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OPTIMIZATION STUDIES FOR INTEGRATED SOLAR COMBINED CYCLE SYSTEMS

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ABSTRACT

The integrated solar plant concept was initially proposed by Luz Solar International [1] as a means of integrating a parabolic trough solar plant with modern combined cycle power plants. An integrated plant consists of a conventional combined cycle plant, a solar collector field, and a solar steam generator. During sunny periods, feedwater is withdrawn from the combined cycle plant heat recovery steam generator, and converted to saturated steam in the solar steam generator. The saturated steam is returned to the heat recovery steam generator, and the combined fossil and solar steam flows are superheated in the heat recovery steam generator. The increased steam flow rate provides an increase in the output of the Rankine cycle. During cloudy periods and at night, the integrated plant operates as a conventional combined cycle facility.

Two studies on integrated plant designs using a General Electric Frame 7(FA) gas turbine and a three pressure heat recovery steam generator are currently being conducted by the authors. Preliminary results include the following items: 1) the most efficient use of solar thermal energy is the production of high pressure saturated steam for addition to the heat recovery steam generator; 2) the quantity of high pressure steam generation duty which can be transferred from the heat recovery steam generator to the solar steam generator is limited; thus, the maximum practical solar contribution is also reasonably well defined; 3) small annual solar thermal contributions to an integrated plant can be converted to electric energy at a higher efficiency than a solar-only parabolic trough plant, and can also raise the overall thermal-to-electric conversion efficiency in the Rankine cycle; and 4) annual solar contributions up to 12 percent in an integrated plant should offer economic advantages over a conventional solar-only parabolic trough power plant.

INTRODUCTION

Between 1984 and 1990, a total of nine Solar Electric Generating Station solar power plants were built in the southern California desert. Each plant used parabolic trough solar collectors to heat either a mineral oil or a synthetic heat transfer oil; thermal energy in the oil was used to generate steam, and the steam drove a conventional Rankine cycle power plant. However, for a variety of economic reasons, no new domestic or international parabolic trough power plants have been built since that time.

The integrated solar plant concept was initially proposed by Luz Solar International as a means of integrating a parabolic trough solar plant with modern combined cycle power plants. A plant schematic diagram is shown below.

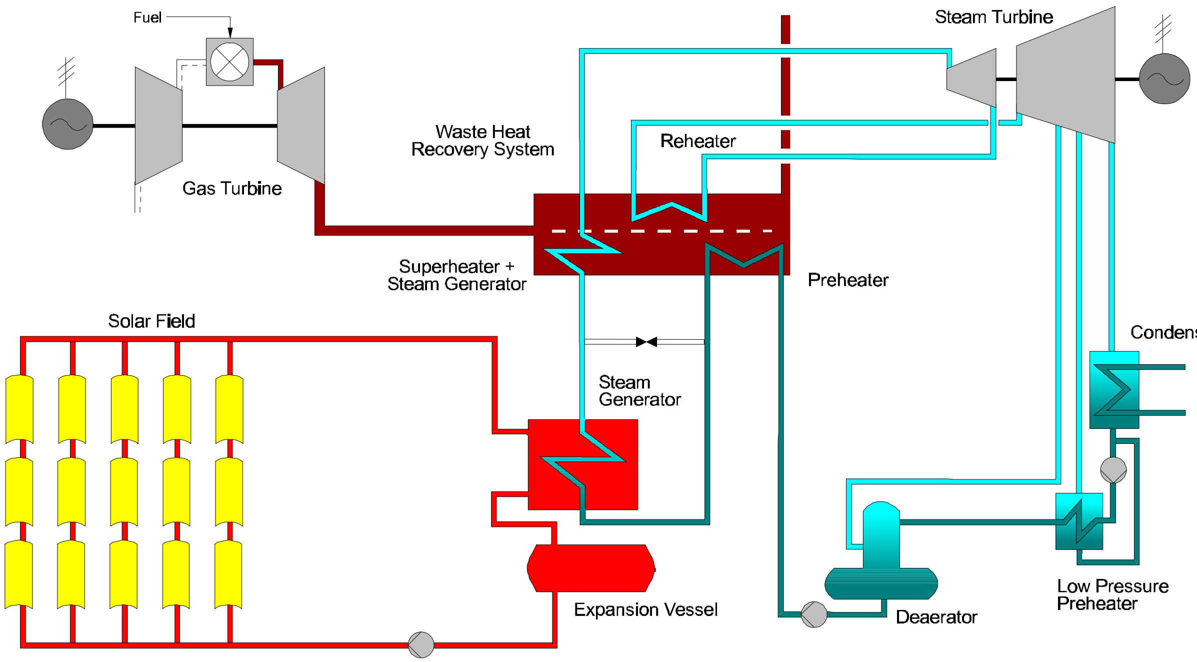

Figure 1: Integrated solar plant schematic diagram.

In comparison with the existing Rankine cycle plants, an integrated plant offers three principal advantages. First, solar energy can be converted to electric energy at a higher efficiency. Second, the incremental unit cost for the larger steam turbine in the integrated plant are less than the overall unit cost in a solar-only plant. Third, an integrated plant does not suffer the thermal inefficiencies associated with the daily startup and shutdown of the steam turbine.

Optimization studies in integrated plant concepts are currently underway in the following projects: 1) USA Trough Initiative project, sponsored by the National Renewable Energy Laboratory [2-4], and 2) the Trough Integration into Power Plants project sponsored by Deutsches Zentrum für Luft- und Raumfahrt; Flabeg Solar International, Inc., the National Renewable Energy Laboratory, and Sandia National Laboratories [5].

Combined Cycle Plant

Each study used as its basis a commercial combined cycle plant with the following equipment:

- General Electric PG7241(FA) gas turbine-generator. The fuel was natural gas, and international standards for oxides of nitrogen emissions were met by using dry, low NOx combustors.
- Three pressure heat recovery steam generator. The high, intermediate, and low pressure steam conditions were as follows: 100 bar and 565 °C; 28 bar and 565 °C; and 4 bar and 290 °C. The fuel was assumed to be free of any contaminants which might produce corrosive compounds in the last stages of the heat recovery steam generator. Thus, a design stack temperature of 80 °C was selected to recover as much energy from the turbine exhaust as possible.
- Single reheat steam turbine cycle.

The site selected for the studies was Barstow, California, with an elevation of 600 m. On a day with an ambient temperature of 25 °C and a relative humidity of 40 percent, the output of the Brayton cycle was 153.9 MWe and the output of the Rankine cycle was 90.0 MWe, for a total plant output of 243.9 MWe. The gross heat rate and plant efficiency, based on the lower heating value of natural gas, were 6,315 kJ/kWhe and 57.0 percent, respectively.

Performance Models

The output from, and fuel consumption of, the gas turbine were strong functions of the ambient temperature. In addition, the output from the steam turbine plant was a strong function of the thermal input from the collector field. Thus, the performance of the plant was best determined by calculating the outputs of the gas turbine and steam turbine on an hourly basis, and summing the results over the course of a year.

The performance of the conventional combined cycle plant was calculated as follows:

- A conceptual plant design was developed using the GateCycle program [6] from Enter Software, Inc.

The ambient temperature for the design point calculations was 25 °C. The program calculated capacities for the gas turbine-generator, the heat recovery steam generator, and the steam turbine plant, and performed heat and mass balances on all of the inter-plant streams.

- A series of GateCycle runs were made at ambient temperatures of -1 ºC, 10 ºC, 27 ºC, 38 ºC, and 49 ºC to determine gas turbine output, steam turbine output, and fuel consumption. Since the ambient temperatures were different than the design point value of 25 °C, each GateCycle calculation was an off-design analysis. Two-dimensional curve fits of the turbine outputs and fuel use, as functions of the ambient temperature, were developed.
- A file of hourly direct normal radiation and coincident dry bulb temperatures was developed for the site.
- An Excel spreadsheet was developed which listed or calculated, for each hour in the year, the dry bulb temperature, gas turbine output, steam turbine output, and fuel use.

The performance of the integrated plant was calculated in a manner similar to that for the conventional combined cycle plant. The principal exception was the performance of the steam turbine, which was a weak function of the ambient temperature and a strong function of the thermal input from the collector field. The thermal input from the collector field was estimated by calculating the following:

- Hour of the day, time before noon, day of the month, and month of the year
- Each of the following angles: solar declination; sun elevation; sun azimuth; and collector incidence. From the collector incidence angle, an incidence angle modifier was calculated to account for the reflected flux which misses the end of the heat collection element during the midday hours.
- Gross field thermal output, by multiplying the following: collector area; collector optical efficiency of 72 percent; incidence angle modifier; and the difference between the direct normal radiation and 84.3 W/m^2. The latter value corresponded to the thermal losses from the heat collection elements.
- Net field thermal output, by multiplying the gross output by 0.9805 to account for thermal losses from the field piping.

For each of 25 potential plant designs evaluated in the studies, a series of 25 GateCycle runs were made at ambient temperatures of -1 ºC, 10 ºC, 27 ºC, 38 ºC, and 49 ºC, concurrently with solar steam flow rates equal to 0, 25, 50, 75, and 100 percent of the design flow rate, to estimate the steam turbine output. In each calculation, the steam turbine operated with flow rates, inlet temperatures, and outlet pressures which were often significantly different from the design point conditions. All of the off design state points and equipment efficiencies were calculated by the GateCycle program.

For each case, a three-dimensional curve fit of ambient temperature, solar steam flow rate, and steam turbine output was developed for incorporation in the Excel spreadsheet. An example of the surface equation is shown in Figure 2.

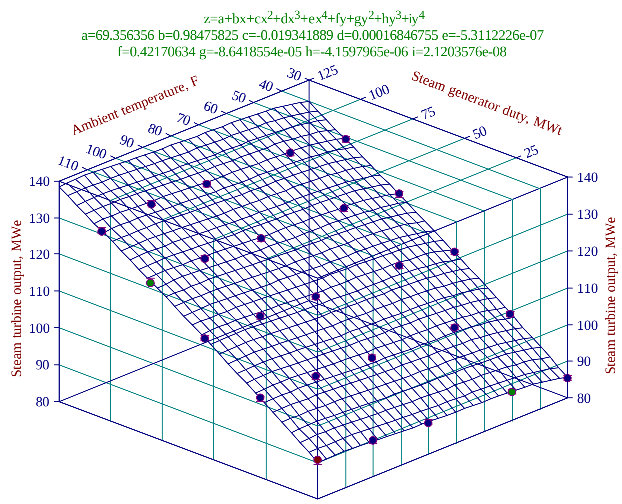

Figure 2: Steam Turbine Output as Functions of Ambient Temperature and Collector Field Output

For simplicity, all of the annual performance calculations were based on plant operation for 8,760 hours each year. Thus, the annual solar contributions discussed below represent the lower limits on the range of possible values. In reality, the plants will follow the demand profiles for the local utility. The plants may be shut down at night, and will likely undergo scheduled maintenance during the winter. As a consequence, the local dispatch requirements will increase the calculated solar contribution.

PRINCIPAL FINDINGS

The studies have examined several aspects of integrated plant designs, including the preferred method for transferring solar energy into the combined cycle, possible range of solar inputs, preferred superheater and reheater heat transfer areas, and possible range of annual solar contributions.

Methods for Transferring Solar Energy

Several methods for transferring solar thermal energy into the combined cycle were evaluated, as follows:

- Withdrawing feedwater from the heat recovery steam generator at either a low or a high temperature; producing low pressure, intermediate pressure, or high pressure saturated steam in the solar steam generator; and returning the saturated steam to the heat recovery steam generator
- Withdrawing feedwater from the heat recovery steam generator at either a low or a high temperature; producing intermediate pressure superheated steam in the solar steam generator; and returning the steam to the gas turbine combustor.
- Adding oil-to-gas heat exchangers in the heat recovery steam generator, and use solar energy to periodically reheat the flue gases.

The most efficient method for converting solar thermal energy to electric energy was to withdraw feedwater from the heat recovery steam generator downstream of the second stage (highest temperature) feedwater economizer, produce high pressure saturated steam, and return the steam to the heat recovery steam generator for superheating and reheating by the gas turbine exhaust.

Range of Possible Solar Inputs

When saturated steam is produced in a solar steam generator, the latent heat transfer duties in the heat recovery steam generator are decreased and the sensible heat transfer duties are increased. The effect is illustrated in the heat transfer diagrams of Figures 3 and 4. The upper line in the figure is the temperature of the turbine exhaust gas in the heat recovery steam generator, and the lower lines are the temperatures in the low, intermediate, and high pressure feedwater and steam sections. Figure 3 represents evening operation, with a steam flow rate in the heat recovery steam generator of 216,000 kg/hr. Figure 4 represents day time operation, with a steam flow rate of 366,000 kg/hr. Of this flow rate, 216,000 kg/hr is provided by thermal energy in the gas turbine exhaust and the balance of 150,000 kg/hr is provided by solar energy.

In Figure 3, the long horizontal line represents the latent heat transfer in the high pressure evaporator. The largest temperature differences in the heat recovery steam generator occur here. In Figure 4, the solar steam generator carries a portion of the saturated steam production duties, and the length of the horizontal temperature line for the high pressure evaporator decreases. As this line shrinks, the average temperature difference in the heat recovery steam generator decreases, and the average temperature of the Rankine cycle working fluid increases. In terms of a heat engine, a kilojoule of energy at a temperature of 350 °C is more useful than a kilojoule of energy at a temperature of 300 °C. As a result, the conversion of fossil energy in the gas turbine exhaust to electric energy in the Rankine cycle is improved by using a solar evaporator in parallel with the heat recovery steam generator.

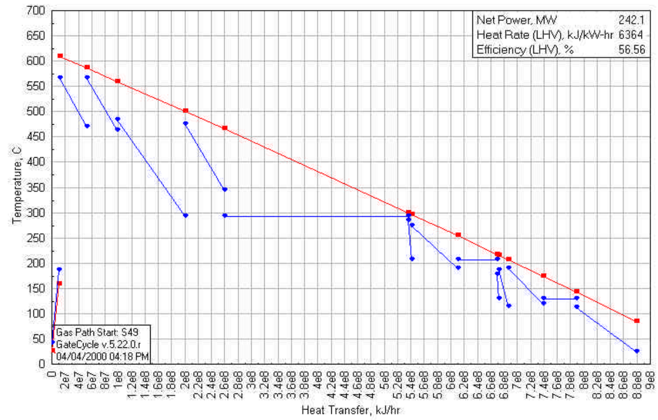

Figure 3: Heat Transfer Diagram with 216,000 Kg/Hr Steam Flow Rate

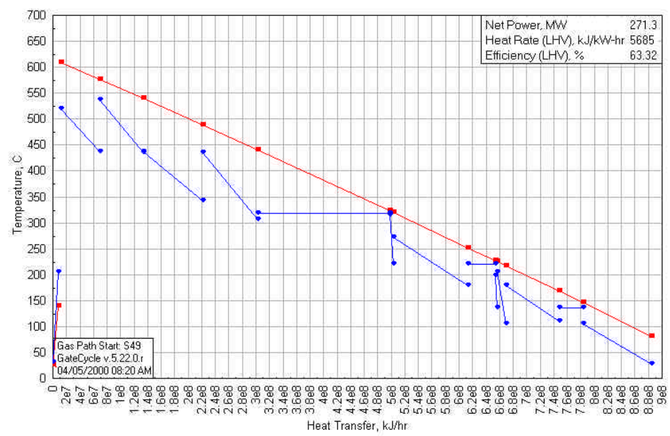

Figure 4: Heat Transfer Diagram with 366,000 Kg/Hr Steam Flow Rate

If the duty of the solar steam generator is small relative to the duty of the high pressure evaporator in the heat recovery steam generator, the latent heat transfer with the largest temperature differences can be avoided. However, as the solar contribution increases, latent heat transfer at smaller temperature differences is avoided, and the potential gains in fossil energy-to-electric energy conversion efficiency decline. For very large solar contributions, solar energy starts to displace some of the sensible heat transfer in the heat recovery steam generator. There is little or no thermodynamic benefit to performing sensible heat transfer in the solar steam generator rather than in the heat recovery steam generator, and the efficiency of converting solar thermal energy to electric energy approaches that of a conventional Rankine cycle. The effect is illustrated in Figure 5, which plots gross annual Rankine cycle efficiency as a function of the design point solar thermal input.

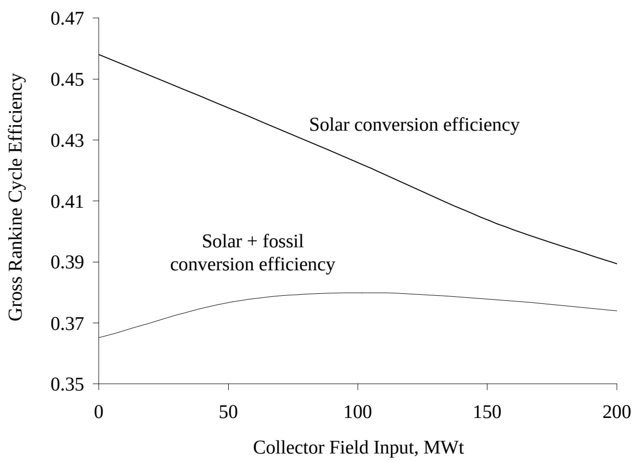

Figure 5: Rankine Cycle Conversion Efficiencies

For the plant designs under consideration, the optimum solar field rating was about 100 MWt. The lower curve illustrates a combination of the following effects:

- For small solar contributions, the solar thermal-to-electric conversion efficiency is very high, and the combined fossil + solar energy-to-conversion efficiency improves. However, as the solar contribution increases, the benefits to the combined fossil + solar efficiency also decrease.
- The steam turbine-generator in an integrated plant must be larger than in a conventional combined cycle plant. Thus, during evening operation and cloudy weather, the steam turbine operates at part load. For a heat engine, the work produced is given by õ dP, where õ is the specific volume. During part load operation, the live steam pressure is lower, and the conversion efficiency is lower. For small solar contributions, the required increase in steam turbine capacity is small, and the part load efficiency penalties are also small. However, for large solar contributions, the steam turbine capacity may be as much as double the size in a conventional plant, and the annual part load efficiency penalty may be 10 to 15 percent of the annual solar thermal contribution to the solar steam generator. The part load efficiency penalty is the reason for the negative slope in the fossil + solar efficiency curve for solar contributions above 100 MWt.

Two items should be noted from a review of Figure 5. First, the optimum solar contribution is not sharply defined; all contributions between 60 and 140 MWt provides essentially equal benefits. Second, the analyses were developed for a three pressure heat recovery steam generator, using a low stack temperature permitted by the natural gas fuel for the gas turbine. Analyses for other plant designs, particularly for sites which have available only fuel oil containing sulfur, may lead to different conclusions regarding the optimum solar contribution.

Superheater and Reheater Sizes

Two approaches was evaluated for selecting the superheater and reheater heat transfer areas in the heat recovery steam generator. In the first approach, the areas were selected to provide the desired steam temperature (for example, 565 ºC) with the combined flow rate of solar steam and fossil steam. Thus, steam temperatures of at least 565 ºC could be maintained during both day and evening operation. However, the heat exchangers were essentially too large for operation only with fossil steam, and feedwater attemperation was required during evenings and cloudy weather.

In the second approach, the areas were selected using the fossil steam flow rate only. Thus, during day time operation with combined solar and fossil steam flow rates, the heat exchangers were not large enough to provide the desired temperature of 565 °C, and live and reheat steam temperatures decayed to values as low as 450 °C. On the positive side, feedwater attemperation was not required, and switching between day and evening operating modes was less complex.

The nominal schedule of live steam pressures and temperatures for the two approaches in a plant with a design live steam pressure and temperature of 125 bar and 565 ºC, respectively, was as follows:

First Design Approach	Steam Pressure	Steam Temperature
Day Operation	125 bar	565 ºC
Evening Operation	70 to 125 bar	565 ºC

Second Design Approach	Steam Pressure	Steam Temperature
Day Operation	125 bar	450 to 565 ºC
Evening Operation	70 to 125 bar	565 ºC

As above, the work performed by the working fluid is given by õ dP. Thus, the first approach offered higher annual solar thermal-to-electric conversion efficiencies than the second because it offered a better combination of high temperature (minimum specific volume) and high pressure. For plant designs with an annual solar contribution of 2 percent, the net solar thermal-to-electric conversion efficiency with the first approach was on the order of 40 percent; with the second, approximately 38 percent. Here, annual solar contribution is defined as follows:

$$\frac{\text{Incremental Rankine output due to solar thermal energy, MWhe}}{\text{Conventional combined cycle plant output + Incremental Rankine cycle output, MWhe}}$$

Range of Annual Solar Contributions

Several integrated plant designs were evaluated to determine the effect of annual solar contribution on the plant performance. From these studies, the following observations could be made:

1) Annual solar contributions in the range of 1 to 2 percent offered net solar thermal-to-electric conversion efficiencies of 40 to 42 percent; however, increasing the solar contribution to 9 percent reduced the net conversion efficiency to values in the range of 32 to 35 percent.
2) As noted in the section above, the optimum annual solar thermal-to-electric conversion efficiencies occur with superheater and reheater heat transfer areas which are large enough to sustain design steam temperatures throughout the day. However, feedwater attemperation between the first and second stage superheaters can occur only up to the point where the steam entering the second stage section is saturated. This limit, in turn, sets the maximum allowable solar thermal contribution at the design point. For the designs analyzed, the maximum annual solar contribution for plants with constant steam temperatures is limited to about 6 percent.
3) To achieve annual solar contributions above 6 percent, plant designs will likely need to use one of the following approaches:

- The superheater and reheater sections are sized such that design steam temperatures are achieved only during non-solar operation
- Superheater and reheater sections can be bypassed, on the steam side, during non-solar periods.

4) To achieve annual solar contributions above 9 percent, the solar steam generator must start to perform some of the heat transfer duties of the feedwater economizers, the intermediate pressure evaporator, and the intermediate pressure superheater in the heat recovery steam generator. Shifting the heat transfer duties for these sections to the solar steam generator does not provide large thermodynamic benefits to the Rankine cycle, and solar thermal-to-electric conversion efficiencies can fall below 35 percent.

As a point of reference, the net annual solar thermal-to-electric conversion efficiency for a new, solar-only parabolic trough power plant is projected to be 32 to 33 percent. Thus, in terms of annual efficiency, integrated plants will be more economic than solar-only plants for annual solar contributions up to about 10 percent.

In addition, the levelized cost of energy in a solar-only plant is a function of the capital cost of the collector system, the capital cost of the Rankine cycle, the annual operation and maintenance cost, and the annual direct normal radiation. Similarly, the levelized cost of solar energy in an integrated plant is a function of the capital cost of the collector system, the incremental cost of the larger Rankine cycle, the incremental operation and maintenance cost of the solar equipment, and the annual direct normal radiation. Ascribing only the incremental costs to the solar facility in an integrated plant provides significant savings compared to a solar-only plant. Detailed economic assessments have yet to be performed, but the savings may allow economic annual solar contributions as large as 12 percent.

CONCLUSIONS

The integrated solar plant concept offers an effective means for the continued development of parabolic trough technology. In a careful plant design, solar thermal-to-electric conversion efficiencies will exceed, often by a significant amount, those of a solar-only parabolic trough project. In addition, an integrated plant bears only the incremental capital cost of a larger Rankine cycle, which provides further reductions in the levelized cost of solar energy.

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