Low Cost Dispatchable Heat for Small Scale Solar Thermal Desalination Systems

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Abstract

The research team at UC Merced is developing a novel low cost non-tracking solar thermal collector/concentrator called the Integrated Compound Parabolic Concentrator (ICPC) and a low-cost phase-change thermal energy storage system. The result is low cost 24/7 solar heat on-demand, and together they will significantly reduce the levelized cost of heat (LCOH) to below \$0.015 per kWh_{th}, while also incorporating features of dispatchability and portability. The final project objective is successful demonstration of an integrated (solar thermal + storage) system which is on track to provide an LCOH < \$0.015/kWh_{th}. The ultra-low cost target (half the price of natural gas) should incentivize marketplace adoption.

Keywords: Solar Thermal Collector, Nonimaging Optics, Solar Desalination, Thermal Storage

1. Introduction

Water handling technologies are vital components of human infrastructure, which are increasingly being stressed due to population growth, competing energy demands, and the disruption of natural hydrologic cycles. Global water demand is expected to rise 20-30% above current levels by 2050[1]. Since most existing freshwater resources have been developed, desalination is increasingly being viewed as a solution which can both recover wastewater generated from existing supplies and add new capacity through the desalination of seawater.

In addition to supplementing municipal water supplies, desalination is also a means to increase water supplies for agriculture and to purify water produced from various industrial processes and from oil and gas exploration. Agriculture has high water demands (69% globally[1], 80% of California's water consumption[2]) with runoff that is typically saline due to salts in the soil and groundwater occurring naturally or from fertilizer use. Increasing soil and groundwater salinity adversely impacts production from irrigated agriculture, and even changes the types of crops that can be grown over time in some areas. Salinity must be managed to avoid progressing into desertification. Industry[1] (including power generation) consumes 19% of the world's water. Many industrial processes generate wastewater, which often cannot be discharged into surface waters. As a result, these industries must pay for disposal and or cleanup costs, which can be especially large for inland industries. A major byproduct of oil and gas production is saline water, also known as produced water, which faces similar disposal issues. Since produced water is often generated away from grid infrastructure, it must be disposed of at high cost by transportation away from the site. In all of these scenarios, a distributed, renewable, and low cost solution is needed.

The most common desalination technologies are Reverse Osmosis (RO), Multi-Stage Flash (MSF), and Multi-Effect Distillation (MED). RO systems use high-pressure pumps to push water through a molecular membrane, leaving behind impurities in a concentrated solution. MSF and MED systems use thermal energy to evaporate water, distilling it and leaving behind the impurities. Since the thermal energy consumption is large, multiple stages are employed to increase the water recovery relative to the heat input. Emerging thermal desalination technologies include forward osmosis (FO), membrane distillation (MD), and humidification-dehumidification (HDH), which may be able to reduce the specific energy consumption (SEC) required to desalinate water compared to the traditional thermal technologies (MSF, MED).

RO systems are the most common technology employed today, due to their low energy costs and overall cost

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[1] WWAP (UNESCO World Water Assessment Programme). 2019. The United Nations World Water Development Report 2019: Leaving No One Behind. Paris, UNESCO.
[2] https://water.ca.gov/Programs/Water-Use-And-Efficiency/Agricultural-Water-Use-Efficiency

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of water production. Thermal desalination, however, is advantageous when dealing with high total dissolved solids (TDS) or variable water supplies. For example, MED systems are particularly attractive because: (1) their low operating temperature (70 °C) minimizes tube corrosion and scale formation, (2) operational and pre-treatment costs are low because the feed water quality is not as essential as for RO, and (3) the heat transfer and water production efficiencies are high and power consumption is low compared to MSF (Ullah et al. 2019). Thermal desalination systems tend to be more robust and better suited for dealing with high total dissolved solids (TDS) streams (such as RO brine), wastewater, and other variable water sources.

The latest concentrating solar power (CSP) systems analysis by the U.S. National Renewable Energy Laboratory (NREL) (Turchi et al. 2019) lists sixteen companies engaged in solar industrial process heat (IPH), three of which have conducted projects related to solar desalination or wastewater treatment. While technically feasible, solar desalination plants have not been realized on a significant scale due to poor economics, which are limited by low efficiencies and high capital costs (Chandrashekara et al. 2017). Solar thermal technologies were developed either for domestic hot water production (< 70°C) or high temperature (> 400°C) concentrating solar power (CSP) applications. Thermal desalination, however, is a low temperature process and can be powered by low quality steam (100-120°C). As a result, hot water collectors have trouble efficiently reaching the temperatures needed for thermal desalination and the CSP collectors are over-engineered and unnecessarily costly for the application. To reduce capital costs, solar technologies which are tailored specifically for the purpose of desalination are needed (Reif et al. 2015).

In this paper, we detail the design of a novel low cost solar thermal energy system that will significantly reduce the levelized cost of heat (LCOH) to below \$0.0015 / kWh_{th}, while incorporating features of dispatchability and portability. It system will reach its LCOH target through a combination of low-cost collector materials (primarily glass and aluminum), ease of installation and operation, increased thermal efficiency, and the use of nonimaging optics—which will enable it to collect all solar irradiance (global), not just direct normal irradiance (DNI). The project includes the design and development of a new collector/concentrator called the Integrated Compound Parabolic Concentrator (ICPC) and, in parallel, the design and development of an accompanying thermal energy storage (TES) system.

2. The Internal Compound Parabolic Concentrator (ICPC)

The integrated compound concentrator (ICPC) was first described by Snail et al. in 1984, and has since undergone several years of development. It is a highly efficient, simple, stationary vacuum tube collector with integrated optics. Previous iterations were expensive due to production and manufacturing constraints of the time, but developments to metal-glass sealing technology and aluminum heat pipes today enable ultra-low material costs for ICPC collectors.

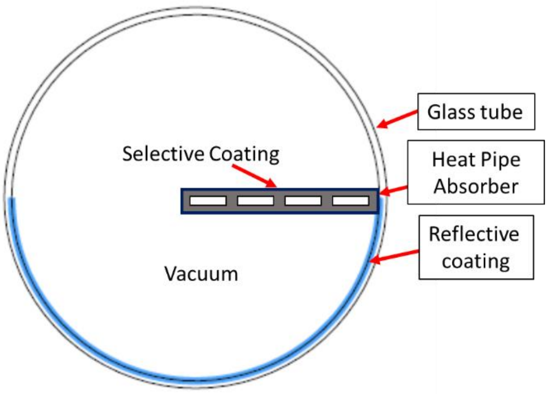

Fig. 1: ICPC solar thermal collector cross-section

The internal compound parabolic collector (ICPC, Figure 1) consists of an (i) evacuated glass tube, (ii) heat pipe absorber, (iii) reflective coating, and (iv) metal-glass seal. The glass tube encases the absorber in a vacuum-insulation jacket, improving thermal efficiency at elevated temperatures by eliminating convective heat loss. A reflective coating applied to the bottom half of the glass tube provides optical access to the bottom half of the absorber. In this way, the ICPC has a geometric concentration ratio of approximately 1 and incoming solar irradiance can be reflected to the bottom surface. The absorber is an aluminum heat pipe, which is selectively coated to have a high solar absorptivity and low thermal emissivity at a target operating surface temperature of 150 °C. This temperature was selected to allow 30 °C for heat transfer between the absorber surface and the final working fluid of 120 °C. The heat pipe absorber passes through an aluminum end cap to expose the condensing section outside of the glass tube. This transition is called the metal-glass-seal, which seals the glass tube to the aluminum end cap and heat pipe and maintains the vacuum inside the tube.

Performance of the ICPC is estimated using ray tracing software and a simple thermal model. The optical model was developed with the specifications in Figure 2. The heat pipe is a rectangular shape positioned horizontally against the glass tube in a 9 o'clock position. At normal incidence, the ICPC has an optical efficiency ($\eta_{optical}$) of 75.6%.

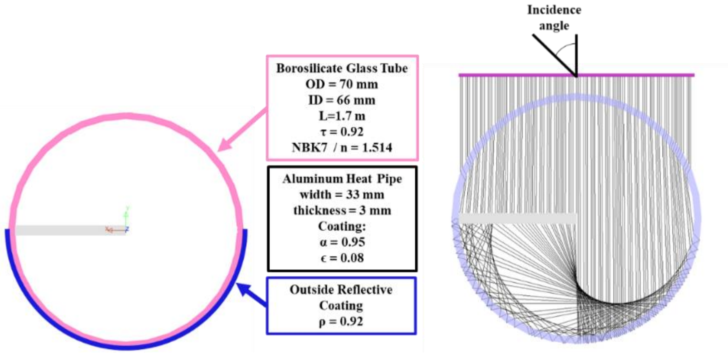

Fig. 2: Ray tracing simulation design and parameters

The incidence angle modifier (IAM) was generated by sweeping the angle of incoming light from -90˚ to 90˚, to understand optical performance at off-axis incidence angles. The IAM in Figure 3 is scaled to the optical efficiency at normal incidence, so it has a value of 1 at zero degree incidence. This is also the angle of the lowest optical performance (due to a larger average number of reflections) and the collector should perform up to 10% better at off-axis angles.

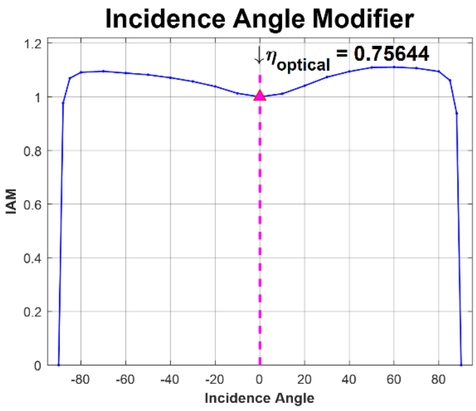

Fig. 3: Incident Angle Modifier (IAM)

A simple thermal model was developed to estimate the thermal efficiency ($\eta_{thermal}$) of the collector at various operating temperatures. Thermal efficiency is calculated according to Equation 1, where $\eta_{optical}$ is the optical efficiency determined from the ray tracing simulation results, emissivity of the selective coating on the absorber is an estimated $\epsilon = 0.08$, ambient temperature of the surrounding environment is $T_{amb} = 25$˚C, and global solar irradiance G = 1000 W/m^2.

$$\eta_{thermal} = \eta_{optical} - radiative\ losses = \rho\tau\alpha - \frac{\epsilon\sigma(T_{abs}^4 - T_{amb}^4)}{\frac{A_{aperture}}{A_{absorber}} * G} \qquad \text{(eq. 1)}$$

No convective losses are considered because the hot absorber operates in an evacuated environment, and therefore optical and radiative losses are the major contributors which determine performance. The thermal efficiency is plotted in Figure 4 for multiple operating temperatures, starting from 40˚C to a stagnation temperature of 375˚C. At an absorber operating temperature of 150˚C, the collector will have a solar-to-thermal efficiency of about 65%.

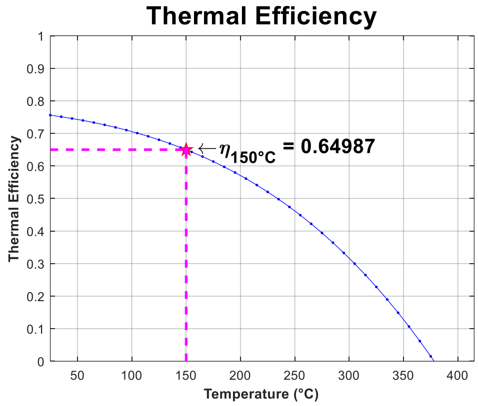

Fig. 4: Thermal efficiency simulation results

Prototype ICPCs are currently being manufactured. Miniature prototypes are shown in Figure 5 and full-scale two-meter long tubes are also being procured for testing. The purple coloring of the absorbers is the selective coating, which has a high solar absortpance and low thermal emittance at operating temperatures to reduce radiative losses. The gas getter, shown as the silver coloring at the bottom of the collectors, provides a chemical buffer to maintain vacuum and also provides is a visual indicator of vacuum integrity. If the gas getter changes from silver to white, the vacuum has been lost.

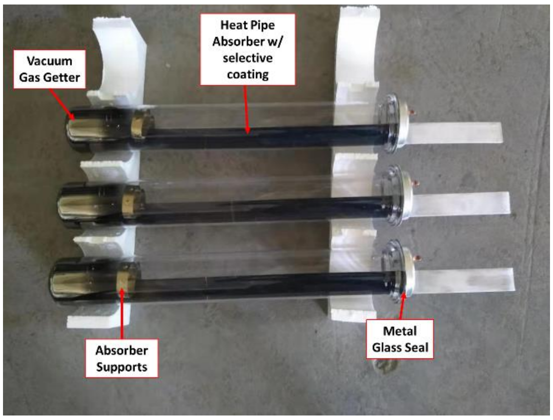

Fig. 5: Miniature Prototype ICPC's

Once the prototypes are completed, a small 20 tube module will be tested on a two axis tracker, on-sun, to quantify thermal performance and confirm the IAM through experimental results.

3. Thermal Energy Storage

A fundamental aspect of solar energy is its intermittency, and solar thermal systems are often greatly enhanced by the availability of thermal storage to allow continuous, reliable heat output. Here we consider the design of a thermal energy storage (TES) system tailored for coupling to the ICPC collector in desalination applications.

Thermal energy can be stored directly as sensible heat corresponding to a change in temperature of the storage media or latent heat corresponding to a change of phase in the storage media as well as by heat of a chemical reaction in the media (Zhang et a. 2016). TES is a well-established technology, but the problem of designing a TES system for solar thermal applications requires simultaneous optimization of cost and thermal performance. These characteristics are tightly intertwined and present several trade-offs. Capital cost can be roughly broken down into 3 categories for TES (considering fluid handling to be external to the TES system):

thermal storage material; heat exchanger material and fabrication; and containment material and fabrication. The thermal performance, meanwhile, can be characterized by the thermal efficiency (heat recovered/heat added) of the charge/discharge cycle and by the temperature difference between the heat transfer fluid entering during charge and exiting during discharge ($\Delta T_{c/dc}$). The former effect is due to heat loss to the environment, while the latter effect represents the generation of entropy and the loss of exergy or quality of the heat stored due to thermodynamic and transport limitations. The interactions of all these elements are complex. E.g., the quantity of thermal storage media can be minimized by increasing specific heat capacity or increasing the temperature range of operation, but the latter decreases thermal performance by increasing the mean $\Delta T_{c/dc}$. Likewise, $\Delta T_{c/dc}$ can be reduced with increased heat exchanger area but at the cost of increased material and fabrication costs.

The use of phase change storage media in direct (or close) contact with the heat transfer fluid offers significant benefit for all these elements. Latent heat storage yields an extremely large effective heat capacity near the phase change temperature, minimizing required storage media quantity and the thermodynamic component of $\Delta T_{c/dc}$. Direct contact heat exchange improves heat transfer performance, reducing the transport component of $\Delta T_{c/dc}$ while eliminating the costs associated with the heat exchanger. Reduced storage media volume and lower mean operating temperatures minimize the cost of the containment structure and the insulation required to minimize heat loss. The advantages of direct contact heat exchange have been extensively applied for both sensible and latent heat storage systems (Kenisaren & Mahkamov, 2007; Wu et al. 2014; Cascetta et al. 2015).

Figure 6 shows a schematic of a direct contact TES. The system consists simply of an insulated enclosure containing a bed of storage media through which the heat transfer fluid flows. Heat transfer resistance between a storage media element and the heat transfer fluid can be decomposed into conduction and convection components. The conduction component corresponds to thermal diffusion between the moving phase interface and the outer surface, as given for a spherical media element in the figure (where k_s is the thermal conductivity of the solid storage media). This resistance is minimized for small storage media elements. The local heat transfer coefficient, h, determines the convection component, along with the element surface area, A_s. For a packed bed of storage media, the local heat transfer coefficient can be determined from a simple correlation for local Nusselt number, which is proportional to the product of Reynolds and Prandtl numbers to the 1/3 power (Bird et al. 2007).

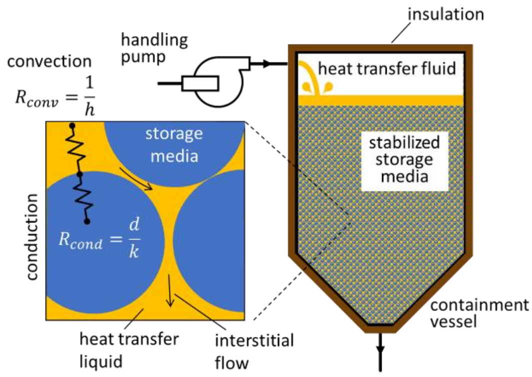

Fig. 6: Schematic of TES utilizing phase change storage media in direct contact with the heat transfer fluid

The storage utilizes high density polyethylene (HDPE) as the storage media. HDPE is compatible with direct contact of the HTF in both solid and molten phases while retaining excellent latent heat capacity. Figure 7 shows a differential scanning calorimetry thermogram for HDPE which has been melted and solidified in propylene glycol. After cycling in HTF, the HPDE storage media retains latent heat of fusion well above 100 J/g.

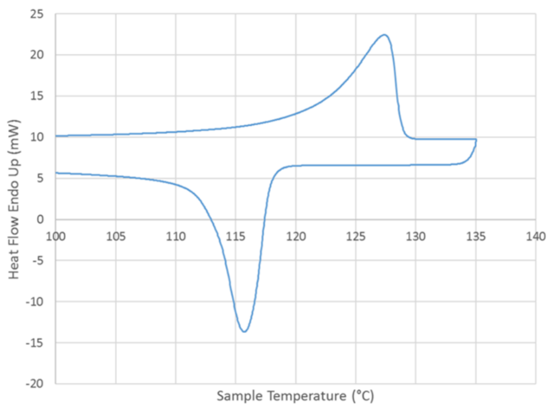

Fig. 7: Differential scanning calorimetry thermogram for HDPE following melting & solidification in propylene glycol (Temperature scan rate is 1˚C/min)

Direct contact between the heat transfer fluid and the storage media leads to an extremely simple and robust TES system, and HDPE offers a low cost/high latent heat storage media compatible with the HTF used in this configuration. The combination of these elements provides significant advantages for application in medium temperature solar thermal applications such as desalination.

4. System Design

Once initial prototyping and performance testing is completed, a pilot solar array and TES system will be built for integrated testing. A 24 kW array (Figure 8), is sized to provide heat to run a desalination system (5kW) and charge the thermal storage (15 kW) during the day; the extra 4 kW account for heat loss to the environment. The solar field will be made of a total 4 banks with 5 modules each, resulting in a total of 400 ICPC collectors.

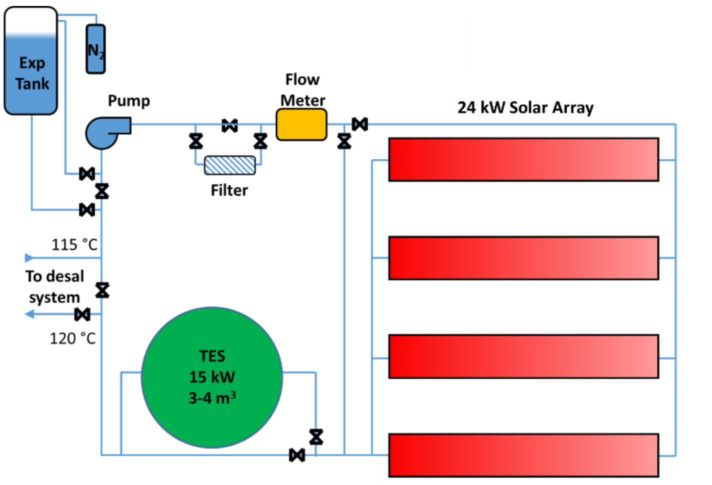

Fig. 8: 24 kW Array + TES + Desalination Schematic

A cost analysis was performed using a bottom-up approach for the main components: Solar Field (ICPC collector module including tubes, manifold box, frame), and the Thermal Energy Storage (TES: phase change material, containment vessel, insulation). The levelized cost of heat for the ICPC and TES is calculated using equation 2. Installed cost includes both direct (sum of the solar field, TES, and balance of system costs) and indirect costs for the solar field, shipping, and installation. The FCR is a factor that includes assumptions regarding financing, tax, and inflation; for this cost breakdown, the FCR is assumed to be a value of 0.083 for 25 years.

$$LCOH = \frac{(Installed\ Cost)*(FCR)+(Annual\ O\&M)}{Annual\ Thermal\ Generation} \qquad \text{(eq. 2)}$$

The annual thermal generation includes the direct normal irradiance (DNI) of the selected geography along with the optical and thermal losses of the system (array, plumbing, storage, & desalination unit). The annual thermal generation is determined using equation 3.

$$Annual\ Thermal\ Generation = (GTI) * (\eta_{collector}) * (\eta_{system}) * 365 \,{}^{days}/_{year} * A_{array\ apert.} \quad \text{(eq. 3)}$$

Here GTI is the global tilted irradiance (annual energy on the plane of a collector array oriented at latitude) of the testing facility in Merced, CA, USA, $\eta_{collector}$ is the thermal efficiency at fluid temperature at the desalination unit of 120˚C, η_{system} is the thermal efficiency of the full system, and $A_{array\ apert.}$ is the required aperture area of the array. The following Table 1 provides the design inputs and economic breakdown, with the resulting LCOH.

Design Inputs		**Economics**	
GHI	5.3 kWh/m^2/day	Total Direct Cost	\$136.49/m^2
Collector Efficiency	64%	Total Indirect Cost	\$6.82/m^2
System Thermal Efficiency	83%	Annual O&M	\$2.73/m^2
Required Array Aperture	53.28 m^2	**LCOH**	**\$0.0142/kWh$_{th}$**
Annual Thermal Generation	54750 kWh/year		

Tab. 1: System design and LCOH breakdown

With the proposed low costing solar collector and accompanying TES, the target goal of a LCOH below \$0.015 / kWh$_{th}$ is met.

5. Conclusion

The ICPC collector is a stationary and low-cost vacuum tube collector. It has a reflective coating applied to the bottom half of the glass tube allowing optical access to the absorber. The combination of these two results in high efficiencies up to 150 °C. The thermal energy storage (TES) system is an insulated tank containing a stabilized phase change material (PCM) storage media. The heat transfer fluid flows in direct contact with the storage media, which allows for charging during the day and discharge at night. The result is low cost 24/7 solar heat on-demand. The combination of low-cost ICPC solar collector with the TES will drastically reduce the levelized cost of heat to below \$0.015 / kWh$_{th}$, which should incentivize deployment at half the price of natural gas. Furthermore, the system's portability will make it an ideal solution for remote areas. Once demonstrated, this technology will be ideal for powering commercial evaporators, multiple effect distillation (MED), multi-stage flash (MSF), forward osmosis (FO), membrane distillation (MD), and humidification de-humidification (HDH) technologies, and enable the renewable management of municipal, agricultural, industrial, produced waters, RO brine, and more.

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