condensation of this working fluid. Our preliminary design indicates that we require a heat transfer coefficient at the reflux-side of this interface of about 2700-3100 W/m²-K. Northrop Grumman and JPL are intensely coordinating joint design options at this interface to satisfy system demonstration requirements. A number of working fluids have been considered including Dowtherm A, naphthalene, NaK, Cs and K [5]. Operating pressures at ~350°C of each of these fluids is a key design criteria. Dowtherm A would operate at about 76 psi pressure and naphthalene would operate at about 162 psi pressure at this temperature, while NaK, Cs and K would all operate substantially below atmospheric pressure. Sub-atmospheric pressure operation creates a significant long-term design challenge which our team is currently evaluating [6].

4.5 I5: Reflux Chamber-Thermal Transfer-Thermal Storage Interface

At the I5 Reflux Chamber-Thermal Transfer-Thermal Storage Interface, thermal energy is transferred into and out of Thermal Storage. This thermal energy is then available for the bottoming part of power system operation using the TAPC device during solar outage and/or post-sunset operating conditions.

On the Reflux Chamber/Thermal Transfer Interface side, convection of the working fluid will transfer heat into the interface. Nominally, on the Thermal Storage side of the I5 interface, thermal conduction into the thermal storage material will be the primary heat transfer mechanism. Additional consideration for enhancing heat transfer into the storage material using devices such as heat pipes and storage container designs with innovative, highly conductive fins are being evaluated. Thermal transport into the TS materials will be governed and passively controlled by temperature differential management between the Reflux Chamber and the TS materials themselves.

In the baseline demonstration system design, a number of different storage materials have been assessed for use. Depending on TAPC device efficiency and available mass for the Thermal Storage unit, different salts and salt mixtures are being traded. Figure 8 shows some of our preliminary TS design materials [7], TAPC-efficiency/TS-mass tradeoffs, and expected TS masses to satisfy our initial 15-minute thermal storage requirement for a 10 kW_e system. Initial TAPC efficiency estimates indicated that about 100 kg of TS materials will be required in this first system demonstration.

Figure 7. I5: Reflux Chamber-Thermal Transfer-Thermal Storage Interface

4.6 I6: System Waste Heat Disposal Interface

At the I6 System Waste Heat Disposal Interface, waste heat is transferred into the surrounding terrestrial environment. Maintaining as low a temperature as possible at this interface is critical to overall system level efficiency and performance in order to maximize ΔT and thermodynamic efficiency of the TAPC.

Figure 9. I6: System Waste Heat Disposal Interface
Waste heat from the cold heat exchanger (CHex) of the TAPC’s heat engine is convectively coupled to the system’s external heat exchanger. Within the baseline demonstration system, this external heat exchanger is exposed to the ambient environment for waste heat disposal – which may be 45°C during midday summer operation in desert conditions.

4.7.16: Reflux Chamber Fluid Options

The high-performance heat transfer in the Reflux Chamber is critical to successful system design and preserving exergy at the system thermal interfaces from the PV modules through to the TAPC HHex design. The expected boiling heat transfer at the Receiver Cavity/Reflux Chamber interfaces and the condensation heat transfer at the Reflux Chamber/TAPC HHex interface must be maximized to the extent possible. The Reflux Chamber fluid selection is critical to accomplishing this design. Our team has focused on two potential candidates that can operate at 350°C: naphthalene and Dowtherm A.

These candidates have much different thermophysical properties, melting temperatures, and boiling and condensation heat transfer behavior. Several boiling heat transfer correlations from Rohsenow, Foster-Zuber and Stephan-Abdelsalam [8, 9, 10] were used to estimate boiling heat fluxes as a function of wall superheat, \( \Delta T = T_{w} - T_{sat} \), at the Reflux Chamber surfaces.

Figure 10 shows the projected boiling heat flux as a function of interface superheat that each fluid can provide in the Reflux Chamber design based on equations from these correlation and \( T_{sat} = 350°C \). It is clear that naphthalene has a significant higher boiling heat flux performance than Dowtherm A, and therefore would provide our system design with significant boiling performance margins or significantly smaller interface temperature drops. The most probable boiling design point (\( \Delta T \)-heat flux) based on our current Reflux Chamber system design is shown in Figure 10. This indicates that PV cells in the Solar Receiver cavity shown in Figure 4 would operate at about 376°C.

Condensation heat transfer on the TAPC’s HHex surfaces is just as critical to the system performance preserving exergy and maximizing efficiency. In this design, condensation heat transfer must occur over a bundle of cylindrical hot head tubes to deliver energy to the TAPC system. The condensation heat transfer over individual tubes and tube bundles was estimated using condensation heat transfer correlations from the Rosenhow et al. [8] and Incropera and Dewitt [10] to estimate condensation heat transfer coefficients as a function of wall subcooling, \( \Delta T^* = T_{sat} - T_{wall} \), where \( T_{wall} \) in this case is below \( T_{sat} \). Figure 11 shows the condensation heat transfer estimates as a function of \( \Delta T^* \) on the TAPC HHex for naphthalene and Dowtherm A as the candidate fluids. Naphthalene once again shows clear performance benefits with condensation heat transfer coefficients about 38-39% higher than achievable with Dowtherm A. Naphthalene is therefore a clear choice in providing our system design with significant condensation heat transfer performance margins or significantly smaller interface temperature differentials. The most probable condensation performance range (\( \Delta T^* \)-heat transfer coefficient range) on our TAPC hot-head tubes based on our current Reflux Chamber system design is shown in Figure 11. This indicates that our TAPC hot-head surfaces shown in Figure 6 would operate at about 341-345°C. It is beneficial to our system design and performance that the condensation heat transfer coefficient increases as the \( \Delta T^* \) decreases, which may provide our system with self-correcting/self-control capabilities. Our Reflux Chamber design will also incorporate active design techniques to ensure that these condensation heat transfer coefficients remain high throughout all operational scenarios.

Figure 11. Naphthalene and Dowtherm A Condensation Heat Transfer Coefficient Projections at Hot-Head Tube Surfaces Using [8, 10]

5. NEXT STEPS AND CONCLUSIONS

Maximizing heat transfer across multiple interfaces for overall system operation efficiency is critical to developing and field testing a demonstration hybrid solar thermal power system as well as an ultimate cost-effective commercial grid-ready power module for the FSPOT-X Project.

Our design is incorporating high-performance boiling and condensation heat transfer techniques to preserve system exergy and maximize efficiency. Additional surface
enhancement techniques will be investigated in the future to augment heat transfer at critical interfaces discussed herein. TES materials and design configurations are integrated within our system design in a manner to provide the best thermal transport between TES and Reflux Chamber sub-systems. At this design stage, our FSPOT-X system design is demonstrating a clear pathway to high heat transfer performance at the minimum temperature differentials required to achieve system performance objectives. We are actively investigating and identifying various system operational scenarios and geometries to ensure our thermal transport design is optimal under all conditions.

As the overall effort moves forward, use of thermal models to refine the heat transfer performance of the identified system-level thermal interfaces as well as measuring their as-built performance on the demonstration system will continue.

ACKNOWLEDGEMENTS

For Jet Propulsion Laboratory, this work was carried out under contract #DE-AR0000466 between NASA and the U.S. Department of Energy, Advanced Research Project Agency – Energy, Washington, DC at the Jet Propulsion Laboratory, California Institute of Technology, under a contract to the National Aeronautics and Space Administration.

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