Cost-effective and reliable design of a solar thermal power plant

Article *in* Transactions- Canadian Society for Mechanical Engineering · January 2009

Impact Factor: 0.19

Amir A. Aliabadi
Massachusetts Institute of Technology

COST-EFFECTIVE AND RELIABLE DESIGN OF A SOLAR THERMAL POWER PLANT

A. A. Aliabadi and J. S. Wallace*
Department of Mechanical Engineering, University of British Columbia*
Department of Mechanical & Industrial Engineering, University of Toronto
2054-2324 Main Mall, Vancouver, BC, V6T 1Z4, Canada *
5 Kings College Road, Toronto, Ontario, M5S 3G8, Canada
Tel: 001-604-827-4273*, Fax: 001-416-978-7753*
E-mail: aliabadi@interchange.ubc.ca*, E-mail: wallace@mie.utoronto.ca

Received January 2009, Accepted March 2009
No. 09-CSME-08, E.I.C. Accession 3094

ABSTRACT

A design study was conducted to evaluate the cost-effectiveness of solar thermal power generation in a 50 kWe power plant that could be used in a remote location. The system combines a solar collector-thermal storage system utilizing a heat transfer fluid and a simple Rankine cycle power generator utilizing R123 refrigerant. Evacuated tube solar collectors heat mineral oil and supply it to a thermal storage tank. A mineral oil to refrigerant heat exchanger generates superheated refrigerant vapor, which drives a radial turbogenerator. Supplemental natural gas firing maintains a constant thermal storage temperature irregardless of solar conditions enabling the system to produce a constant 50 kWe output. A simulation was carried out to predict the performance of the system in the hottest summer day and the coldest winter day for southern California solar conditions. A rigorous economic analysis was conducted. The system offers advantages over advanced solar thermal power plants by implementing simple fixed evacuated tube collectors, which are less prone to damage in harsh desert environment. Also, backed up by fossil fuel power generation, it is possible to obtain continued operation even during low insolation sky conditions and at night, a feature that stand-alone PV systems do not offer.

UNE CONCEPTION RENTABLE ET FIABLE D'UNE CENTRALE THERMIQUE SOLAIRE

RÉSUMÉ

Nous avons entrepris une étude pour évaluer la rentabilité d'un générateur d'énergie thermique solaire dans une centrale de 50 kWe, qui pourrait être utilisée en des lieux isolés. Le système est une combinaison d'un système de stockage par collecteur solaire thermique utilisant un fluide caloporteur et un simple générateur d'énergie de cycle Rankine utilisant un réfrigérant R123. Les collecteurs solaires à tubes sous vide chauffent l'huile minérale et la met en réserve dans le réservoir de stockage thermique. Un échangeur de chaleur à l'huile minérale comme réfrigérant génère une vapeur surchauffée frigorigène, laquelle entraîne un radiateur turbo-générateur. Un chauffage supplémentaire au gaz naturel maintient une température de stockage thermique constante, quelque soit les conditions de rayonnement solaire, permettant ainsi au système d'atteindre une production constante de 50 kWe. Une simulation a permis de prévoir la performance du système par une très chaude journée d'été, et par une journée très froide d'hiver dans les conditions de rayonnement solaire du sud de la Californie. Une analyse économique rigoureuse a été menée. Le système offre des avantages sur la centrale thermique solaire de technologie avancée en implantant de simples collecteurs solaires à tube sous vide fixes, lesquels subissent moins de dommages reliés à l'environnement aride du désert. De plus, appuyé par un système de centrale thermique à combustible fossile, il est possible d'atteindre une opération en continu, même en période de faible ensoleillement et la nuit, une caractéristique que les systèmes autonomes PV/thermiques ne peuvent offrir.

1. Introduction

Solar thermal power plants have been in commercial operation since 1985 in the Mojave Desert region of California and with little fanfare have proven to be reliable and cost effective. Nine plants, named SEGS I to IX, have been built with total power output capacity of 350 MWe. These Rankine cycle plants have the capability of continuing operation during non-daylight hours fueled by natural gas.

Smaller power units – kWe rather than MWe sizes – are needed for remote power. Photovoltaic devices are typically considered for this type of application, but solar dishes with Stirling engines are also being developed. The success of the SEGS-type system warrants a closer look for lower power applications as well.

The study reported here explores the feasibility of designing a simple 50 kWe solar thermal power unit for remote power. Use of fixed, non-focusing solar collectors together with a basic Rankine cycle result in lower initial cost but reduced efficiency. The study explores the trade-off between initial cost and performance. Southern California is chosen for the study as representative of a high solar insolation site.

2. The Power Plant Cycle

The solar thermal system, figure 1, consists of a collector-storage loop (low temperature storage tank to solar collector to high temperature storage tank to boiler and back to the low temperature storage tank) and a simple Rankine cycle loop (boiler to turbine-generator to condenser to feed pump and back to the boiler).

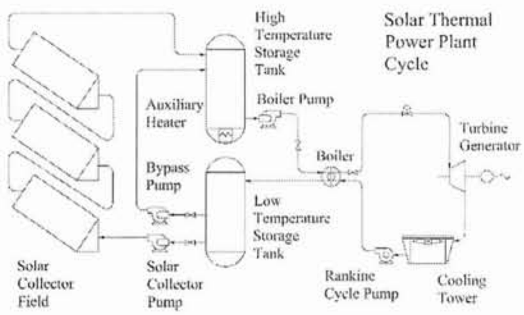

Fig. 1. Solar thermal power plant cycle

In the collector-storage loop, a heat transfer fluid is pumped through the solar collectors, heated to a high temperature, and then stored in an insulated high temperature storage tank, which is also equipped with an auxiliary natural gas burner for temperature control of the tank. The auxiliary gas burner makes up the difference between the collected solar energy and the heat input needed for the refrigerant loop Rankine cycle to produce 50 kWe. Heat transfer fluid discharged from the boiler is collected in the low temperature storage tank to be pumped through the solar collector again. Refrigerant in the Rankine cycle is evaporated in the boiler, expanded through the turbine, condensed, and then pumped back to the boiler.

2.1. Working fluid selection - The Rankine cycle working fluid selection criteria are flammability, toxicity, Ozone Depletion Potential (ODP), Global Warming Potential (GWP), consideration of cycle temperatures and pressures, and the 1st and 2nd law efficiencies. R123 is chosen over all other candidates considered because it is a good thermodynamic match to the operating cycle temperatures, poses no flammability or toxicity hazard, and has acceptable ODP and GWP potentials.

Mineral oil is selected for effective heat transfer of solar radiation from the collector to the cycle boiler. This oil is well below its boiling point at the designed 160 °C solar collector output temperature, which allows the collector to operate at atmospheric pressure. This simplifies collector design and enhances safety.

2.2. System Components - Table 1 presents the three major categories of solar collectors that are widely used. Note the estimated temperature ranges and combined collector and Rankine cycle efficiencies [1].

Table 1. Solar collector types

Temperature Range [°C]	Collector Type	Rankine Cycle Efficiency [%]
90-120	Flat Plate	5
150-200	Evacuated Tube	10
>315	Tracking Concentrator	>11

As a compromise between efficiency and overall costs for this low power system, an evacuated tube collector, able to raise the temperature of the circulating heat transfer media up to 150-200 °C, is chosen. In this type of collector, figure 2, the absorber is inside an evacuated glass tube to reduce convective heat losses compared to flat plate collectors.

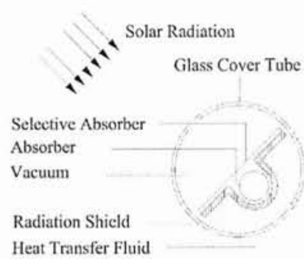

Fig. 2. Solar collector schematic

The present design concept employs rows of evacuated tube collectors, permanently fixed at the latitude angle and oriented to the south just as is done for flat plate collectors. Figure 3 shows an artist's conception of this arrangement, minus the connecting piping.

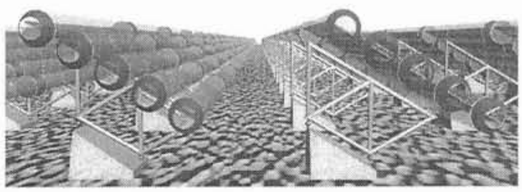

Fig. 3. Solar collector arrangement

A simple unitary boiler design is a key to cost effectiveness of the system. While no suitable small boiler is commercially available, we propose adapting a vapor generator design shown in figure 4 originally developed by Babcock and Wilcox for nuclear power plants [2]. The heat transfer fluid from the high temperature storage tank passes through the tubes of the shell and tube heat exchanger. The refrigerant to be evaporated enters the shell side and is confined to the lower section of the shell by internal baffles. The refrigerant vapor is superheated by passing along the shell side of the upper part of the tubes, where the incoming oil is hottest.

This boiler is the only custom component of the power plant, but it employs conventional heat exchanger technology so no fabrication difficulty is anticipated.

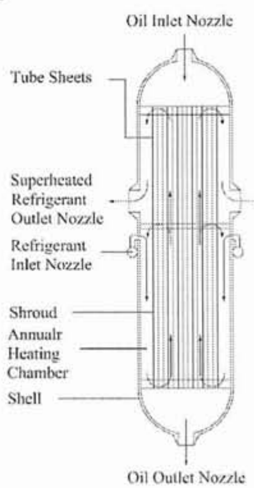

Fig. 4. Boiler schematic [2]

Radial turbines are recommended for low power and low head applications and also for refrigerant Rankine cycles [3]. Due to the finite number of turbines available on the market, a specific turbine in the desired power range is chosen, and then all other system components are selected to match. A Barber Nichol 50 kWe turbogenerator for R123 duty is chosen. The unit combines a radial turbine and an integral 3 phase-60 Hz induction generator. With 1,300 kPa and 140 °C turbine inlet conditions, the turbogenerator fluid to electricity efficiency is about 71.5 % [4].

Dry and evaporative condensers are each considered but the high parasitic fan power requirement for dry cooling eliminates that possibility. Evaporative condensers need much less power and are far more compact but do require some make-up water. Evaporative cooling can provide hot water for effective waste heat recovery. Waste heat recovery potentially improves the economics of the system but is not taken into account in this study. A Baltimore Aircoil model CXV-119 designed for use with refrigerants was selected.

Most solar thermal systems, e.g. the SEGS plants in California and the Plataforma Solar de Almeria in Spain, incorporate thermal storage of energy. Thermal storage "...provides an output management tool to prolong operation after sunset, to shift energy sales from low revenue off-peak hours to high revenue peak demand hours, and to contribute to guaranteed output" [5]. Also, it provides buffer capacity to level changes in solar insolation [5]. The present design is based on a dual storage tank system in which the hot liquid is stored in a high temperature tank, and the cold liquid is stored in a low temperature tank. The high temperature tank incorporates a supplemental natural gas fired heater. The tanks were sized to enable the solar plant to be able to operate 24 hours per day during the summer. Cloud cover requires some supplemental natural gas burning and more is required during the winter when there are fewer hours of insolation. The simulation results will illustrate this point.

3. Solar Thermal Power Plant Operation

In order to produce a constant nominal 50 kWe power output, the Rankine cycle system is designed to run in steady state condition 24 hours a day. By means of the variable heat transfer fluid mass flow rate and the thermal storage, the system buffers the heat input to the Rankine cycle so that constant heat input is provided for steady state operation of the turbo-generator set.

There are three system control parameters: the mass flow rate of heat transfer fluid through the solar collector, the temperature in the hot tank (which in turn controls the auxiliary natural gas burner), and the exit liquid refrigerant temperature (which in turn controls the variable speed cooling fan in the condenser).

4. Technical Analysis

Thermodynamic analysis was performed to find the Rankine cycle efficiencies possible for this design. R123 has a boiling point of 27.9 °C at 1 atm, which makes it ideal for low pressure Rankine cycles. Engineering Equation Solver (EES) is used to model a simple ideal Rankine cycle. As seen in figure 5, efficiencies competitive with PV solar cells are attainable using R123 as the working fluid. The condenser temperature and pressure are considered to be 40 °C and 200 kPa respectively. The high temperature of the Rankine cycle is considered to be 140 °C. Note that the pressure was varied from 400 kPa to 1,300 kPa. Since both 1st law and 2nd law efficiencies increase with pressure, it is desired to run the Rankine cycle at the highest pressure possible (i.e. the maximum turbine inlet pressure). With an inlet pressure of 1,300 kPa, a theoretical 1st law efficiency of 17 % and a theoretical 2nd law efficiency of 70 % are possible. However, due to various losses and turbine-generator inefficiencies mentioned earlier, an overall efficiency of 10 % from boiler heat input to electric power is assumed for the simulation analysis.

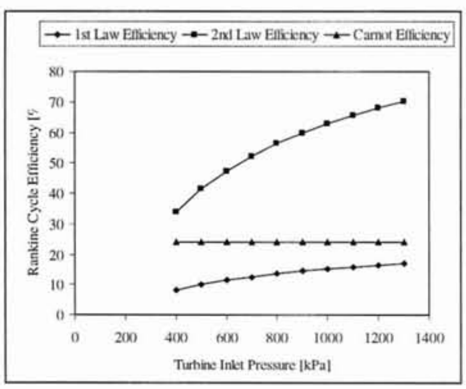

Fig. 5. R123 ideal Rankine cycle efficiency, 2nd law efficiency, and corresponding Carnot cycle efficiency

Figure 6 shows the states for the two working fluids. Tables 2 and 3 show the numerical state variables.

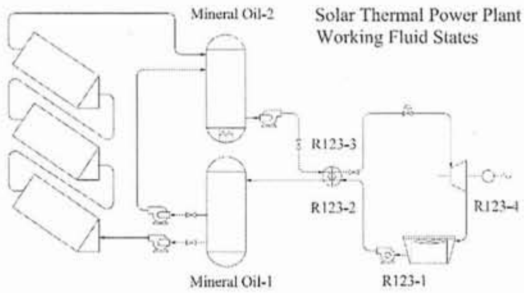

Fig. 6. Working fluid states

Table 2. Thermodynamic states for R123

State	T [°C]	P [kPa]	h [kJ/kg]	s [kJ/kg]	v [m^3/kg]
R123-1	40	154.7	241.9	1.142	0.0007019
R123-2	40.44	1300	242.7	1.142	0.0007005
R123-3	140	1300	471.1	1.744	0.0137
R123-4	72.21	154.7	431.4	1.744	0.1165

Table 3. Thermodynamic states for mineral Oil

State	T [°C]	P [kPa]
Mineral oil-1	60	Ambient
Mineral oil-2	160	Ambient

Simulink is used to predict the performance of the proposed solar thermal power plant in the summer and the winter. A variety of input parameters are required to perform the simulation by defining the system. Figure 7 shows the input and output (in brackets) parameters. The values for many inputs reflect design decisions already described.

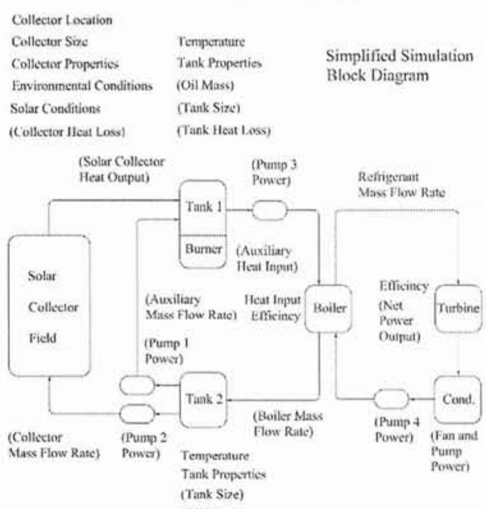

Fig. 7. Simplified simulation block diagram

The methodology is to predict the performance in the extreme conditions in terms of the solar insolation availability. Simulations are performed for the n^{th} day of the year, where n=1 corresponds to January 1^{st}. In the analysis n=200 and n=15 are used, and they represent the highest and lowest solar insolation during the year respectively. The Latitude, L, for southern California is 32.5 degrees. As described earlier, the tilt angle, β, is chosen to be 32.5 degrees as well. The equations used for the model are based on the methodology of Goswami, Kreith, and Kreider [6], summarized as follows. The solar declination angle, δ_s, in degrees, can be written as

$$\delta_s = 23.45 \sin\left[360(284+n)/365\right] \qquad (1)$$

Using this parameter, the solar hour angle, h_s, the solar altitude angle, α, the solar azimuth angle, α_S, and the incident angle, i, can be all calculated, in degrees, as

$$h_s = \text{Minutes from solar noon } / 4 \qquad (2)$$

$$\alpha = \sin^{-1}\left[\sin L \sin\delta_s + \cos L \cos\delta_s \cos h_s\right] \qquad (3)$$

$$\alpha_s = \sin^{-1}\left[\cos\delta_s \sin h_s / \cos\alpha\right] \qquad (4)$$

$$i = \cos^{-1}\left[\cos\alpha\cos(\alpha_s - \alpha)\sin\beta + \sin\alpha\cos\beta\right] \qquad (5)$$

Various sky optical properties were considered. The sky clearness, C_n, is assumed to be 1 for all practical purposes. Sky diffusive factor, C, varies from day to day, and is assumed to be 0.14 for n=200 and 0.06 for n=15. The sky optical depth, k, is assumed to be 0.21 for n=200 and 0.14 for n=15 [6].

To determine the solar insolation incident on the collector, I_C, various calculations must be performed. Solar irradiation reaching the earth's surface, I_{bn}, can be calculated using the solar constant, I_0, and the solar irradiation on top of the atmosphere, I, for any given n. These parameters can be calculated, in W/m^2 as

$$I = I_0 \times (1 + 0.034\cos(365n/365.65) \qquad (6)$$

$$I_{bn} = C_n I e^{-k/\sin\alpha} \qquad (7)$$

Next, the horizontal, direct, diffuse, and reflective components of the solar irradiation, I_h, I_{bc}, I_{dc}, I_{rc}, respectively, need to be calculated. The incident angle, the tilt angle, and the desert ground reflectivity, ρ=0.2, are required for these calculations. Finally, the solar insolation reaching the collector, I_C, is the sum of the three direct, diffuse, and reflective components. In W/m^2, the equations can be shown as

$$I_h = C_n I\left[e^{-k/\sin\alpha}\right]\left[C + \sin\alpha\right] \qquad (8)$$

$$I_{bc} = I_{b,n}\cos i \qquad (9)$$

$$I_{dc} = C I_{b,n}(1+\cos\beta)/2 \qquad (10)$$

$$I_{rc} = \rho I_h (1-\cos\beta)/2 \qquad (11)$$

$$I_C = I_{bc} + I_{dc} + I_{rc} \qquad (12)$$

The heat input to the collector is expressed using various parameters outlined before. In addition, the collector constant, (ατ)=0.85, collector area, A_C=2000 m^2, average collector heat transfer coefficient, U=0.9 W/m^2K, collector pipe diameter, d_C=0.1 m, collector accumulated length, L_C, average absorber temperature, T_{abs}=110 °C, collector and glass temperature difference, ΔT_C, ambient temperature, T_{amb}, and overall collector emittance, ε=0.09 are used. A first law energy balance would give the heat input to the collector, in W, as

$$\dot{Q}_C = (\alpha\tau) I_C A_C - U\pi d_C L_C (T_{Abs} - \Delta T_c - T_{Amb}) \qquad (13)$$
$$- \varepsilon\sigma A_C((T_{Abs}+273)^4 - (T_{Amb}+273)^4)$$

The most significant parasitic loss in the system is associated with pumping the mineral oil through the solar collector. The oil is highly viscous and turbulent (Re≈3000). For this flow, the Weisbach formulation applies, and gives a good estimate for the pressure differential needed across the collector to drive the flow [7]. Using the mineral oil mass flow rate, collector length and diameter, an approximate pipe friction factor can be read from the Moody chart. The pressure drop, in kPa, can be calculated as

$$P_1 = \frac{0.001 \times 8 \times 0.02 L_C \dot{m}_C^{\,2}}{\pi^2 d^5{}_C \times 1005} \qquad (14)$$

Finally, it is possible to calculate the net power output of the system. Condenser and other pump losses (P_C, P_{P1}, P_{P2}, P_{P3}, and P_{P4}) can be found using empirical equations. As described before, a constant heating power is provided to the Rankine cycle by the boiler. The power output of the system is this constant heating power times the Rankine and boiler efficiencies (η_R=0.1 and η_B=0.95). The net power output from the system, in W, is written as

$$P = \eta_R \eta_B \dot{Q}_R - P_{P1} - P_{P2} - P_{P3} - P_{P4} - P_C \quad (15)$$

Other parameters and internal variables formulated such as the mass flow rates, natural gas consumption, etc… are not shown in this paper. Figure 8 shows the summer and winter ambient temperature profiles obtained from meteorological data [8].

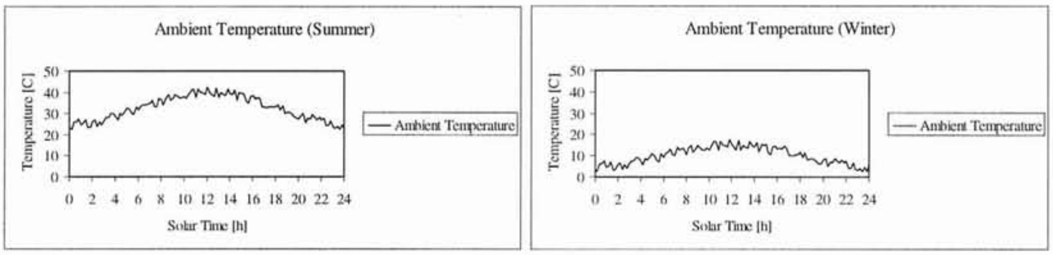

Fig. 8. Seasonal ambient temperatures [8]

Figure 9 displays summer and winter collector storage loop results. A random noise generator was used in Simulink to model cloud cover fluctuations (the difference between the ideal and actual solar collector output shown). are shown. In the summer, natural gas firing ("auxiliary heat input") is only required to make up for cloud cover. In the winter, natural gas firing provides a steady input that reflects the difference between winter and summer insolation levels in addition to the cloud cover losses.

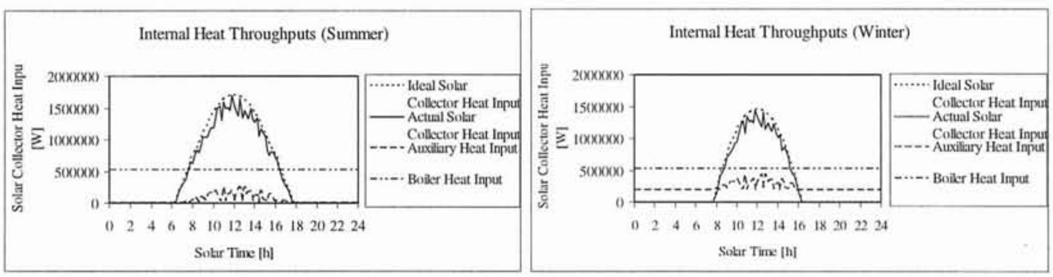

Fig. 9. Solar heat collection, auxiliary heating requirement and boiler heat input

Figure 10 shows the diurnal variation in the mass flow of heat transfer fluid through the solar collector. A preprogrammed mass flow rate schedule is followed to achieve the desired solar collector output temperature (160 °C) despite varying solar insolation.

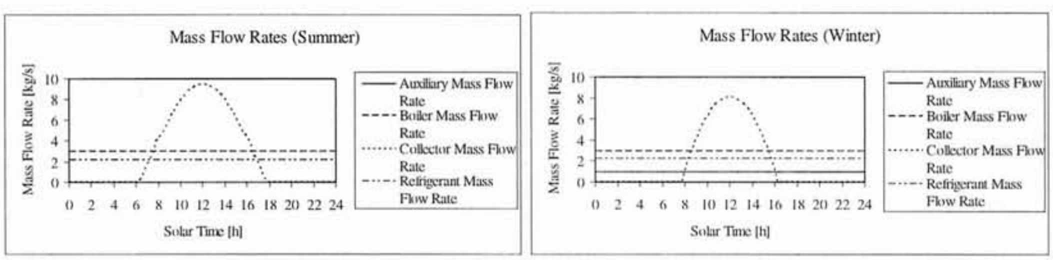

Fig. 10. Mass flow rates

Figure 11 shows pump and condenser fan parasitic losses for the two seasons. A higher heat transfer fluid flow rate is required during the summer to collect the increased solar energy; similarly greater fan power is necessary to achieve the same condenser temperature. These parasitic losses have a significant effect on the net system daytime output as shown in figure 12.

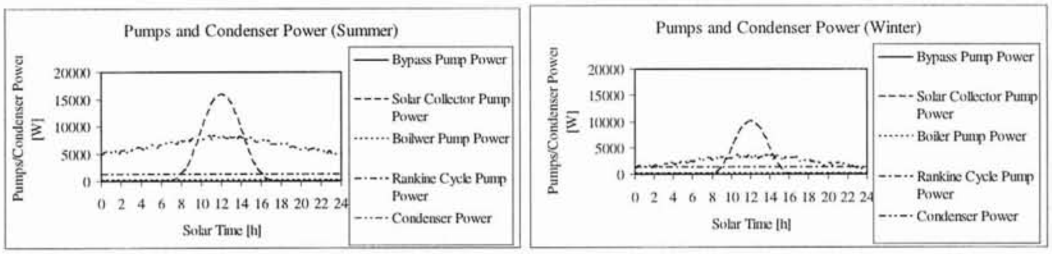

Fig. 11. Parasitic losses for pumps and fans

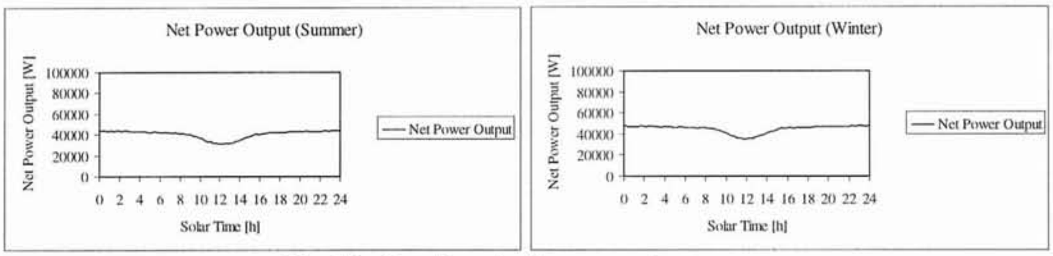

Fig. 12. Net diurnal solar power plant output

5. Economic Analysis

Although solar radiation is free, the equipment required to produce electricity is expensive. It is rarely cost-effective to supply all the energy requirements from the sun. Sizing a solar collector system is always a difficult choice – if the system is sized to meet the energy demand for the worst set of conditions – inclement weather under winter insolation – it would be oversized for all less severe conditions and operation below the maximum capacity leads to low efficiency. The oversizing problem is eliminated in this system by incorporating a gas-fired auxiliary burner into the high temperature storage tank. The solar collector is sized to provide the heat input to the boiler necessary to produce 50 kWe over a 24 hours period under ideal summer insolation. During a non-clear sky period or during reduced winter solar insolation, the auxiliary heater supplements the necessary heat input to the system.

A solar thermal system is fundamentally different from a conventional fossil fuel or electrical energy system such that it requires an economic analysis reflecting the benefits accrued by solar usage during the entire life time of the system. The concept of life cycle costing considers both the initial capital and the year-to-year operating costs in making economic decisions [6].

The optimization of a solar energy system is achieved when all components are sized to provide the least-cost mix of solar energy and conventional fuel for the application. For example, a critical question to answer is "*what proportion should the solar heating be in relation to the auxiliary heating?*" One common method to optimize the component sizing of the plant is to use the "*annualized solar cost minimization*" method [6]. In this method, the total cost of solar system on an annual basis is calculated. The annualized cost of installation, operation and maintenance of the proposed solar thermal system is estimated. These include the capital for the equipment, the labor, the taxes, the insurance, and the fuel costs. Table 4 shows the financial rates assumed in the analysis.

Table 4. Assumed financial rates

Assumptions	Value
Energy of Natural Gas [MJ/m^3]	37.20
Interest Rate of Borrowing Money [%]	6
Inflation Rate [%]	3
Number of Years to Study [yr]	15
Renewable Technology Investment Credit [%]	15
Income Tax Bracket [%]	30

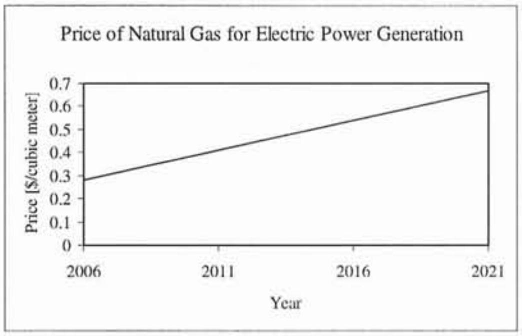

Fig. 13. Projected US natural gas price [9]

Table 5 summarizes the estimated costs for the system construction and operation. The largest initial cost is associated with the solar collector; yet, the largest operating cost is associated with the natural gas consumption. The cost of evacuated tube collector has been estimated to be \$300 /m^2 [10]. Natural gas prices used in the cost analysis are shown in figure 13. These prices account for the historical increase over the last 15 years [9].Due to the importance of these costly items a sensitivity analysis was conducted in which the cost of the solar collector and natural gas was varied by ±15 %. The annualized system cost varied by ±1 % for the solar collector and ±5% for the natural gas. Thus, the strongest dependence is associated with the future natural gas prices.

Figure 14 shows that the annualized cost falls as collector area increases up to 2000 m^2, an area that provides full system heat requirements under ideal summer sky conditions. The remaining input energy required to produce 50 kWe is provided by the natural gas fired auxiliary heater. Above 2000 m^2, additional collector costs, and increased thermal storage capacity, more than offset fuel savings. The fuel efficiency of the system is more than 45 % on an annual average basis, due to the solar contribution, whereas the typical efficiency of a simple Rankine cycle power plant is about 25 %. This means that this solar thermal power plant, in the right comparison, reduces greenhouse emissions by a factor of almost two.

Table 5. Estimated costs

Costs	
Solar Collector	$600,000
Collector Support Structure	$20,000
Thermal Storage Tank	$10,000
Synthetic Oil	$42,000
Auxiliary Heater	$1,000
Pumps	$21,500
Condenser	$15,000
Turbine Generator	$35,000
Pipes and Valves	$15,800
Boiler Heat exchanger	$17,500
Controls	$9,500
Wiring	$5,400
Labor	$120,000
Total (Present Worth)	$912,700
Maintenance [/yr]	$3,000
Operation Labor [/yr]	$33,000
Insurance [/yr]	$1,500
Total Fuel Cost (Present Worth)	$613,332
End of Life Value of the Plant (Future Worth)	($40,000)
Annualized Cost	**$139,554**

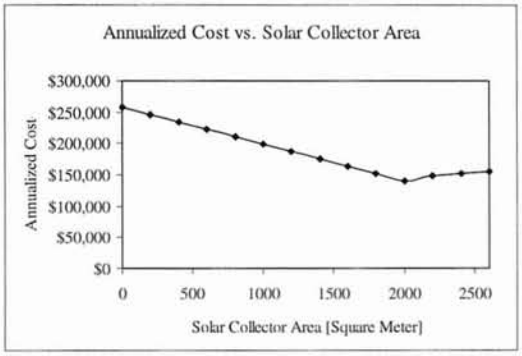

Fig. 14. Annualized system cost ($US)

6. Discussion

Despite the advantages, it goes without saying that the proposed solar thermal power plant poses special requirements on its installation site. As illustrated by Fig. 15 in the appendix, the power plant requires 1 hectare of desert land, primarily covered with solar collectors that preclude use of the land for other purposes (but that is not a problem for the usually unutilized desert land). An additional resource requirement is created by the evaporative condenser that consumes water. An evaporative rather than a dry condenser is necessary to achieve a heat rejection temperature not too much above ambient. Due to the low heat input temperature of the solar thermal plant the condenser temperature has a large impact on system efficiency. If the waste heat could be utilized – not considered in this analysis - the overall efficiency and therefore the economics of the project would be much improved.

Considering the amortization of the system for 15 years and assuming 350 days of full-time operation per year, the power plant is capable of providing electricity at $0.33 per kWh. For comparison, as Quaschning explains, the current solar thermal electricity prices are estimated to be €0.15 ($0.21) per kWh [11]. The cost of the larger system benefits from economy of scale and is a realistic number based on actual installations. However, the cost estimates in this paper are conservative projections for the installed cost of a unique prototype system. On the other hand, as opposed to tracking and focusing collectors, the fixed evacuated tube collectors operate more reliably, do not need frequent maintenance, and are less prone to damage in harsh desert environment.

A further comparison can be made with stand-alone photovoltaic systems. According to Van der Vleuten, PV system developments still mainly rely on government subsidies in the form of tariffs and only benefit from economic returns if considered for large scale (MW) installations [12]. This, often, results in high financial risks to be taken by governments and venture capitalists. In 2007, the IEA PVPS Task 8 report was presented in Athens, Greece, organized by European Photovoltaic Industry Association (EPIA). As calculated in this report in 2006, Beneking explains that stand-alone PV power costs €0.24-€0.38 ($0.33-$0.53) per kWh [12]. On the other hand many solar thermal power plants, including SEGS, are privately commercialized and generate power more economically. Also, stand-alone PV systems do not have the same functionality as the proposed solar thermal system, which produces power 24 hours a day.

In order to make a comparison with a PV system having the same functionality as the proposed solar thermal power system, it would be necessary to consider a hybrid PV system with battery storage and possibly a backup generator. A PV–diesel generator hybrid unit is under construction for a mountain hut in the Lambardia Alps. This system is required to produce a maximum of 28.8 kWh per day. The system has been optimized and economic calculations are performed for a 25 year amortization period. Under best scenarios, the cost of producing electricity for this system is projected as €0.31 ($0.43) per kWh [13]. The electricity cost produced by this hybrid PV system is higher than that of the proposed solar thermal power system. This is mainly because such hybrid systems need expensive battery storage in addition to the costly generators and PV panels.

7. Conclusions

The goal of this study was to propose a cost-effective and reliable solar thermal power plant. The power plant was designed for a small scale application which produces 50 kWe in the sunny climate of southern California. The design emphasis was to reduce cost by using simple technologies. The proposed power plant is a combined cycle binary working fluid system that operates on solar energy and natural gas. The main heat input to the system is from solar thermal energy collected by evacuated tubes. This energy is supplemented by a natural gas fired heater which provides necessary heat input when the solar energy is not available in abundance. A Rankine cycle converts the heat input to the system by a turbine generator set. Performance of the proposed system has been simulated and used in the economic analysis. Using life cycle costing analysis, the cost of producing electricity from a plant of this design is estimated at $0.33 /kWh.

Compared to advanced solar thermal and photovoltaic power plants, the proposed system offers cost savings and reliability. The proposed plant utilizes evacuated tube collectors that remain efficient even in slightly cloudy sky conditions. These collectors require less maintenance and operate more reliably than tracking concentrators in the harsh desert environment. The proposed system is base loaded by natural gas, so that it can provide power 24 hours a day, regardless of solar insolation availability.

Acknowledgement

The authors gratefully acknowledge the artistic work of Reza Aliabadi who created the magnificent 3D model of the proposed solar thermal power plant.

Nomenclature

A_C = collector area

C = sky diffusive factor

C_n = sky clearness

I = solar irradiation on top of atmosphere

I_0 = solar constant

I_{bc} = direct solar irradiation

I_{bn} = solar irradiation reaching the earth

I_C = solar insolation reaching collector

I_{dc} = diffuse solar irradiation

I_h = horizontal solar irradiation

I_{rc} = reflective solar irradiation

L = latitude

L_C = collector accumulated length

P = pressure – power output

P_1 = collector pressure drop

P_{Pi} = pump i loss

P_C = condenser loss

$\dot{Q}_R$ = Rankine cycle heat input

Re = Reynolds number

T = temperature

T_{abs} = average absorber temperature

T_{amb} = ambient temperature

U = average collector heat transfer coefficient

d_C = collector pipe diameter

h = enthalpy

h_s = solar hour angle

i = incident angle

k = sky optical depth

$\dot{m}_C$ = collector mass flow rate

n = day

s = entropy

ΔT_C = collector and glass temperature difference

α = solar altitude angle

α_S = solar azimuth angle

$\alpha\tau$ = collector constant

β = collector tilt angle

δ_s = solar declination angle

ε = collector emittance

η_B = boiler efficiency

η_R = Rankine cycle efficiency

ρ = ground reflectivity

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Appendix

Fig. 15. Solar thermal power plant 3D model