SOFTWARE FOR DESIGN, SIMULATION, AND COST ESTIMATION OF SOLAR THERMAL POWER AND HEAT CYCLES

Patrick Griffin[1], Karsten Huschka[2], Gabriel Morin[3]

[1] Sc.D., Vice President, Thermoflow, Inc., 29 Hudson Road, Sudbury, MA, USA, Tel.: +1-978-579-7999, griffin@thermoflow.com

[2] Dipl.-Ing, Director, Thermoflow Europe, GmbH, Gartenstraße 18, D-35469, Allendorf, Germany

[3] Dipl.-Wi.-Ing., Co-ordinator of Market Area Solar Power Plants, Dept. Materials Research and Applied Optics, Fraunhofer-Institut für Solare Energiesysteme ISE, Heidenhofstraße 2, 79110 Freiburg, Germany

Abstract

THERMOFLEX is a commercially-available heat balance program for design, simulation, and cost estimation of power and heat facilities. It includes a number of features to model solar thermal power and heating cycles. It provides design point heat balance, physical equipment size, off-design performance, and cost estimates for parabolic trough collector (PTC) fields, linear Fresnel collector (LFC) fields, and related components. The collector model includes optical and geometric inputs to model existing and proposed reflector and receiver technologies, and computes heat transfer and pressure drop for both single-phase and multi-phase working fluids. Engineers involved in development and construction of solar thermal power and heating systems are the primary audience for these capabilities. This includes project developers, engineering firms, utility planning departments, and equipment manufacturers. The program allows creation of complete models, from solar irradiance to high voltage electric power, for pure solar thermal power plants, hybrid solar-fossil plants, and other solar heated cycles such as absorption chillers and thermal desalination. A design-point model of the SEGS VI plant is presented to benchmark program functionality. Other design-point and off-design models are presented to demonstrate how the software can be used to model various systems that utilize both conventional and novel concepts.

Keywords: heat balance software, solar thermal power, cost estimation.

1. Introduction

Development of a new solar power or heating facility requires efforts in many areas including legal, finance, accounting, and engineering. The engineering work addresses a broad spectrum of issues requiring expertise in environmental, civil, electrical, and mechanical disciplines. Mechanical engineers carry out three analytical steps where the key issues of concern are the plant's <u>performance</u>, <u>size</u>, <u>cost</u>, and its <u>economic viability</u>. Output from that analysis is essential to establishing fundamental plant design which sets the direction for the project. Step one is conceptual design where various technologies are evaluated. Step two is detailed design where short-list technologies are considered in light of site-specific constraints. Step three is simulated system operation where detailed plant design is used to estimate expected facility performance over a range of operating scenarios; potentially identifying ways to improve the design. These steps are repeated for each development opportunity and each set of site-specific conditions. Many more development plans are created on paper than are ever built.

A program for heat balance design, off-design simulation, and cost estimation that helps to quickly and effectively carry out these three key development steps is the focus of this paper. The solar field is featured in each model, along with turbines, heat exchangers, cooling systems, etc. Sample model scenarios are presented to demonstrate the scope of software utility. Specifically, four models are discussed:

- Design point Kramer Junction SEGS VI model is presented as a benchmark. Computed plant performance and size data match original SEGS VI data very well.
- Design point model of a solar thermal power plant using a linear Fresnel collector with direct steam generation is presented. Summary results from off-design simulation for a typical year are presented.

• Year long simulation of a gas turbine combined cycle and an integrated solar combined cycle is used to compare performance, cost, and economic viability.

• Conceptual gas turbine models are compared to quantify efficiency improvements possible with the use of solar heat addition.

2. Thermodynamic and Physical Model Features

The software has two basic modes: design and simulation. Design mode computes heat balance and establishes the physical size, weights, and dimensions for plant equipment. Simulation mode computes heat balance with fixed-size equipment at off-design operating conditions. Design mode inputs consist of (1) overall plant configuration (e.g. single reheat Rankine cycle with a dry cooling system, etc.), (2) specific equipment layout details (e.g. two low pressure feedwater heaters, one contact feedwater heater, and two high pressure feedwater heaters, etc.), (3) thermodynamic cycle constraints (e.g. main turbine admission pressure and temperature, reheat temperature, etc.), and (4) equipment characteristics (e.g. tubing details, material choices, inclusion or exclusion of optional features, etc.). Design mode output is a comprehensive heat, mass and material balance for the cycle together with a physical description of the designed equipment, and in some situations estimated installed cost.

In simulation mode, the plant equipment size, determined by the design calculation, is available for user editing. Simulation inputs are prevailing ambient conditions, desired equipment loading, current equipment availability and cleanliness. Off-design simulation output is a comprehensive heat, mass, and material balance, and in some situations estimated installed cost.

The system model is synthesized by the user who connects component icons together to create the flowsheet. At the user's discretion, the model may consist of a small subset of a complete plant, or it can be a comprehensive all-encompassing model of an entire facility. Over one hundred different component icons are available to model both common and novel thermal power cycles [1].

3. Solar Field Model Features

The solar field model can use parabolic troughs, or linear Fresnel collectors. The field can be a single flow path along a single collector row, or a large number of flow paths, each arranged in one or more collector rows. The field can heat single phase liquids or it can generate and superheat steam. The software includes properties for water in all its phases, and for twenty-five commercially available heat transfer fluids such as Therminol VP-1, DOWTHERM A, molten salt, etc. One-dimensional heat transfer and fluid flow calculations are performed step-wise along the receiver yielding distribution of fluid pressure, temperature, and heat loss, from inlet to exit and at intermediate points along the flow path. An alternative exists, mostly for use by solar field manufacturers, to directly specify the working fluid's exit state from the solar field based on calculations done using their proprietary codes, outside this software. The latter method provides full-freedom to model any solar field arrangement that produces a hot working fluid.

The software includes a physically-based thermo-optical model for parabolic collectors. This model allows the user to make reasonable assumptions about collector technology and create a collector model without need for detailed manufacturer data. This is useful for scoping studies. The software also includes a data-defined line-collector model that can be used to model parabolic troughs and linear Fresnel collectors. The data-defined model has options to select from a library of existing collectors. Alternatively, the user can specify the manufacturer-supplied thermo-optical properties directly. This feature is useful for modeling a particular collector available from a specific vendor.

For the data-defined collector, the design-point collector inputs describe the physical and optical characteristics of the collector and its receiver. This includes nominal optical efficiency, incident angle modifier transfer function table (IAM[θ_T, θ_L]), geometric cross-section data, receiver tube hydraulic parameters, receiver element heat loss characteristics, and desired field layout data.

Design-point collector outputs are presented for each flow path and for the field as a whole. Model outputs include mass flow rate, pressure drop, velocities, gross heat absorption, net heat absorption, heat loss, number of flow paths, number of collector rows, length of each row, number of collector row banks in field, field land area, and estimated field installed cost.

Off-design collector inputs include current irradiance data, adjustments to collector row and field geometry, optical efficiency reduction due to dirt and age, number of collector rows in-service, control strategies to limit fluid exit temperature and/or total field flowrate, and logic to shut down the field under low irradiance conditions. Parameters reported for off-design runs are the same as for design point runs.

Irradiance can be specified as (1) DNI with site longitude, latitude, time zone, day of year, and local time of day, (2) DNI with azimuth and zenith angles, (3) a theoretical sun-atmosphere model with latitude, time of day, day of year, and haze index, or (4) aperture normal irradiance (ANI).

4. Kramer Junction SEGS VI Plant Model

A design point model of the Kramer Junction SEGS VI plant was built to demonstrate functionality with a well-known facility. The "100% Solar Load" condition heat balance for the power block, and overall thermal performance of the solar field came from Lippke [2]. The field's total aperture area was provided in a report by Cable [3]. Other parameters, not readily available in the public domain, were assumed using reasonable engineering judgment. Figure 1 shows the design point heat balance result. Computed steam flows and generator gross power match published heat balance data to within 0.5%. Predicted Therminol VP-1 flow rate (410 kg/s) is 3.7% higher than the reference data. This difference is due to the fact that the specific heat used in the reference was 3.67% higher [2] than actual values quoted by fluid manufacturer, Solutia [4]. After correcting for specific heat difference, the predicted Therminol VP-1 flow rate matches the reference to within 0.5%. The solar field aperture, 188,323 m^2, was specified from the data. The land area, 63.5 hectare, was set to match estimates made using Google Earth.

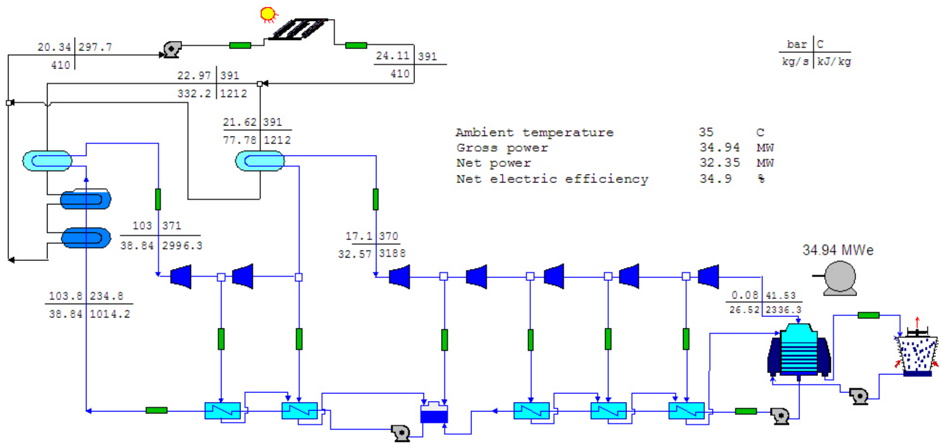


Figure 1 - Design point heat balance model of SEGS VI plant in Kramer Junction, CA.

5. Hybrid Solar-Fossil Rankine Cycle with Direct Steam Generation in Solar Field

The software can model heat transfer to receiver tubes carrying single phase heat transfer fluids, as well as water, steam, and two-phase water/steam mixtures. Direct Steam Generation (DSG) designs circulate water and steam in the solar field, thereby avoiding use of intermediate heat transfer fluid, associated heat exchangers, and related efficiency loss. This makes better thermodynamic use of the heated stream, and may reduce plant complexity and cost; albeit with the need to circulate high pressure water and steam throughout the solar field.

5.1. System Layout and Calculation of Annual Energy Yield

A design and simulation model of an 11 MWe (peak) power plant using Linear Fresnel Collector (LFC) technology with direct steam generation was built. The LFC uses optical characteristics and heat loss characteristics that are similar to a Novatec Biosol AG concept [5]. The power plant includes a natural-gas fired backup boiler, in parallel with the solar field, to generate steam when the field is unavailable due to maintenance, weather, or time-of-day. The backup boiler facilitates firm electric dispatch, without storage.

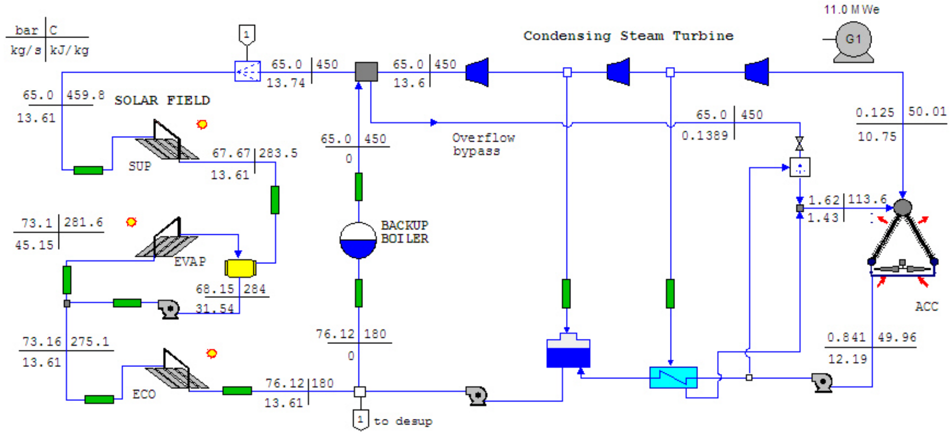


Figure 2 - Hybrid solar-fossil power plant schematic & design-point heat balance.

Figure 2 shows the model; a condensing steam turbine power plant with an air-cooled condenser (ACC), a low pressure feedwater heater, and a deaerator. Steam is directly generated in a Linear Fresnel Collector (LFC) solar field and/or by a gas-fired package boiler installed in parallel. The solar field consists of three sections, one to preheat water, one to evaporate water, and the final section to superheat steam. The evaporator is designed to produce 30% quality steam. A steam drum separates the phases; liquid recirculates to evaporator inlet, and dry steam flows to the superheater field. The solar field is sized to produce 13.6 kg/s at 65 bar, 460 C. Steam turbine generator makes 11 MWe at this condition with the ACC operating at 125 mbar in a 32 C ambient. This plant design minimizes plant makeup water requirements, consistent with desert-like site conditions present at many solar sites.

The steam cycle is small, does not include reheat and has few heaters. Therefore the base cycle efficiency is relatively low. However, this plant is also relatively simple, inexpensive, and easily capable of operation in full solar mode, full gas-fired mode, or in hybrid mode when some steam is generated in the field and the balance is provided by the fired boiler. So, it is flexible.

In this model, a steam flow controller maintains plant net power output between 8 and 11 MW, regardless of ambient condition. When conditions allow the solar field to generate minimum steam flow, the backup boiler is shutdown. When the field is incapable of generating minimum required steam, the backup boiler is loaded to makeup the difference. Excess steam is desuperheated and sent to condenser.

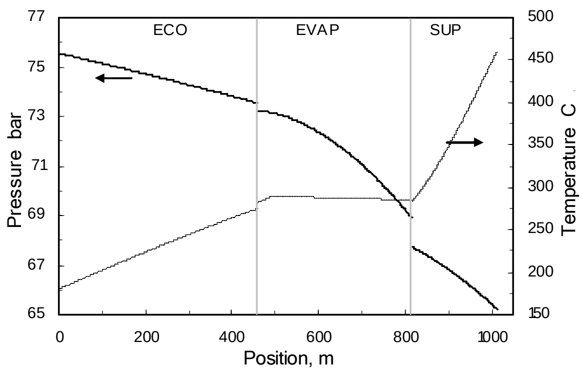
This model was used to simulate operation over a year using ambient and irradiance conditions typical of Daggett California, USA. The plant was run on a 24 hour schedule for 8000 hours per year. The annual average net LHV (lower heating value) efficiency was computed from the sums of net power produced and net fuel consumed; (GWhr electric export / GWhr LHV fuel consumption). Results of the yearly simulation show this relatively low efficiency steam cycle operates at 41% effective net LHV electric efficiency, a high value by Rankine cycle standards. This efficiency would be far higher if the plant shutdown overnight, and would be lower in locations with poorer solar characteristics. From a fuel consumption viewpoint, this plant has good solar leverage.

5.2. Solar Field Heat Transfer & Pressure Drop Model Details

Estimates for pressure gradient and heat transfer coefficient inside the receiver tube in a two-phase flow are more complicated than for single-phase situations. In this model, the tubes are assumed to be essentially horizontal, thereby simplifying the physics while still capturing most practical cases. The calculation of heat transfer to/from the receiver tube uses a one dimensional model where the flow path is discretized into a number of steps. The bulk fluid state and properties are assumed uniform in each step. The model estimates step-wise local values for internal heat transfer coefficient and pressure gradient. The heat transfer coefficient for single phase flows is computed using the familiar Dittus-Boelter equation. The single-phase pressure gradient is computed using the Darcy-Weisbach friction factor accounting for pipe length, roughness, and equivalent length of all fittings. The single phase results are modified using two-phase flow modifiers that depend strongly on flow conditions and properties [1]. Computation marches from tube inlet, where conditions are known, to its exit, computing intermediate states and updating properties along the way.

Figure 3 shows distribution of computed pressure, temperature, pressure gradient, heat transfer coefficient, mass flux, and bulk velocity from economizer inlet to superheater exit at design conditions. Each plot has three separate regions, corresponding to the distinct solar fields. The economizer field design produced three flow paths. The flow length of each path (including risers and returns) is 460 m. This field's total aperture is just over 15,000 m^2. The evaporator field design produced 14 flow paths. The flow length of each path is 355 m. The evaporator field's total aperture is 54,500 m^2. The superheater field design produced ten flow paths, each with total flow length of 195 m. The superheater field aperture is 20,700 m^2.

The pressure plot is discontinuous because of pressure losses in piping systems between fields. The temperature plot is discontinuous between economizer and evaporator because subcooled economizer exit water mixes with saturated liquid recirculated back from steam drum. Final steam temperature exceeds turbine inlet by 10 C, requiring use of desuperheater between field and turbine. The pressure gradient and heat transfer coefficient distributions are discontinuous in value because the mass flux in each field is different, to ensure reasonable velocities in each section. The slope of pressure gradient in evaporator is discontinuous because inlet water is slightly subcooled. The sharp discontinuity in value of heat transfer coefficient between evaporator exit where steam quality is 30%, and superheater inlet illustrates how dramatically this differs between wet low quality steam and dry vapor. The number of paths in each field is different, although the receiver tube diameters are the same throughout (70 mm OD). Therefore, the mass flux in each section is different, and the velocities are discontinuous at field boundaries. Velocity varies inversely with density along the flow path.



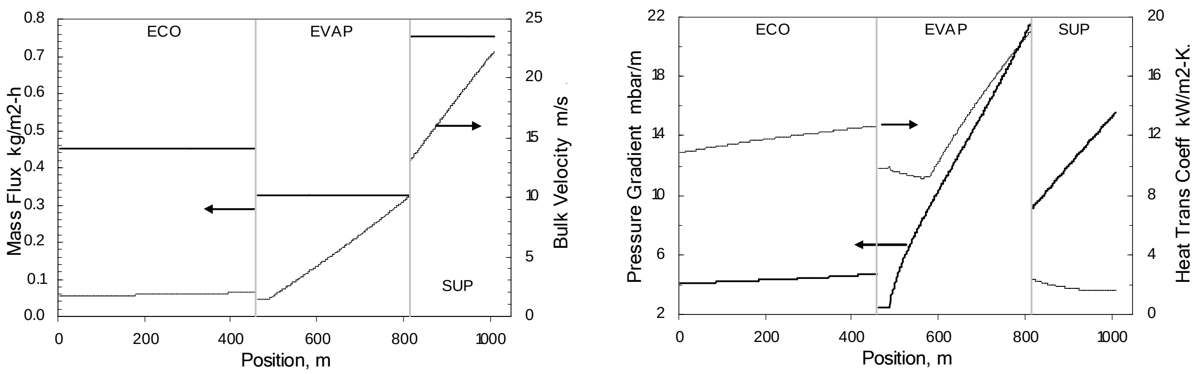


Figure 3 - Design-point distribution of pressure, temperature, pressure gradient, and heat transfer coefficient, mass flux and velocity throughout the economiser, evaporator, and superheater fields.

6. Integrated Solar Combined Cycle

Gas turbine combined cycles (GTCC) are commonly built to meet demand for new capacity. GTCCs are available in a wide size range, with moderate to high efficiency, and low emissions. They can be designed for site constraints related to ambient conditions, availability of water for makeup and cooling, fuels, etc.

Gas turbine combined cycle (GTCC) power plants combined with solar fields that directly or indirectly generate steam for the steam turbine are called Integrated Solar Combined Cycles (ISCC). Solar contribution can increase power without burning more fuel, or it can produce the same power whilst burning less fuel.

Two models were built to compare performance, cost, and gross operating profit. The concept here is to compare use of solar heat to boost power capacity, rather than save fuel. Base model is a 500 MW class plant with two GTs exhausting into two unfired three pressure reheat HRSGs feeding a single ST. Steam conditions are HP 105 bar / 565 C, IP/RHT 25 bar / 565 C, LP steam 4.2 bar / 295 C. Gas turbine inlet air is cooled with an evaporative cooler. The condenser is serviced by a wet cooling tower. The plant burns high pressure pipeline natural gas, preheated with IPE water to 185 C. Assumed location is San Bernardino County, CA, USA, proximate to Palmdale airport for which TMY3 data are available. This is nearby the permitted site for the Victorville 2 ISCC, and the plants analyzed here are similar to the proposed plant.

The alternate model uses the same base configuration but includes a parabolic trough solar field heating Therminol VP-1 which passes through a "solar boiler" to make partially superheated high pressure steam. The solar field sits on 101 hectare (250 acre) of land and consists of 310 collector rows (155 flow paths) with a total aperture of 344,342 m^2, oriented north-south. Solar field model automatically tracks the sun such that the central ray and trough axis are always coplanar.

The solar boiler gets water from HP feedpumps. Solar steam is combined with partially superheated HP steam in the HRSG, and the combined steam flow is further superheated for admission to the steam turbine. Condensate forwarding and HP feedpumps are oversized to handle extra flow from solar field operation.

These two off-design models were used to simulate 24 hour base load operation for one year by simulating 8760 cases. Hourly TMY3 data for site number 723820 (Palmdale Airport) provides local standard time, day of year, DNI, ambient pressure, temperature, and relative humidity for each hour. An automated procedure to run 8760 cases using ELINK, a bi-directional link between THERMOFLEX and MS Excel [1], was used to expedite the calculation process. Sums from the 8760 results were scaled to 8100 operating hours (92.5% availability) without prejudice to time of day or day of year. Table 1 shows summary results.

	Unit	GTCC	ISCC	ISCC-GTCC
Net Power	GWhr	3893	4015	122
Fuel Consumption (LHV)	GWhr	7059	7059	0
Effective Net Electric Capacity	MW	480.6	495.7	15.1
Effective Net Electric Efficiency (LHV)	%	55.2	56.9	1.7

Table 1. Summary comparison of year long simulations of GTCC versus ISCC.

During the best solar hour of the year, the ISCC makes 55.6 MW more power than the GTCC, and has an LHV efficiency of 62%, almost 7% points higher than the GTCC. However, averaged over 24 hour-per-day operation for the entire year, the differences are more modest as shown in the last column of Table 1.

The ISCC plant layout is the same as the GTCC, but modifications were made to accomodate solar generated steam. The ISCC steam turbine, generator, and transformer are 50 MW larger than for the GTCC; 235 MW versus 175 MW. The HP feedpumps and condensate forwarding pumps are also oversized to accomodate additional flow to/from solar boiler. Companion software programs, GT PRO and PEACE [6], were used to estimate installed owner's costs for these two combined cycles, without solar field and related appertances. These cost results are summarized in Table 2. Results indicate the incremental cost for the larger steam turbine equipment and pumps is relatively small, only about 3% of base GTCC cost. This means the extra 50 MW of capacity is purchased for about 240 US$/kW, a consequence of marginal economy of scale.

California site location, 2009 dollars	Unit	GTCC	ISCC	ISCC-GTCC
Installed Combined Cycle Cost	MM US$	446	458	12
Installed Solar System Cost Adder	MM US$	-	155	155
Installed Total Plant Cost	MM US$	446	613	167

Table 2. Owner's estimated installed plant costs for GTCC and ISCC, millions US$.

THERMOFLEX was used to estimate installed solar system cost adder which includes solar field, heat transfer fluid, solar boiler, pumps, pipes, etc. Estimated cost at California site is 450 US$/$m^2$ (aperture).

On a yearlong basis, the solar field contributes about 15 MW additional generating capacity for a 24-hour dispatch cycle. This additional 15 MW capacity has an overall specific capital cost of 11,135 US$/kW. Over the course of the year, the GTCC and ISCC consume the same amount of fuel, but the ISCC makes more power. Based on pricing for this region from 2001 to 2006 the average peak power price in 2006 dollars was 65 US$/MWhr. In today's terms that's about 75 US$/MWhr. Using this power price, the ISCC generates an incremental revenue of 9 MM US$ per year. This translates to about 19 years to repay the incremental capital cost of the ISCC versus the GTCC. This analysis is crude. A more refined analysis would include a price premium for solar power, other renewable incentives, investment tax credits, and perhaps CO2 reduction credits, etc. Nonetheless, until fuel prices rise to reflect fuel's intrinsic value, and the price of power naturally follows, the economics of extra power produced or fuel savings *alone*, cannot foster ISCC development.

7. Solar Assisted Gas Turbine Cycle

SOLGATE investigated use of a "solar combustor" to augment or supplant a conventional fuel-burning gas turbine combustor [7]. This idea exploits the ability of concentrating collectors to generate large heat fluxes. It also uses captured heat at high temperature, which provides a strong thermodynamic advantage over other solar thermal power systems. Compared with solar-heated Rankine cycles, these advantages translate into higher efficiency and power density, and/or lower system cost. Assuming (mostly material-related) technical challenges are surmounted; the result could be reasonably sized dispatchable solar-fossil plants.

The software was used to analyze a different modification to a gas turbine cycle, with some of the same goals as the SOLGATE project. Here a GE LMS100 gas turbine was modified by using hot fluid from a solar field to preheat compressor discharge air before combustion. This modification will save fuel when the solar system is available, and has inherent backup via fossil fuel when the solar system is unavailable.

The GE LMS100 gas turbine is a high pressure ratio three-spool aeroderivative with compressor intercooling to improve efficiency. This engine's gross LHV efficiency is 44.5% at ISO ambient conditions burning natural gas with no water injection. This is the highest efficiency gas turbine available today, and is targeted at peaking duty, often coincident with best solar conditions. This efficiency is about 2.5% points higher than non-intercooled aeroderivatives (GE LM6000, RR TRENT), about 5% points higher than advanced steam cooled industrial engines (MHI 501G, Siemens SGT6-6000G), and about 6.5 to 7.5% points above F-class industrial machines from Alstom, GE, Mitsubishi, and Siemens.

Adding a solar system to preheat high pressure combustion air can raise the efficiency beyond 44.5%. The LMS100 has a high pressure ratio (37:1), so solar heat is added to dense air thereby minimizing the size and cost of the preheater. While the air preheater cost is difficult to estimate, it is likely only a small fraction of the cost of a solar boiler together with dedicated steam turbine and cooling system.

The software was used to make a design-point model of the LMS100 engine that was tuned to reproduce published performance from the manufacturer. The process of building and tuning the base model established assumptions about the technology such as compressor and expander efficiencies, cycle pressures and temperatures, and maximum metal temperatures. These reverse-engineered assumptions were then used to build several proposed designs, all using the same core engine assumptions (i.e. technology level).

	Unit	**Base**	**Mod 1**	**Mod 2**
Gross (Generator) Power	MW	99	105.4	103.9
Gross (Generator) LHV Efficiency	%	44.5	46.9	54.4
Net (Plant) Power	MW	97.9	103.8	102.4
Net (Plant) LHV Efficiency	%	44.0	46.2	53.6
Final compressor exit temperature	C	370.5	144.4	144.4

Air preheater exit temperature	C	-	387.8	551.7
Firing temperature	C	1399	1399	1399
Land area for solar field	hectare	-	38	86
Total collector aperture area	m^2	-	136,345	338,647
Net collector efficiency	%	-	45.2	34.5
Estimated field installed cost (CA, USA - 400 US$/m^2)	MM USD$	-	55	136

Table 3. LMS100-like plant models operating at ISO ambient and 700 W/m^2 aperture irradiance.

Table 3 illustrates salient points for each model run at ISO ambient conditions, with irradiance normal to aperture of 700 W/m^2. **Base** model is the LMS100 as available from GE. An open cycle GE LMS100 requires a site of 1.5 to 2 hectare. Installed plant cost is about 94 MM US$ in California, USA. **Mod 1** includes another intercooler to further reduce compressor work and compressor exit temperature. It also includes a parabolic trough solar field heating Therminol VP-1 to 399 C. The heat from solar field is used to preheat compressor discharge air to 388 C, about 20% of the air heating needed to reach rated firing temperature. Fuel savings and added power (from reduced expander cooling load) increases gross efficiency by 2.4% points, and net efficiency by 2.2% points after including auxiliaries for intercooler and solar field. **Mod 2** is similar to **Mod 1** except that the field uses molten salt (60% $NaNO_3$ + 40% KNO_3 by weight). Salt can operate at up to 590 C and can therefore preheat the air to higher temperatures than possible with Therminol. In **Mod 2**, solar field heats salt to 565 C and the air is preheated to 552 C, about 32.5% of the heating needed to reach rated firing temperature. The corresponding increase in net LHV efficiency is 9.6% points. However, the higher operating temperature of this field means significantly higher thermal losses and lower collector efficiency, leading to a larger and more expensive field. The solar salt field is about 2.5 times larger than the Therminol field, and the solar salt must be kept warm overnight; two significant drawbacks.

A quick marginal economic analysis at today's power and fuel prices in California, USA indicates the direct payback period for these modified designs is in the 40 to 60 year range. That means based on incremental revenues from extra power sales and saved fuel, these modifications would not pay for themselves without other revenue sources.

8. Conclusion

THERMOFLEX is a program that can be used for heat balance design, simulation, and cost estimation of power and heat facilities that include solar thermal fields and related equipment. It can model existing and planned facilities using current technologies, and also novel systems based on future technologies. The software is useful to solar equipment suppliers, EPCs, investors, and others involved in the engineering and analysis of solar thermal power and heat cycles. The program provides its users with modeling capabilities that yield valuable insight into the technical and economic tradeoffs inherent in every project.

References

[1] Thermoflow, Inc., "THERMOFLEX User Manual, Version 19", 2009.

[2] Lippke, F., "Simulation of the Part-Load Behavior of a 30 MWe SEGS Plant", Sandia National Laboratories Technical Report SAND95-1293.

[3] Cable, R., "Solar Trough Generation – The California Experience", presented at ASES Forum 2001, Washington D.C.

[4] Solutia, Inc., "THERMINOL Heat Transfer Fluids by Solutia, A Selection Guide", 2001

[5] Morin, G., "Technical and Economic Models for Parabolic Trough Collectors and Linear Fresnel Collectors", MAO1-GM-0803-E02, Fraunhofer ISE, Prepared for Thermoflow, Inc., 2008.

[6] Thermoflow, Inc., "GT PRO / PEACE User Manual, Version 19", 2009.

[7] European Commission Directorate-General for Research, "SOLGATE Final Publishable Report, EUR 21615", 2005.