Gas Turbine/Solar Parabolic Trough Hybrid Designs

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GAS TURBINE/SOLAR PARABOLIC TROUGH HYBRID DESIGNS

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ABSTRACT

A strength of parabolic trough concentrating solar power (CSP) plants is the ability to provide reliable power by incorporating either thermal energy storage or backup heat from fossil fuels. Yet these benefits have not been fully realized because thermal energy storage remains expensive at trough operating temperatures and gas usage in CSP plants is less efficient than in dedicated combined cycle plants. For example, while a modern combined cycle plant can achieve an overall efficiency in excess of 55%; auxiliary heaters in a parabolic trough plant convert gas to electricity at below 40%. Thus, one can argue the more effective use of natural gas is in a combined cycle plant, not as backup to a CSP plant. Integrated solar combined cycle (ISCC) systems avoid this pitfall by injecting solar steam into the fossil power cycle; however, these designs are limited to about 10% total solar enhancement. Without reliable, cost-effective energy storage or backup power, renewable sources will struggle to achieve a high penetration in the electric grid. This paper describes a novel gas turbine / parabolic trough hybrid design that combines solar contribution of 57% and higher with gas heat rates that rival that for combined cycle natural gas plants. The design integrates proven solar and fossil technologies, thereby offering high reliability and low financial risk while promoting deployment of solar thermal power.

INTRODUCTION

CSP plants use sunlight to heat a fluid that is used to drive a thermodynamic heat cycle, often a Rankine steam cycle. CSP technologies include parabolic trough, power tower and dish/engine systems. Parabolic troughs, represented by the SEGS plants in southern California, Nevada Solar One outside Las Vegas, and multiple plants in Spain, are the most mature CSP technology. The mirrored collectors track the sun from east to west during the day, to ensure that the sunlight is continuously focused on a linear receiver. A heat transfer fluid (HTF) is circulated through the receiver and returns to a series of heat exchangers in the power block where the fluid is used to generate high-pressure, superheated steam. The superheated steam flows to a conventional Rankine-cycle steam turbine to generate electricity, see Figure 1. Linear Fresnel systems are conceptually similar to parabolic trough plants but use a sequence of flat or near-flat mirrors instead of a parabolic collector.

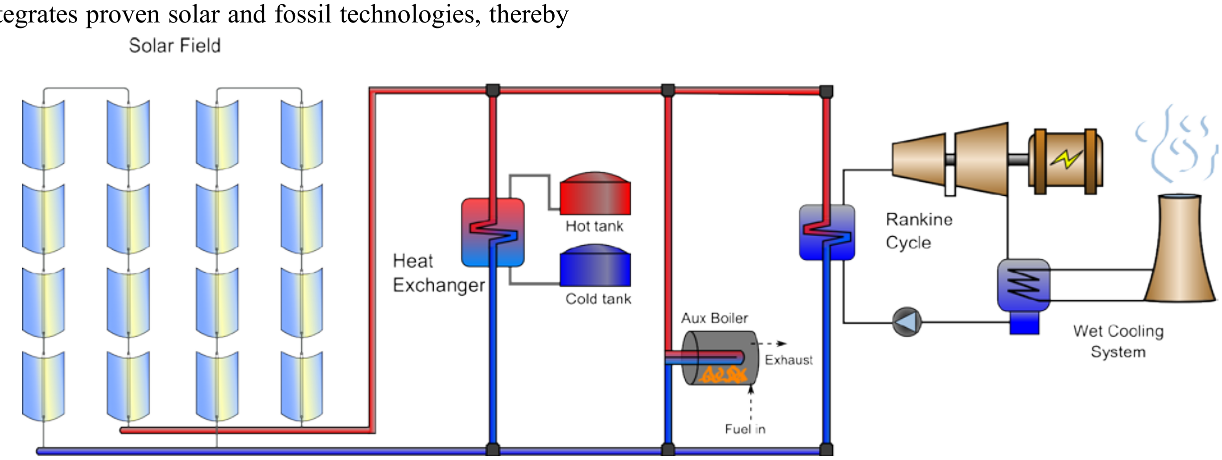


Figure 1. Representation of a parabolic trough plant with thermal energy storage and auxiliary fossil backup. (Courtesy of Mike Wagner, NREL)

A solar power tower, also known as a central receiver, generates electric power from sunlight by focusing concentrated solar radiation on a tower-mounted heat exchanger that serves as the "receiver." The system uses hundreds to thousands of sun-tracking mirrors called heliostats to reflect the incident sunlight onto the receiver. The HTF in a power tower is usually water/steam or molten nitrate salt. Power towers differ from troughs in their ability to achieve higher steam temperatures. Designs planned for the United States report steam conditions as high as 565ºC and 140 bar, which is comparable to fossil-fired Rankine plants. While direct steam generation towers are simpler, towers using molten salt HTF can easily integrate thermal storage at minimal cost as shown in Figure 2. This efficient integration of energy storage is unique among renewable energy technologies.

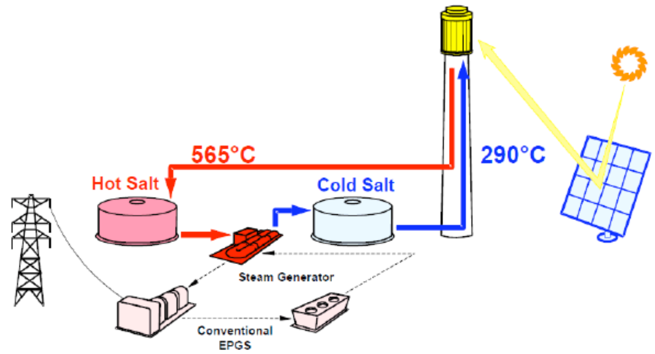


Figure 2. Molten salt Power Tower [1].

The fourth major CSP technology uses a 2-axis tracking parabolic dish to continuously focus sunlight onto the receiver of a Stirling engine. Dish/engine systems range up to 25kW and large plants consist of thousands of units. The lack of a circulating HTF makes hybridization or integration of thermal energy storage difficult and these systems are not considered in the discussion here.

As noted above, parabolic trough, linear Fresnel, and power tower systems can integrate thermal energy storage by storing hot HTF directly or indirectly heating a storage media. These technologies can also utilize natural gas to aid system startup and provide backup power. Both features serve to convert the intermittent solar energy into a reliable, dispatchable resource. Thermal storage and fossil backup are valuable attributes of these CSP technologies. As shown in Table 1, thermal energy storage and fossil backup provide the same benefits with differing cost drivers.

Although TES and fossil backup provide similar benefits there are important distinctions between the two approaches. TES systems maintain a full solar fuel source, but require the installation of substantial hardware, especially for parabolic troughs and linear Fresnel systems. (Molten salt power towers use storage more efficiently because of their higher temperatures and can store salt by simply increasing their tank size and salt inventory.) The cost of tanks and storage media is not trivial, but the greatest increase to capital cost is the increase in solar field size required to provide the energy used to charge storage. In short, adding TES will substantially increase the installed cost of the solar plant. In addition, although conceptually simple, the TES technology is relatively new and entails an added risk for the project.

Table 1. Thermal energy storage and fossil backup both serve to increase reliability and dispatchability from the CSP power plant.

Attribute	TES	Fossil Backup (hybridization)
Generation during clouds	Yes	Yes
Generation after sunset	Yes	Yes
Renewable energy source	Yes	Solar fraction only
Technical risk	Moderate	Low
Capital cost	High	Low
Operating cost	Low	Function of gas price

In contrast, backup via fossil burners has a relatively low investment cost and is mature, low risk technology. While it does not provide renewable power, the solar fraction of the total plant can still be quite high. The greatest downside to the use of natural gas in this fashion is the argument that it would be better burned in a dedicated combined-cycle power plant. A modern natural gas combined cycle (NGCC) plant can achieve thermal cycle efficiencies greater than 55% (heat rate less than 6200 BTU/kWh); whereas a parabolic trough plant has a thermal cycle efficiency of less than 40%. The use of small amounts of gas backup may be justified by the investment in the solar plant infrastructure, but the economics of burning natural gas in auxiliary boilers falls rapidly as gas consumption increases.

Overview of Integrated Solar/Fossil Designs

Various types of integrated solar/fossil plants designs have been described and some ISCC systems have been built, for example the Martin Next Generation Solar Energy Center in Florida. In the 1990s Luz, the builders of the SEGS trough plants in California, proposed hybrid plant designs where solar steam would be used to supplement a combined cycle power plant. Under contract to NREL, Kelly and coworkers [2,3] examined the potential of such designs through a detailed analysis of power cycle performance using GateCycle. The studies focused on sizing equipment for extracting water from feedwater heaters to produce solar steam that was fed to the superheater along with the fossil-produced steam. Overall conclusions held that such a hybrid plant offered three advantages:

- Solar energy was converted to electric energy with an efficiency of about 39 percent; in contrast, the efficiency of the Rankine cycle plants was estimated at about 37 percent;
- The incremental unit cost for the larger steam turbine in the integrated plant was less than the overall unit cost in a solar-only plant; and
- A hybrid plant did not suffer the thermal inefficiencies associated with the daily startup and shutdown of the steam turbine.

Nonetheless, the integrated concept did suffer from two distinct disadvantages. First, in the absence of the solar contribution the steam turbine operated at partial loads during cloudy weather and at night. Operation at partial loads caused an increase in the fossil fuel heat rate, which effectively subtracted from the annual thermal contribution of the collector field to the heat recovery steam generator. Second, the annual solar contribution for plants was in the range of only 2 to 8 percent because higher contributions led to lower overall efficiency.

Recent work by the Electric Power Research Institute (EPRI) examined the optimum means for augmenting coal and NGCC plants with solar thermal energy [4]. While the EPRI study also examined feedwater extraction for solar steam production, the study highlighted the advantages of integrating solar steam into NGCC plants that are designed with gas-fired duct heaters between the gas turbine and boiler. Plants with duct firing are designed with slightly oversized steam turbines and the use of natural gas for duct firing is inherently less efficient than gas used in the gas turbine. In addition, most duct-firing occurs during hot afternoons that correspond to peak demand in the U.S. and lower gas turbine performance. Thus, replacing gas-fired duct heating with solar thermal energy is a natural match.

EPRI's analysis examined a host of different extraction and injection strategies for solar augmentation. The best cases led to improved solar use efficiency (versus a solar-only plant) and improvements in gas heat rate. In general, solar use efficiency drops and heat rate improves as the amount of injected solar thermal energy increases and the optimum combination for maximum solar contribution and efficiency occurred in the range of about 10% solar contribution. This is consistent with the design proposed for most ISCC projects in the United States.

Bohn and Williams [5,6] evaluated a molten salt power tower hybrid design that provides preheated combustion air to a conventional NGCC power plant. Three plants of capacities (30, 100 and 300 MWe) were examined and compared with a solar-only 100 MWe plant and with a NGCC plant of similar capacity. The work studied hybrid plant solar fraction, capacity factor, material cost, and fuel cost effects on economic competitiveness. The study concluded that the hybrid design was more cost effective than any of the solar-only configurations. Because of the higher temperatures required, this concept applies to power tower only.

The Solar Hybrid Gas Turbine Electric Power System (SOLGATE) uses a high temperature power tower for direct air heating to drive a gas turbine [7]. The design focused on concentrating collectors to generate air temperatures in excess of 1000ºC through 3-stage heaters. SOLGATE uses solar energy to raise air temperature close to gas combustion temperature in a gas turbine combustor and then uses a booster burner to bring the gas temperature to a desired level for maximum efficiency. SOLGATE achieved 47.5% efficiency on a combined cycle configuration, or 37.3% in simple cycle with turbine exhaust heat recuperation. The solar contributions on both cases are high too, with 56.8% in combined cycle and 35.04% for simple cycle. The concept is only applicable to power tower systems and has significant material challenges to handle the high solar fluxes; however, the combination of efficiency and solar contribution is promising. In general, the higher temperatures possible with power tower configurations grant them a greater potential for efficient hybridization.

Integrating Aeroderivative Turbines with CSP

Aeroderivative turbines offer a unique set of features for fossil backup of parabolic trough plants. (The concept described here is equally applicable the linear Fresnel systems, but further discussion will focus on the trough design.) As illustrated by Wacek and Ferguson [2], aeroderivative turbines excel at quick and frequent cycling. A 10 minute profile from cold-start to full load for a GE LM6000 is shown in Figure 3. The aeroderivative turbine heats up very quickly (less than six minutes) due to the low mass compared to a frame gas turbine. Ramping to full load takes about four minutes at a ramping rate of ~12MW/min. These attributes indicate that such turbines can transition from cold to full load in about 10 minutes, and cycle from standby to full load in just few minutes. Load following in time domains of seconds is also possible.

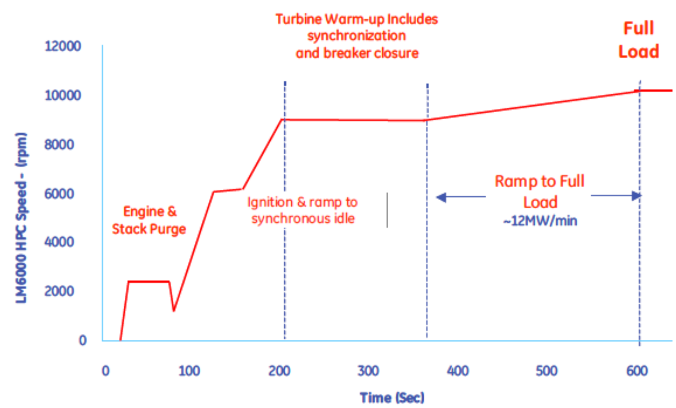


Figure 3. General Electric LM6000 Startup Profile [2].

Wacek & Ferguson highlight these characteristics in their discussion of the use of aeroderivative turbines as backup systems for wind farms. While such complementary operation is valuable, the turbines are not integrated with the wind generators and exist as totally separate generation systems with no shared infrastructure. Representative GE aeroderivative turbine properties are given in Table 1; other turbine suppliers include Rolls Royce and Pratt & Whitney.

Table 1. Selected GE Aeroderivative Gas Turbine Specifications [9,10].

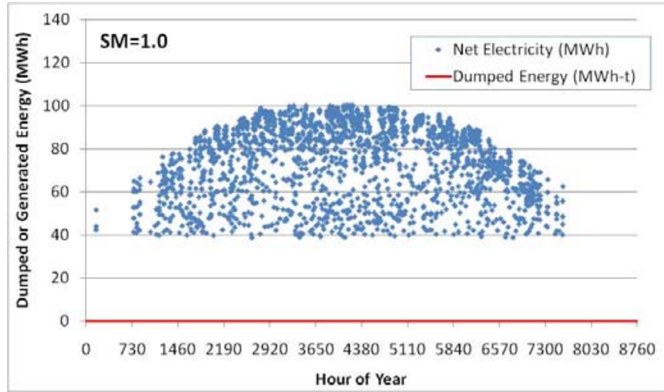
Model	Rated Power (MW)	Heat Rate (BTU/kWh) LHV	Exhaust Temp (C)	Efficiency
LMS100DLE	100	7600	415	44.5%
LM6000PC	42.6	8323	451	41.1%
LM2500PK	30.7	8815	515	38.7%

Modern parabolic trough plants produce superheated steam at just under 390°C, while power towers can operate at steam temperatures near 560°C. As shown in Table 1, the exhaust heat from aeroderivative turbines would be a suitable source of backup heat for parabolic trough plants. This temperature match, combined with the rapid startup and load following capabilities of aeroderivative turbines makes them an excellent match for integration with a parabolic trough plant. In such a role the turbine provides backup heat and additional power, realizing the greater operational efficiency inherent in a combined heat & power system. This design largely overcomes the traditional complaint that gas use for CSP backup is inefficient.

The flexibility of the aeroderivative turbine allows for multiple hybrid configurations with parabolic troughs. One simple conceptual design for aeroderivative turbine backup of a parabolic trough plant is shown in Figure 5. The aeroderivative turbine is started when backup heat is anticipated. Within 10 minutes the system is at full power, providing power from its generator and enough thermal energy to the solar HTF to support operation of the steam cycle power block. The size and number of turbines can be varied as desired, but we will examine cases using relatively small turbines in order to maintain a high solar contribution. Based on this constraint, the aeroderivative turbine(s) are normally providing only a fraction of the full design thermal energy to the steam cycle. Depending on the design, this may or may not be sufficient to run the steam cycle without solar input.

A key design parameter for CSP plants is the solar multiple – the ratio of solar field thermal energy to power block thermal demand at design point conditions. A trough plant without thermal energy storage is normally designed with a solar multiple between 1.2 and 1.5, assuming the reference solar conditions correspond to a clear summer afternoon for the site in question. A design using a solar multiple of 1.0 would be capable of design-point operation for relatively few hours per year (Figure 4, top), forcing part load steam turbine operation for the remainder of the year. Accordingly, trough plants are designed with a solar multiple greater than one. The greater solar multiple allows the plant to run at design point for a large fraction of the year, but necessitates dumping excess solar energy for much of the summer (Figure 4, bottom).

Figure 4 shows that a plant with a solar multiple equal to one will use all solar energy, but operate within 10% of design (>90 MW) for only 24% of its operating hours. In contrast, the design with a solar multiple = 1.4 runs near design point for approximately 60% of its operating hours, but must dump 86,000 MWh of thermal energy over the course of the year. Using NREL's Solar Advisor Model (SAM, version 2010-10-08), one can show that despite costing approximately $100M to pay for the larger solar field (at default costs in SAM), the latter case provides the lower levelized cost of electricity. However, clearly neither is optimal. One path to avoid this difficulty is to install thermal energy storage, thereby avoiding the dumping of excess energy. This adds cost and complexity to the plant, and at current TES prices, slightly raises the LCOE for parabolic trough plants. The alternative described here is to use one or more aeroderivative turbines to fill-in the energy shortfall of the low solar multiple plant.



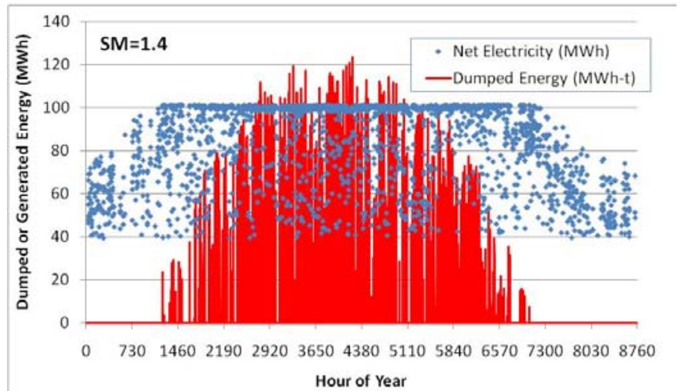


Figure 4. Net electricity production and dumped thermal energy for every hour of the year for 100 MW trough plant with a solar multiple = 1.0 (top) or solar multiple = 1.4 (bottom).

Perhaps the simplest integration into a trough plant is to allow the turbine exhaust gases to heat the circulating HTF. This configuration is suited to "topping off" the thermal energy during cloud transients or less than ideal solar resource. This configuration would be a relatively simple retrofit for existing trough plants. Providing heat to the HTF facilitates the use of molten salt HTF in troughs or linear Fresnel systems because the exhaust heat from gas turbine provides sufficient energy to maintain the circulating salt at a temperature well above its freezing point. This design could run a small gas turbine overnight or during extended shutdowns to keep the molten salt at the desired temperature. Furthermore, molten salt HTF allows for easy integration of direct storage of the HTF for thermal energy storage, thereby opening new options for alternative operating modes and design optimization.

Waste heat from the turbines could provide thermal energy to keep the steam turbine and/or cooling system in a standby mode, thereby avoiding the need to break and reset steam seals overnight or during shutdowns. Conceptually this uses the gas turbine more like a combined heat and power system rather than a combined cycle system.

Improvements in cycle efficiency can be realized through a variety of alternative configurations. For example, gas turbine exhaust exiting the HTF heat exchanger may be used within one or more feedwater preheaters as shown in Figure 5. Gas turbine exhaust could be used to provide steam superheat or reheat. A reheat mode is attractive because it could extract a substantial fraction of energy from the turbine exhaust. Both superheat and reheat configurations would likely require the turbine(s) to run whenever the solar field is in operation. With multiple different configurations possible, a process performance model will be an essential tool in selecting and optimizing designs.

Lastly, while these designs are described for parabolic trough and linear Fresnel systems, it is possible that similar options could be applied to power tower CSP systems, especially if higher exhaust temperatures can be achieved from the gas turbines. The concepts could also be used with CSP plants using water steam HTF, that is, direct steam generation systems.

Hybrid Plant Design

The thermal energy balance for the integration of aeroderivative turbines into a trough plant was undertaken using IPSEPro. The IPSEPro software package has components for solar troughs, Rankine steam cycles, and aeroderivative turbines. The IPSEPro simulation indicated a single LM6000 turbine running at design could supply sufficient thermal energy for the equivalent of approximately 10 MWe from the steam cycle when integrated via an HTF heater and feedwater heat exchanger (see Figure 5). For comparison, twin LM6000s can drive a 55-MW steam cycle when configured specifically for that purpose in GE's 2-on-1 combined cycle plant. The larger steam cycle contribution in the 2-on-1 plant design likely reflects the higher steam conditions and dedicated hardware yielding better heat recovery.

As the gas turbines become large relative to the steam cycle, their output dominates the total plant output. Their ability to respond to changes in solar energy is limited by the variation in total plant output caused by the gas turbine electric output. That is, starting 80 MW of gas turbines to respond to a 20 MW drop in solar power may not be desirable. Because one goal of the hybridization is to minimize plant variability, a design was selected where the turbines are relatively small compared to the steam cycle. While the AGT waste heat maybe insufficient to run the steam cycle in the absence of solar energy, it is sufficient to preheat the steam turbine and provide freeze protection heating to the HTF.

The configuration used for the following analysis assumes a single GE LM6000 turbine combined with a 100 MW parabolic trough plant as depicted in Figure 5. A simpler configuration without use of an air-to-feedwater heater was also examined but ultimately discarded due to its lower thermal-use efficiency. The design included the following assumptions:

- a gas/HTF heat exchanger sized to produce 395°C oil matching the solar field exit design,
- a dedicated gas/water feedwater heater used downstream of the gas/HTF heat exchanger and sized to heat boiler feedwater to same temperature as the regular feedwater heater train,
- a feedwater heater based on steam turbine extraction flows (not shown in Figure 5) shuts off when the gas turbine is operating,
- a vacuum deaerator is used instead of an open feedwater heater using steam turbine extraction flow,
- back pressure on the turbine is increased by 4 inch water (10 mbar) to account for the downstream heat exchangers, and
- gas turbine output was derated based on ambient temperature for the site using the same weather file that supplied the hourly solar input and IPSEPro's input for turbine performance as a function of temperature.

This design approach led to an operating strategy where the AGT is started prior to sunrise every morning, run at full load throughout the day, and shut down shortly after sunset (see Figure 6). A low solar multiple allows the contribution from the AGT waste heat to be incorporated into the steam cycle throughout most of the year, while rarely having to dump any solar energy.

RESULTS

Four different plant configurations were compared for the initial analysis. These included two solar-only plants without storage and two solar trough / AGT hybrids. The designs are summarized in Table 2. Trough plant steam cycle performance was modeled in SAM using Daggett, CA as the test site. The SAM fossil backup option was used to simulate the presence of waste heat from the AGT. Hourly net electricity, fossil energy consumption, and ambient temperature were exported from SAM to Excel where the hourly contribution from the AGT was added to arrive at total generation.

The total installed cost for the solar hardware was based on default values provided in SAM version 2010-10-08. These costs include combined solar field, site preparation and HTF System at \$410/m^2 and wet-cooled power block at \$940/kWe. A 10% contingency and 24.7% indirect cost multiplier are applied to the direct costs. Gas turbine direct costs were assumed to be \$900/kWe and the same contingency and indirect cost multipliers were used to arrive at an installed cost.

The first two plants in Table 2 are solar-only designs differing in the solar multiple. Case S1.4 costs \$74M more than S1.1 due to the larger solar field, but S1.4 produces 17% more energy over the course of the year and is able to run near design point for about 55% of its operating hours versus 36% for the smaller S1.1 plant. However, S1.4 has lower solar-use efficiency because it must dump sunlight during the summer. The LCOE for S1.4 is slightly lower, suggesting it is the favored solar-only design.

For the hybrid plants H1.1 and H1.2, operation of the gas turbine was nominally begun an hour before sunrise and ended an hour after sunset every day. Starting the AGT before sunrise allowed the waste heat to be used for steam turbine startup. The gas turbine was run at full load when used and the waste heat was incorporated into the steam cycle whenever there was sufficient capacity. Shutting off the AGT during high-insolation periods may improve its annual efficiency, but these periods often coincide with high demand and for this analysis the turbine was assumed to run throughout the day. Backpressure from the downstream heat exchangers lowered the AGT capacity from a nominal 40 MW to 37.8 MW.

Table 2. Solar Only and Hybrid Configurations.

Case	S1.1 Solar Only	S1.4 Solar Only	H1.1 Solar + AGT	H1.2 Solar + AGT
Solar Multiple	1.1	1.4	1.1	1.2
Steam Turbine Capacity (MW)	100	100	100	100
AGT Capacity (MW)	-	-	37.8	37.8
Steam Turbine Annual Gen. (MWh)	210,300	245,900	231,000	244,100
AGT Annual Gen. (MWh)	-	-	143,800	143,800
Total Annual Gen. (MWh)	210,300	245,900	374,800	387,900
Solar Fraction	1.00	1.00	0.57	0.59
Annual solar efficiency (MWhe/MWh)	15.9%	14.5%	16.1%	15.7%
Total Installed Cost ($M)	410	484	459	485
Effective heat rate (BTU/kWh)	-	-	7360	7460
Gas Consumption (MMBTU/yr)	-	-	1,193,000	1,193,000
Total LCOE * ($/MWh)	243	240	175	176
Gas LCOE * ($/MWh)	-	-	105	105
Solar LCOE * ($/MWh)	243	240	228	226

* levelized cost of electricity, no incentives included

Comparison of heat rates indicates the benefit of integrating the gas turbines with the steam cycle. The heat rate for the LM6000 is about 8300 BTU/kWh, based on values reported by GE [1]. When the additional power production from the thermal contribution to the steam turbine is included, the effective heat rate drops to 7360 to 7460 BTU/kWh depending on design assumptions. The heat rate is slightly lower for the H1.1 case because the smaller solar field leaves capacity for the AGT heat in the steam cycle for more of the year. The low effective heat rate indicates that the goal of utilizing gas at efficiencies comparable to NGCC plants is possible.

A simple LCOE analysis is included in Table 2 based on an 8% real discount rate and 20-year analysis period giving a uniform capital cost recovery factor (UCRF) of 0.1018. Total 20-year life cycle costs were multiplied by the UCRF and divided by annual energy production to yield LCOE [11]. Solar hardware O&M cost was calculated using SAM's default value of $70/kW-yr. Natural gas price was set at $6/MMBTU and, like other O&M costs, was escalated with inflation. (Note: gas price is quoted based on higher heating value HHV, whereas values in Table 2 use lower heating value, LHV. Operating cost is adjusted to account for this difference of ~10%) The Solar LCOE for the hybrid plant is calculated by solving:

$$(LCOE)_{total} = x_{gas} * (LCOE)_{gas} + x_{solar} * (LCOE)_{solar}$$

where x_{gas} and x_{solar} are the fractional contributions to total generation. No investment tax credits are applied in the LCOE calculations, but accelerated depreciation is used for the solar hardware. The LCOE and annual solar efficiency indicate that the hybrid plant is slightly more efficient in its use of solar energy. These costs are based on energy value alone and no credit has been given for the greater reliability/dispatchability of the hybrid plant.

A comparison of the hybrid H1.1 design and the solar-only S1.4 design reveals the following advantages of the hybrid system:

- ✓ lower installed cost
- ✓ lower solar LCOE
- ✓ greater annual generation
- ✓ higher solar efficiency
- ✓ lower heat rate (vs. combustion turbine only)

While each of these benefits is relatively modest, combined they indicate a clear advantage for the hybrid design. In addition to these quantitative advantages, the hybrid system utilizes commercially proven technologies and provides greater dispatch reliability.

CONCLUSIONS

Solar/fossil hybrid designs reduce the impact of solar intermittency by providing either fossil backup to the solar plant or integrating solar output into a much larger fossil power installation. Hybrid designs utilize shared infrastructure that reduces the capital cost compared to separate stand-alone plants. However, traditional solar/fossil hybrid designs have been hampered by poor gas utilization efficiency and/or limited solar contribution. Incorporation of aeroderivative gas turbines overcomes these limitations and expands the hybrid design options available to developers.

In the preliminary analysis provided here, it is shown that a single 40-MW aeroderivative gas turbine mated with a 100-MW parabolic trough plant can be more efficient than two separate power plants. The estimated solar fraction for the concepts examined was 57% to 59%. The design utilizes proven parabolic trough and aeroderivative turbine hardware. While various integration schemes are proposed, this initial analysis looked at only one configuration. Further examination is expected to yield alternatives that provide greater operating or efficiency benefits than that described here.

Ongoing work is exploring alternative integration options by modeling in IPSEPro process simulation software. Results from IPSEPro are combined with SAM simulations to predict hourly system performance for a full year. Further research will investigate how to incorporate thermal energy storage and ways to optimize use of the gas turbines to maximize power production during peak demand periods and minimize overall ramp rates from the combined steam turbine / gas turbine output.

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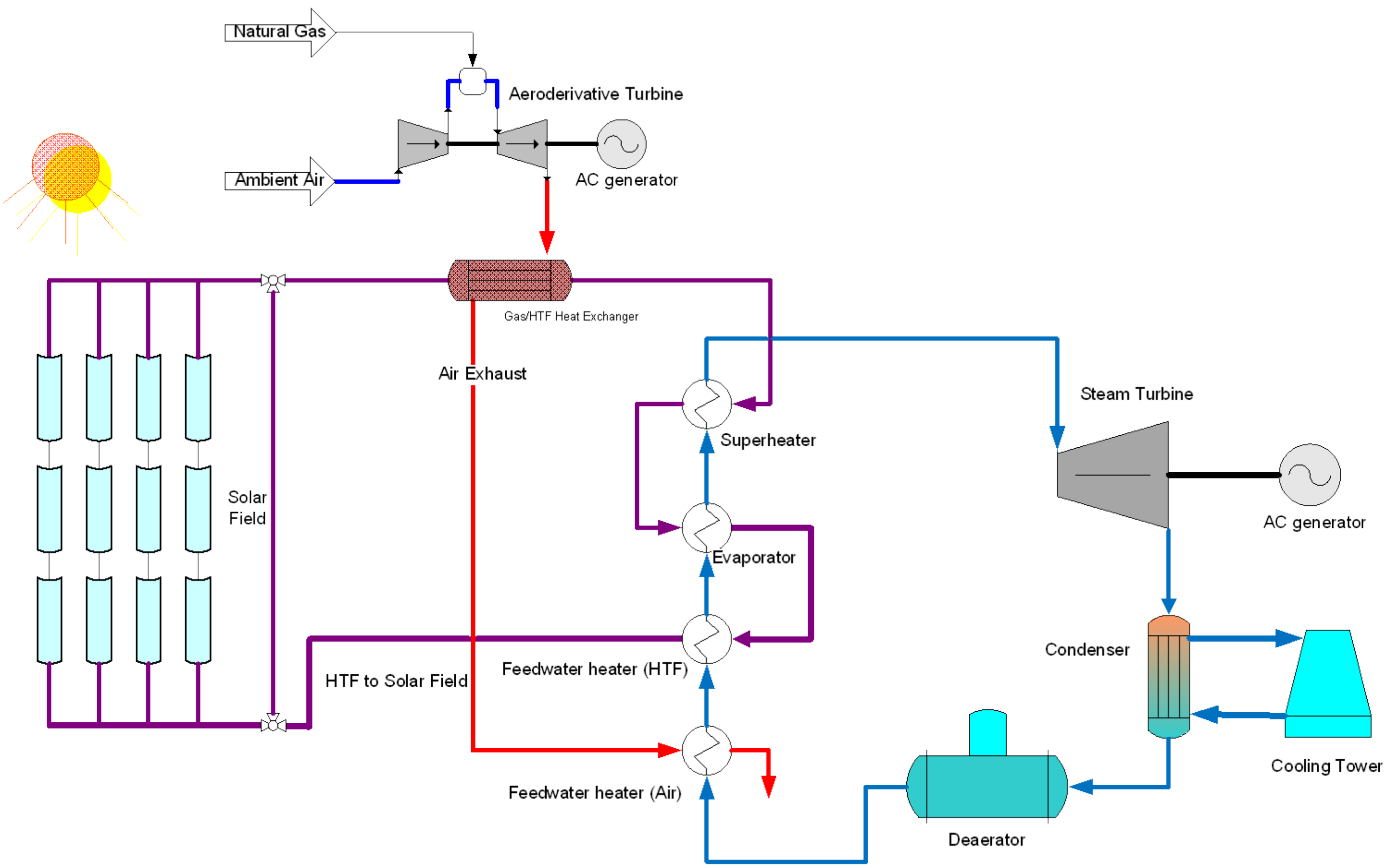

Figure 5. Simplified schematic of aeroderivative turbine backup for a parabolic trough plant. Additional feedwater heaters using steam extraction are not shown.

	Hour of Day																							
	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**
Jan	0	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0
Feb	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0
Mar	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
Apr	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
May	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Jun	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Jul	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Aug	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0
Sep	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0
Oct	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0
Nov	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0
Dec	0	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	0	0	0	0	0	0	0	0

Figure 6. Operating strategy for the aeroderivative turbine integrated with a solar trough plant (1= running; 0=off).

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A strength of parabolic trough concentrating solar power (CSP) plants is the ability to provide reliable power by incorporating either thermal energy storage or backup heat from fossil fuels. Yet these benefits have not been fully realized because thermal energy storage remains expensive at trough operating temperatures and gas usage in CSP plants is less efficient than in dedicated combined cycle plants. For example, while a modern combined cycle plant can achieve an overall efficiency in excess of 55%; auxiliary heaters in a parabolic trough plant convert gas to electricity at below 40%. Thus, one can argue the more effective use of natural gas is in a combined cycle plant, not as backup to a CSP plant. Integrated solar combined cycle (ISCC) systems avoid this pitfall by injecting solar steam into the fossil power cycle; however, these designs are limited to about 10% total solar enhancement. Without reliable, cost-effective energy storage or backup power, renewable sources will struggle to achieve a high penetration in the electric grid. This paper describes a novel gas turbine / parabolic trough hybrid design that combines solar contribution of 57% and higher with gas heat rates that rival that for combined cycle natural gas plants. The design integrates proven solar and fossil technologies, thereby offering high reliability and low financial risk while promoting deployment of solar thermal power.

15. SUBJECT TERMS

parabolic trough; aeroderivative turbine; solar; hybrid

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