



Assessing the Impact of Heat Rejection Technology on CSP Plant Revenue

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ASSESSING THE IMPACT OF HEAT REJECTION TECHNOLOGY ON CSP PLANT REVENUE

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1 Introduction and background

Utility-scale concentrating solar power (CSP) plants operate by absorbing thermal energy from concentrated solar irradiation and converting the high-temperature thermal energy into electrical power via a Rankine steam cycle. The Rankine thermodynamic cycle requires the flow of energy from a hot thermal reservoir (the solar field) to a cold thermal reservoir to generate mechanical power. The total cycle efficiency is subject to the temperatures of the hot and cold reservoirs; a higher-temperature hot reservoir or a lower-temperature cold reservoir both serve to improve cycle efficiency and maximize power output, but the flow of some of the heat into the cold reservoir (“heat rejection”) is a prerequisite to cycle operation.¹

The heat rejection system can take one of several forms. Traditionally, wet cooling has been used since it provides a low-temperature heat rejection reservoir with the wet-bulb temperature. However, this mechanism consumes a large amount of water via evaporation, so wet cooling is untenable in locations where the water supply is limited. The practical alternative to wet cooling is air cooling. This configuration is subject to the much warmer ambient dry-bulb temperature and a large temperature rise in the air stream due to the low specific heat capacity of air. Consequently, the negative impact on plant performance is accentuated during the hot summer afternoon hours when both peak electricity demand and opportunity for plant revenue are highest. This is shown in Table 1 for Daggett, CA, using time-of-delivery (TOD) rate factors for Southern California Edison [1].

TOD	Rate Factor	\overline{T}_{db}	\overline{T}_{wb}	$\overline{\Delta T}$
1	3.13	35.7	18.1	17.7
2	1.35	30.5	16.6	13.9
3	0.75	26.6	15.2	11.4
4	1.00	18.2	8.7	9.5
5	0.83	14.8	7.1	7.6
6	0.61	10.3	5.2	5.1

Table 1: The time-of-delivery payback rate factor for a utility is generally inverse to the average ambient temperature (°C) in Daggett, CA, for the corresponding period. Rate factor values indicate the multiplying factor applied to the market purchase price for the TOD period.

An alternative heat rejection methodology has been proposed where a wet condenser is placed in parallel with an air-cooled condenser to share the heat rejection load and improve cycle performance (see Figure 1). Previous work by [2] and [3] discuss the performance, cost and water-use impacts of various heat rejection approaches. However, no detailed analysis has been done to evaluate the impact of wet, dry, and hybrid cooling on plant revenue and profitability. This paper explores the impact of cooling technology on revenue for hybrid-cooled plants with varying wet cooling penetration for four representative locations in the American Southwest. The impact of ACC design-point initial temperature difference (ITD - the

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difference between the condensing steam temperature and ambient dry-bulb) is also included in the analysis.

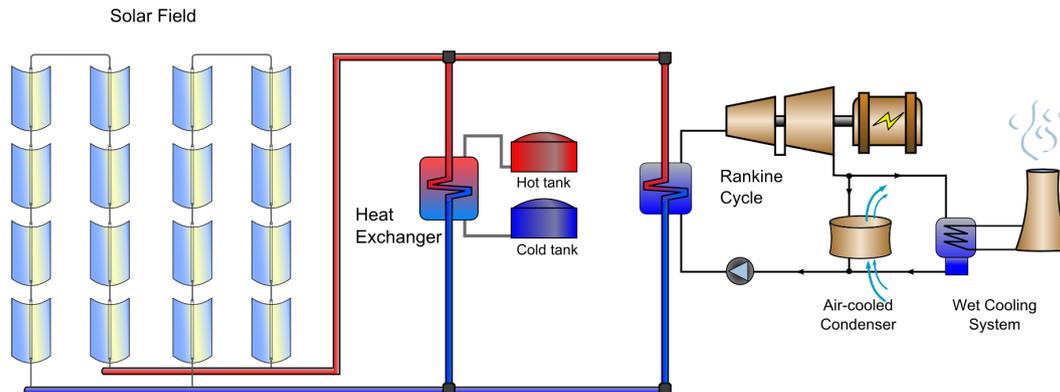


Figure 1: Schematic of the modeled trough plant with hybrid cooling. Note the air-cooled and wet-cooled condensers in parallel in the power block.

2 Performance modeling methodology

System performance for this analysis was modeled using the Solar Advisor Model (SAM) [4] physical parabolic trough model. SAM is a modeling tool with hourly performance models and financial calculation algorithms for CSP, PV, and other renewable technologies. SAM is maintained and developed by NREL, Sandia, and the U.S. Department of Energy, and internal modifications are handled in part by the authors. Though not yet released publicly, the authors have implemented a hybrid wet/dry cooling model in SAM for use in this analysis.

2.1 Plant control

The plant operation strategy in the SAM trough model allows the user to control thermal storage dispatch, the power block gross power output set-point, and the hybrid wet cooling load for each unique TOD period. The storage dispatch fractions for this analysis are all set to zero so that storage dispatch isn't curtailed, and the turbine operating point is set to 100% rated load except for TOD #1, where 105% of design operation is allowed. The payment allocation factors use the Southern California Edison Company dispatch schedule presented in Table 1 above. Figure 2 shows the weekday time-of-delivery period for each hour of the day (horizontal axis) and month of the year (vertical axis). The weekend schedule does not include any peaking TOD rate factors.

Weekday Schedule

	12am	1am	2am	3am	4am	5am	6am	7am	8am	9am	10am	11am	12pm	1pm	2pm	3pm	4pm	5pm	6pm	7pm	8pm	9pm	10pm	11pm
Jan	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Feb	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Mar	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Apr	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
May	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Jun	3	3	3	3	3	3	3	3	2	2	2	2	1	1	1	1	1	1	2	2	2	2	2	3
Jul	3	3	3	3	3	3	3	3	2	2	2	2	1	1	1	1	1	1	2	2	2	2	2	3
Aug	3	3	3	3	3	3	3	3	2	2	2	2	1	1	1	1	1	1	2	2	2	2	2	3
Sep	3	3	3	3	3	3	3	3	2	2	2	2	1	1	1	1	1	1	2	2	2	2	2	3
Oct	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Nov	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5
Dec	6	6	6	6	6	6	5	5	4	4	4	4	4	4	4	4	4	4	4	4	4	5	5	5

Figure 2: Time-of-delivery schedule used in this analysis. Each dispatch period corresponds to a potentially unique dispatch scheme. Hybrid cooling is applied in this analysis during periods 1 & 2.

The hybrid cooling model, like the air and wet cooling models, employs a straightforward energy balance

approach in calculating system performance. The details of the model are not of direct interest to this paper; however, the control logic and equipment sizing strategy are highly relevant. The goal of the parallel wet/dry system is to use wet cooling only as a supplement to the ACC; consequently the ACC is nominally sized to handle the full cooling load throughout the year, though specific design parameters are varied. The wet cooling system is sized (along with the ACC) to meet the maximum heat rejection load that it will be required to meet during the year. The effectiveness of the heat rejection with wet and dry cooling systems depends on the cooling water and air mass flow rates, respectively. Mass flow rates are determined to meet the desired condenser temperature at design, where condenser temperature is found as shown in Eq.'s [1 - 2] for the dry and wet system, respectively.

$$T_{cond,ACC} = T_{db} + \Delta T_{air} + \Delta T_{hot} \quad (1)$$

$$T_{cond,wc} = T_{wb} + \Delta T_{approach} + \Delta T_{cw} + \Delta T_{hot} \quad (2)$$

These equations assume a fixed condenser hot side temperature difference ΔT_{hot} of $3^\circ C$ and make use of the dry-bulb (T_{db}) and wet-bulb (T_{wb}) temperatures. The ACC ITD is equal to the sum of the ΔT_{hot} and ΔT_{air} values. The cooling stream temperature rise determines the required mass flow rate at design, and this mass flow rate is used during the simulation to predict the realized temperature rise.

$$\dot{m}_{air,des} = \frac{\dot{q}_{rej,des}}{c_{p,air} \Delta T_{air,des}} \quad (3)$$

$$\dot{m}_{cw,des} = \frac{\dot{q}_{rej,des} f_{wc,max}}{c_{p,cw} \Delta T_{cw,des}} \quad (4)$$

The maximum wet cooling load is expressed as the total heat rejection load at design times the maximum fraction of the load that is rejected by the wet system throughout the year ($f_{wc,max}$). During off-design operation, the mass flow rate and temperature rise switch places in Eq.'s [3 - 4] such that the temperature rise depends on the heat rejection load and the cooling stream mass flow rate. Condenser pressure drives power cycle efficiency, and this pressure is determined for a hybrid system by using the maximum calculated condenser temperature of either the wet or dry side, according to Eq.'s [1 - 2].

SAM enforces a lower limit on the condenser pressure to maintain practical steam velocities throughout the cooling equipment piping and accommodate pressure limitations in the low-pressure turbine blades. During those time steps where the condenser pressure falls below the minimum value, the mass flow rate of the cooling stream is incrementally reduced (and the condensing temperature increased) until the condenser pressure returns to an acceptable level. The number of increments in the turndown process is determined by a user setting in SAM, and the hybrid system turns down both the wet and dry systems simultaneously in under-limit situations.

In addition to power cycle performance, water use is of primary interest. Water consumption in CSP plants stems from several independent uses, and this study accounts for all of the major water use mechanisms. These include evaporative loss, drift loss, and blowdown from the wet cooling system, steam cycle blowdown, and mirror washing. The water use calculations assume that all blowdown flows are discharged to an evaporation pond and aren't returned to the originating source water supply.

2.2 Experimental design

The goal of this analysis is to provide insight on plant revenue for various cooling system configurations, plant locations, plant configurations, and water-use scenarios. Thus, a relatively wide range of design configurations are considered. Table 2 summarizes the parameterized values in this analysis.

Other experimental design considerations include:

- The modeled plant uses 6 hours of full-load thermal storage, a design solar field thermal output to power block input ratio (solar multiple) of 2.0, and a solar field area of $910,500 m^2$. The power block gross output rating is 110 MWe at 35.48% thermodynamic cycle efficiency.

Parameter	Lower	Upper	Units	Levels
Hybrid cooling fraction (TOD 1 & 2)	0	95	%	20
ACC initial temp. difference	14	30	°C	9
Ambient design temperature	43	27		
Location (TM2)	Phoenix, AZ Daggett, CA Alamosa, CO Las Vegas, NV			4

Table 2: Summary of variable parameters for this analysis. The variables are linked to vary simultaneously, while other variables are varied independently.

- The hybrid fraction is applied to TOD periods 1 & 2 where the market purchase price rate factor is greater than 1.0.
- In the case of the varying ITD, the reference power cycle efficiency is held constant. The ambient temperature at which the rated cycle efficiency is achieved is simultaneously adjusted to reflect the changing effectiveness of the cooling system.
- Dry cooling reference cases correspond to hybrid cooling fractions of 0.0, and fixed wet cooling equipment costs are set to 0 for this case. Wet cooling reference cases are run separately from this parametric table.

3 Cost model

3.1 Equipment cost estimation

In order to capture the impact of cooling system cost on the total installed cost of the plant, a cost model was developed to estimate the additional expense incurred for a hybrid plant. The air cooled and wet cooled systems were sized independently according to the total heat rejection load and the maximum fraction of that load required for the wet cooling system. All non-power-block plant cost values are equal to the SAM defaults for the parabolic trough technology. The total power block cost value is determined by adding the calculated configuration-dependent cooling system cost to the fixed baseline cost that accounts for all of the power cycle equipment except for the cooling system. Table 3 provides a summary of the major cost items and selected financial assumptions used in this analysis.

Item description	Value	Units
Site, Solar Field & HTF system	420	$\$/m^2 \text{ aperture}$
Storage	70	$\$/kWh$
Power plant (non-cooling)	860	$\$/kWe$
Indirect costs, contingency, & tax	34.8	%
O&M by capacity	80	$\$/kW-yr$
O&M by generation	3	$\$/MWh$
Analysis period/Loan term	30/20	years
Federal investment tax credit	30	%
Internal rate of return (IRR)	15	%

Table 3: Financial and cost assumptions used for all system configurations.

The goal of the ACC cost equation is to express the condenser cost in terms of total fin surface area. This approach accounts for scaling in total size due to heat rejection load and size increase due to the design-point ITD. For the ACC system, conductance (UA) per thermal load rejected is a function of the ITD. The IPSEpro process modeling software [5] was used to develop a correlation for UA per rejected load for an analogous air-cooled system, as shown in Eq.[5].

$$C_{UA,acc} = 1310.48 \cdot \Delta T_{ITD,des}^{-0.793} \quad (5)$$

Total heat exchanger conductance is equal to the coefficient in Eq.[5] times the heat rejection load. Since UA provides a measure of the product of the heat transfer coefficient U and total heat transfer area A , heat exchanger area is calculated by dividing UA by the design heat transfer coefficient U_{acc} . This value was selected based on manufacturer specifications for a representative system [6]. Expressed in terms of design gross power output and cycle efficiency, the ACC fin area is:

$$A_{acc} = \dot{W}_{gross} \left(\frac{1}{\eta_{cycle}} - 1 \right) \frac{C_{UA,acc}}{U_{acc}} \quad (6)$$

The wet cooling system equipment costs are broken down into a wet surface condenser cost and a cooling tower cost. The surface condenser cost is based on a UA per rejection load value determined using IPSEpro, and the total condenser surface area is calculated in the same method previously discussed for the ACC. Condenser cost is calculated using a cost correlation obtained from vendor data [7]. The condenser surface area A_{wc} is used to determine the material cost coefficient, where $c_{mat} = 1.222 \cdot A_{wc}^{0.13}$. The pressure coefficient c_{psi} is equal to 1.05 for a full vacuum system.

$$Cost_{wc} = 1909 \cdot A_{wc}^{.497} c_{mat} c_{psi} \quad (7)$$

The cooling tower cost estimate assumes a forced-draft configuration, and makes use of a cost scaling equation provided by vendor data [7] for stainless steel surface condensers.

$$Cost_{ct} = 313259 \cdot (\dot{q}_{rej,des} f_{wc,max})^{0.6} \quad (8)$$

The coefficients and constants used in Eq.'s[5-8] are summarized in Table 4. All costs are brought forward to 2010 dollars using the Chemical Engineering *heat exchangers and tanks* cost index [8].

Item Description	Variable	Value	Units
ACC heat transfer coef.	U_{acc}	38.2	$W/m^2 \cdot K$
ACC cost per fin area		36.5	$\$/m^2$
Wet condenser heat transfer coef.	U_{wc}	2721	$W/m^2 \cdot K$
Wet condenser UA per rejection load	$C_{UA,wc}$	150	$kW/K \cdot MWt$

Table 4: Cost model coefficients and constants.

3.2 Bid price

The parties involved in the development and operation of utility-scale CSP projects value electricity production in different ways. Electric utilities are primarily concerned with maintaining a steady power supply (even during peaking loads) and with minimizing electricity cost to the consumer. In many cases, utilities dealing with heavy peak loads have been forced to extend base-load capacity with dispatchable but expensive power sources. Consequently, utilities are often willing to pay significantly more to purchase electricity from producers during peak load periods, as illustrated by the rate factors in Table 1.

Plant performance is commonly measured in terms of levelized cost of electricity (LCOE), annual output, or conversion efficiency, but a CSP plant owner isn't necessarily constrained by these metrics if production is disproportionately targeted for specific high-revenue periods. From the owner's perspective, the paramount metric over a plant's life is the internal rate of return (IRR) on the capital investment. The desired IRR, plant cost, and projected revenue are all rolled into the *bid price* that's provided to the utility at the beginning of the project. Thus, the bid price serves as an excellent combined metric of plant performance with respect to cost, revenue, profitability, and utility rate factors. The revenue for year n is calculated by summing the product of bid price (p_{bid}), total electricity production ($E_{tot,i}$), and rate factor (f_{mpr}) for each individual TOD period i over the year.

$$R_n = \sum_{i=1}^{\#TOD} p_{bid} E_{tot,i} f_{mpr} \quad (9)$$

In mathematical terms, the bid price is iteratively calculated such that the annual revenue for year n (R_n) satisfies Eq.[10] for net present value (NPV). After-tax cash flow for year n ($C_{AfterTax,n}$) is equal to $Tax Savings + Incentives - Operating Costs - Debt Total Payment + Revenues$.

$$NPV = 0 = \sum_{n=1}^N \frac{R_n - C_{AfterTax,n}}{(1 + IRR)^n} + C_{AfterTax0} \quad (10)$$

4 Results

The goal of parallel wet/dry cooling is to boost power production (and revenue) for ACC systems during the most profitable peaking TOD periods, thereby benefiting utilities in satisfying peak demand and plant owners in increasing production revenue. By measuring performance with the bid price, the analysis in this paper shows that traditional metrics can fail to produce plant configurations that are optimized for profitability. For example, Figure 3 shows both the LCOE and bid price plotted for a range of ACC ITD's and wet cooling fractions for a plant in Las Vegas, NV.

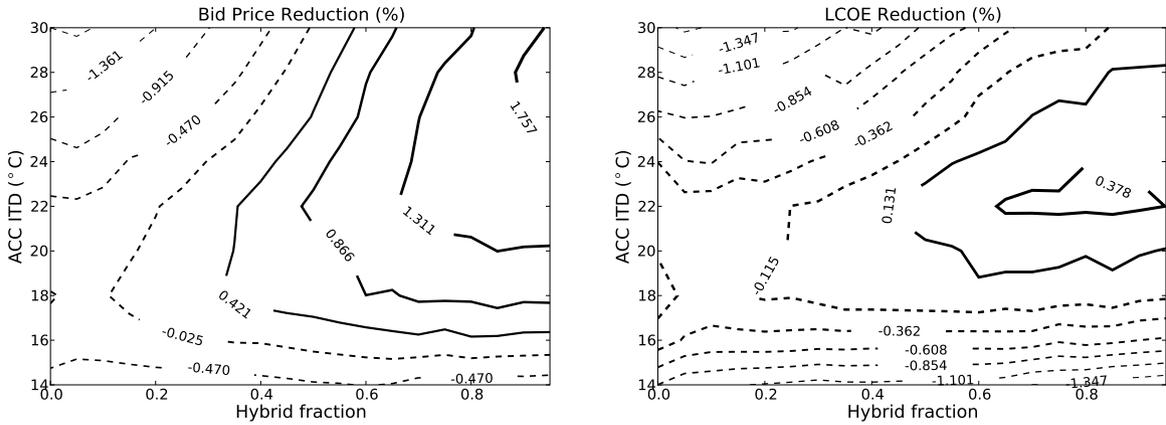


Figure 3: Contour plots for a plant in Las Vegas, NV, showing the bid price and LCOE reduction for hybrid cooling compared to an optimized air cooled plant (positive values correspond to a reduction in bid price or LCOE). The financial metrics are plotted for a range of design-point ACC ITD's and for wet cooling loads from 0 to 95%.

Notably, the increase in wet hybridization has a markedly greater impact on bid price than LCOE, and the maximum LCOE reduction occurs at a much lower ACC ITD than the maximum bid price reduction. The plot of minimum bid price for each hybrid cooling fraction in Figure 4 shows that hybrid cooling shows promise even in cooler climates like Alamosa, CO, for heavily weighted TOD schedules.

Results from the full analysis are summarized in Table 5. Dry and hybrid cases are compared to the baseline wet cooling system for each plant location, and optimal plant configurations for the LCOE and bid price metrics are presented separately.

5 Conclusions

The results of this analysis show that parallel wet/dry cooling offers an opportunity for significant reduction in water use compared to wet cooling while providing a noticeable improvement in a bid price. Several trends in the results are of interest and have applications in the design of heat rejection systems:

- Bid price minimization offers advantages over LCOE- or production-based approaches, and captures the potential viability of hybrid cooling for some markets.
- The bid-price optimized ACC ITD for an exclusively dry cooled plant is generally lower than for a hybrid cooling plant. The difference is most pronounced in hot climates.

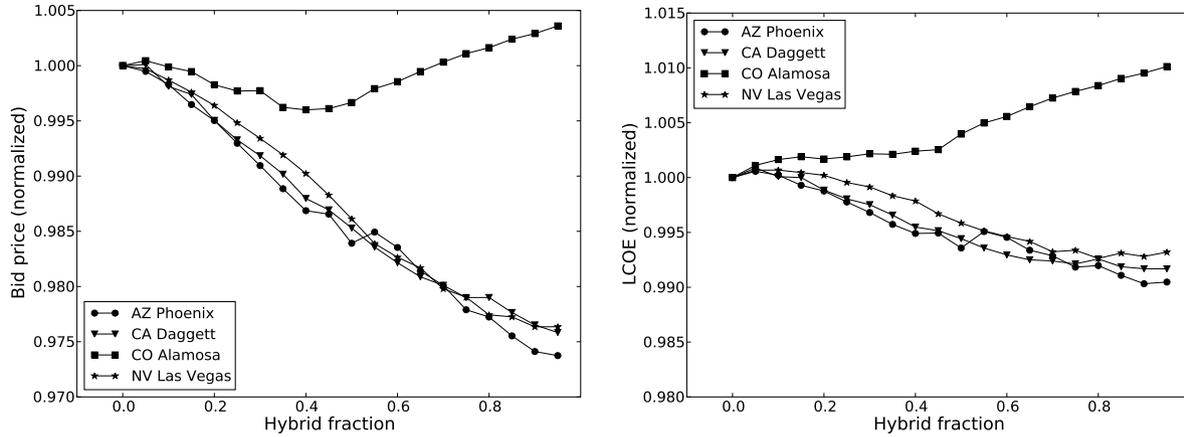


Figure 4: Minimum bid price and LCOE for all modeled ITD's over a range of wet cooling fractions.

	<i>Units</i>	Phoenix	Daggett	Alamosa	Las Vegas
Wet cooling reference case					
LCOE (real)	\$/MWh	169.9	159.8	192.0	171.7
Bid price	\$/MWh	152.8	143.1	169.0	152.0
Annual output	GWh	338.71	360.32	299.08	334.91
Annual water use	m^3	1,363,000	1,409,000	1,129,000	1,312,000

Dry cooling case		Minimize LCOE			
Optimal ITD	$^{\circ}C$	18	20	28	18
LCOE penalty	%	7.83	7.40	5.55	8.02
Annual output penalty	%	3.76	3.70	2.84	3.93
Annual water use reduction	%	94.5	94.6	93.6	94.3

Hybrid cases		Minimize LCOE			
Hybrid ACC ITD	$^{\circ}C$	22	22	28	22
Wet cooling load fraction	-	0.90	0.65	0.50	0.85
LCOE penalty	%	7.40	6.96	5.55	7.57
Annual output penalty	%	2.67	2.50	1.91	2.86
Annual water use reduction	%	70.7	76.9	79.7	70.6

Dry cooling case		Minimize bid price			
Dry cooling ITD	$^{\circ}C$	16	16	24	18
Bid price penalty	%	7.45	7.26	5.65	7.87
Annual output penalty	%	3.42	3.12	2.51	3.93
Annual water use reduction	%	94.5	94.6	93.6	94.3

Hybrid cases		Minimize bid price			
Hybrid ACC ITD	$^{\circ}C$	24	28	28	28
Wet cooling load fraction	-	0.95	0.95	0.55	0.95
Bid price penalty	%	5.62	5.30	4.84	5.89
Annual output penalty	%	3.04	3.02	1.90	3.61
Annual water use reduction	%	69.4	68.7	78.4	67.7

Table 5: A summary of results from this analysis. Wet cooling and dry cooling cases are compared with optimal hybrid cooling configurations in terms of LCOE (real) and bid price. Note the reduction in the bid price penalty for hybrid cooling compared to dry cooling for all locations.

- The penalty on bid price for switching from wet to dry cooling is between 5.65% and 7.87% for cool and hot climates, respectively.
- Hybrid cooling during peak TOD periods can reduce the dry cooling bid price penalty by nearly 2% in hot climates and somewhat less in a cooler climate.
- Bid price optimization configures the cooling system such that the penalty on annual output increases. However, this is offset by reduced system cost.
- The impact of hybrid cooling on revenue is tied directly to a heavily weighted TOD rate structure. The LCOE optimization more closely represents typical results for a non-weighted TOD schedule.

Typically, plant designers select an ACC ITD value to meet a desired power cycle efficiency at a relatively high design ambient temperature. This is evidenced by the air cooling optimization results shown in Table 5. But during most of the year, the ACC operates at a comparatively low ambient temperature, and the power cycle efficiency is limited not by the size of the ACC, but by the minimum allowable condenser pressure. For systems using hybrid cooling, the over-sized ACC is no longer optimal since the wet cooling system can share the heat rejection load during periods of high ambient temperature and significantly reduce the operating ITD. Therefore, the ITD design point for hybrid cooling should be selected to more closely represent the average operating condition rather than the maximum operating temperature.

Hybrid cooling also offers a heat rejection option that mitigates water use relative to traditional wet cooling. The magnitude of the bid price reduction for hybrid cooling is inversely proportional to the amount of water used on an annual basis; consequently, plants designed for heavily weighted TOD markets will benefit from maximizing water use during the hottest and most heavily weighted TOD periods. This is illustrated in Figure 4 above. CSP plant design should account for the local water rights restrictions, but this analysis shows that strategic water use coupled with a bid-price-optimized design can provide an improvement in project IRR.

References

- [1] Public Utilities Commission of the State of California. Energy Division Resolution E-4298. Website, December 2009. http://docs.cpuc.ca.gov/PUBLISHED/FINAL_RESOLUTION/111386.htm.
- [2] M. Wagner and C. Kutscher. The impact of hybrid wet/dry cooling on concentrating solar power plant performance. Phoenix, Arizona, USA, 2010. Proceedings of the 4th International Conference on Energy Sustainability, ASME.
- [3] C. Richter and J. Dersch. Methods for reducing cooling water consumption in solar thermal power plants. Berlin, Germany, 2009. Proceedings of the SolarPACES International Symposium, SolarPACES.
- [4] P. Gilman. Solar Advisor Model User Manual. Technical report, National Renewable Energy Laboratory, Golden, Colorado (USA), April 2010.
- [5] M. Erbes, J. Rheinländer, and S. Bergmann. Technical and economic performance of parabolic trough solar power plants - a computational tool for plant feasibility studies. Las Vegas, Nevada, USA, March 2008. Proceedings of the SolarPACES International Symposium.
- [6] C. DeBellis, M. Williams, and Pallavi Dalvi. The basics of air cooled heat exchangers, v2.1. Program applet, 2008. <http://www.hudsonproducts.com/download.html>.
- [7] Matches. Cost information exchange. Website, 2010. <http://www.matche.com/EquipCost/Cooling.htm>.
- [8] Chemical Engineering. Plant cost index. Website, 2010 June. www.che.com/pci.

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14. ABSTRACT (Maximum 200 Words) This paper explores the impact of cooling technology on revenue for hybrid-cooled plants with varying wet cooling penetration for four representative locations in the American Southwest. The impact of ACC design-point initial temperature difference (ITD - the difference between the condensing steam temperature and ambient dry-bulb) is also included in the analysis.						
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