Modeling of a Power Plant by Integration of Solar Combined Cycle System: A Case Study.

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ABSTRACT

The use of natural gas as a source of fuel for a boiler in a steam power plant (Rankine cycle), for generation of electricity could be very expensive leading to high electricity tariff. Aside from the finite nature of fossil fuel and the high operational cost, the combustion of this fossil fuel brings about environmental pollution. This research combines parabolic trough collector and natural gas to power the boiler to minimize the said problem. Three Models of integration of solar combined cycle system (ISCCS) were designed and the best Model was selected.

(Keywords: pollution, electricity, combustion, parabolic, model, integration, solar, Rankine cycle)

INTRODUCTION

The average price of natural gas in the energy market is about \$0.65/mcf and the Egbin Power Plant requires an average of 14050.54 mmscf of natural gas per year to power its boiler for one 220MW unit. This is equivalent to \$9,132,851 and N, 397,326,203 per year. Aside from the finite nature of fossil fuel and the high operational costs, the combustion of this fossil fuel brings about environmental pollution and a daily emission of 473,616m^3 of CO_2 a greenhouse gas.

As it is globally known, continuous emission of CO_2 into the atmosphere is the major cause of global warming. Global warming has been a major concern to the world, and it was discussed at the last United Nation Energy Summit which took place in Durban. Part of the decisions taken at the summit is to slash CO_2 emission globally by 18% and also urge each country to utilize more of renewable source of energy than the non-renewable sources. Therefore, there is need to find an alternative means of energy which will minimize the said problem.

Natural gas is a non-renewable source of energy and also an exhaustible one. The combustion of this fossil fuel is expensive and also brings about environmental pollution which in turn causes the depletion in the ozone layer and global warming. On the other hand, the energy from the sun (solar energy) is renewable, clean and inexhaustible. Therefore this environmental-friendly source of energy should be utilized in order to reduce ozone layer depletion and global warming in this part of the world.

The aim of this research is to minimize cost of operation and generation of electricity at the Egbin thermal power station and also reduce the environmental pollution and emission of greenhouse gases which causes global warming. The objectives of this research are:

(i) Model a solar collector to be able to work optimally at the Power plant
(ii) Evaluate the cost implication of the retrofit
(iii) Determine the volume of CO_2 emission for the existing plant and the newly modified plant for comparison

Egbin Thermal Power Plant

Egbin thermal power plant is used as a case study in this work. The construction work of the Egbin thermal plant which started in 1982, is a steam power plant with six installed units each having a capacity of generating 220MW at maximum continuous rating (MCR) and a total installed power capacity of 1320MW. It is located in Ikorodu area of Lagos State, Nigeria on Lagos Lagoon.

The power plant was commissioned in the year 1985. The plant boilers are designed for dual firing of Natural Gas and Low /High Pour Fuel Oil (LPFO/HPFO). The plant was commissioned on oil firing and in October 1988 after piping Natural gas from Delta state by Nigeria Gas Company, Limited, a subsidiary of Nigeria National Petroleum Company (NNPC) located very close to the power plant. The first unit to be completed is Steam turbine (ST) - 3 which was commissioned on 11th May 1985. The remaining five units were commissioned one after the other within intervals of 6 months. Therefore between May 1985 and November 1987, the entire six units were handed over for commercial operation in the order: ST. 3, 2, 1, 4, 5, & 6, by Marubeni/Hitachi of Japan.

Since commissioning, the station has remained the single largest power station in the country, contributing between 30% - 40% of the grid required. It is also the biggest power station in West Africa sub-region.

Operation of Egbin Power Plant

One unit out of the six similar units in Egbin thermal plant will be used for this analysis. A schematic diagram of the continuous mass flow diagram of the plant is shown in Figure 1. Egbin Power Plant, has three turbines, i.e. high-pressure turbine (HPT), intermediate pressure turbine (IPT) and low-pressure turbine (LPT) which are mounted on a single shaft and the generator is coupled directly to them. It also consist of condensers, drain cooler, pumps, low pressure heaters, drum, economizer, primary and secondary super heaters and reheater.

Fuel (Natural gas) is burnt in the furnace where much of the heat is transferred to the boiler tubes which line the walls. The water used is gotten from the well and it's demineralized before use. The steam-water mixture formed in the boiler tube proceeds to a steam drum, where the vapour and liquid are separated. The liquid is returned to the boiler tube and the vapor proceeds to a super-heater. There, the hot gases leaving the flame region super-heat the saturated steam produced in the boiler section. The super-heated steam proceeds to the HPT, Single stage reheating is employed between the HPT and IPT stages. The exhaust steam leaving the HPT is reheated and sent to the IPT, and then goes directly to the LPT.

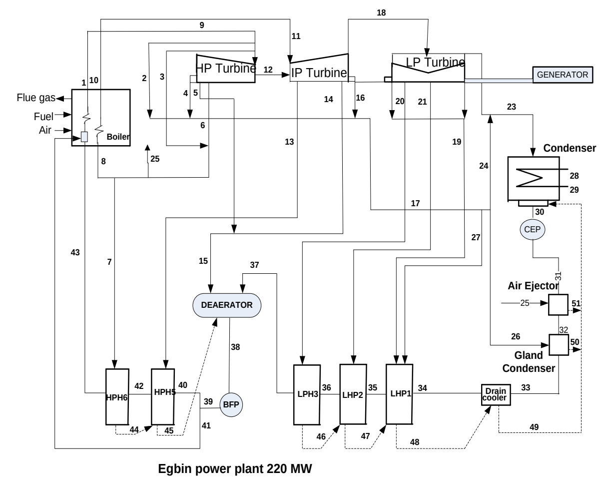

Figure 1: Mass Flow Diagram of Egbin Power Plant.

The steam exiting from the LPT proceeds to the condenser which will produce a vacuum or desired back pressure at the turbine exhaust. One of the streams exiting LPT is fed to the deaerator by condensate extraction pumps (CEP) through three low-pressure feed water heaters (LPH). The extraction streams for these heaters are bled from different stages of the LPT. The pegging steam for deaerator comes from the IPT exhaust. The drips from the LPHs are cascaded backward and the final drip from the No 1 LPH is fed to the condenser.

The feed water from the deaerator is fed to the economizer through boiler feed pumps (BFP) and two high-pressure feed water heaters (HPH). The extraction steams for HPHs are coming from IPT and the cold reheat line (CRH), respectively. The feed water exit the economizer through the drum to primary super heater and secondary super heater and back to the HPT and on and on it repeats the journey. The turbine is throttle-governed with control valves before the HP and IP stages.

The generator is directly coupled to the rotor of the turbine so they both turn at 3000rpm. It generates a 3- phase AC power of 220MW at full capacity. Its windings are excited with a DC 440V. The windings are cooled with hydrogen gas at a pressure of 210kPa.The generator current is 8.87A, with output voltage of 16kV, before being stepped up to 330kV by the generator transformer for onward transmission to load centre. The unit transformer steps down the voltage from 16kV to 6.6kV for Unit auxiliaries' use. The generated energy is evacuated out through the 330kV Ikeja West lines 1 & 2, Aja lines 3 & 4 and Benin lines 7 & 8 and 132kV Ikorodu lines 5 and 6 to the National grid.. The operation sequence and parameters were given by the Managing Director of the plant Engr. Omokhode (Omokhode, 2010).

BACKGROUND STUDY

Solar power cycle is integrated into existing thermal power plant. Therefore, the design involves consideration of the two systems - solar field and the Rankine cycle. The plant is an integration of different components which work independently to form a unit. The ISCCS is a new design concept that integrates a parabolic trough plant with a Rankine combined-cycle plant (Price et al., 2002). The ISCCS has generated much interest because it offers an innovative way to reduce cost and improve the overall solar-to-electric efficiency.

A process flow diagram for an ISCCS is shown in Figure 2. The ISCCS uses solar heat to preheat water to a reasonable temperature before entering into the boiler, so that the boiler will require less fuel to attain its final required temperature. In this design, solar energy is generally used to generate additional steam by preheat and steam superheating.

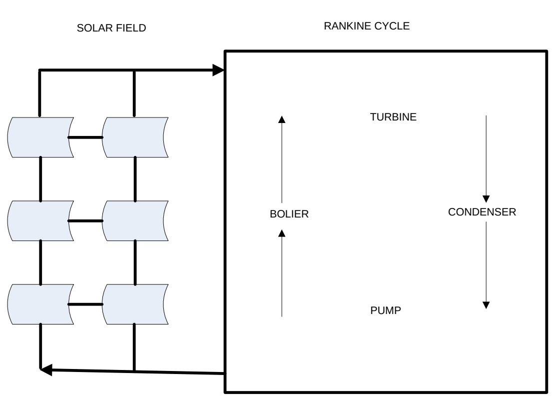

Figure 2: Schematic Diagram of ISCCS.

Earlier Works on Solar Power Plant

In 1912, the American engineer Frank Shuman put into operation the first large scale solar powered plant in Cairo, Egypt. Its job was to provide irrigation water from the Nile. He used a trough-type parabolic collector to focus the sun rays onto a black metal pipe to produce steam (Butti and Perlin, 1980). The systems' peak output was about 50kW. The total collector area was about 13,000ft^2(1207m^2).

Also research into organized, large scale development of solar collectors for power generation began in the U.S in the mid-1970 under the Energy Research and Development Administration (ERDA), and continued with the establishment of the U.S. Department of Energy (DOE) in 1978 (Gee, 2001).

Parabolic trough collectors capable of generating temperatures greater than 500^{O}C (932^{O}F) were initially developed for industrial process heat (IPH) applications. Much of the early development was conducted by or sponsored through Sandia National Laboratories in Albuquerque, New Mexico. Numerous process heat applications, ranging in size from a few hundred to about 5000 m^2 of collector area, were put into service. Parabolic trough development was also taking place in Europe and culminated with the construction of the IEA Small Solar Power Systems Project/Distributed Collector System (SSPS/DCS) in Tabernas, Spain, in 1981. And, in 1982, Luz International Limited (Luz) developed a parabolic trough collector for IPH applications that was based largely on the experience that had been gained by DOE/Sandia and the SSPS projects, which marked the era of solar on commercialized scale (Price et al., 2005).

The first commercial use of solar energy to drive Rankine cycle was in 1983 by LUZ (Gee, 2001). In 1983, LUZ negotiated a power purchase agreement with Southern California Edison (SCE) for the Solar Electric Generating Systems (SEGS) I and II plants. Later, with the advent of the California Standard Offer power purchase contracts for qualifying facilities under the Public Utility Regulatory Policy Act (PURPA), LUZ was able to sign a number of SO contracts with SCE that led to the development of the SEGS III through SEGS IX projects. Initially, the plants were limited by PURPA to 30 MW in size; later this limit was raised to 80 MW. These plants, developed by Luz International Limited (Luz) between 1984 and 1990, range in size from 14–80 MW and comprise 354 MW of installed electric generating capacity. More than 2,000,000 m^2 of parabolic trough collectors have been operating at the SEGS sites daily for up to 20 years and, as the year 2003 ended, these plants had accumulated to 154 years of operational experience (Price et al., 2005).

LUZ plants use synthetic oil as the heating medium, which limits the working temperature of the cycle due to the collapse of the properties of oil at a very high temperature- above 400^{o}C, (Odeh, 1998). Limitation of the oil working temperature imposes limitation on the plant efficiency. The use of a primary heating medium also increases the installation and the operational cost of the whole plant which made the competition against the conventional fuel unfavorable. A direct steam generation collector (DSG) was therefore earlier proposed as a future generation of the LUZ type trough collector (Cohen and Kearney, 1994); with stated following advantages – elimination of the costly synthetic oil, intermediate heat transport piping loop and oil to stream heat exchanger (Odeh et al., 1998).

The collector thermal loss with respect to the synthetic oil working fluid temperature was then later modeled by (Odeh et al., 1998) with vacuum, air and varying wind speed. They also modeled heat transfer coefficient for the single phase water and dry steam regions using water and steam properties to evaluate the Reynolds number, Prandtl number and the Nusselt number. To find the heat transfer coefficient in the two phase zone (for water as the heating medium) the type of flow pattern was determined and the overall collector efficiency evaluated by determining the extent of each phase region (Odeh et al., 1998).

In 1998, Steinmann and Goebel developed and implemented models to describe the dynamic and steady state behavior of DSG in simulation program (Steinmann and Goebel, 1998). DISS (Direct Solar Steam) project was later sponsored, in 1999, by the European Union in Spain to study the behavior of direct steam generation (DSG) in parabolic trough collectors with maximum working capacity of about 2MW (Steinmann and Eck, 2001). Various tests on the project showed the feasibility of DSG in horizontal absorber tubes and no critical situations was observed during the steady state and the transient state. (Eduardo et al., 2001).

The combination of solar cycle and rankine cycle for the generation of electricity has previously been carried out by Cohen and Kearney (1994), Odeh et al. (1998) Spellings et al. (2011) , Steinmann and Goebel (1998), Steinmann and Eck (2001), Eduardo et al., (2001), Price et al., (2005) and Spellings et al., (2011). However none of them took into consideration the economic implication and also the position and arrangement of the solar collector to attain a model that best fit the ISCCS.

Parabolic Trough

A parabolic trough is a type of solar thermal energy collector. It is constructed as a long parabolic mirror (usually coated silver or polished aluminum) with a Dewar tube running its length at the focal point (Price et al, 2002). Sunlight is reflected by the mirror and concentrated on the Dewar tube. The trough is usually aligned on a north-south axis, and rotated to track the sun as it moves across the sky each day.

THEORY AND METHODOLOGY

Mass and Energy Balance: Energy balance is based on the first law of thermodynamics. This states that energy can neither be created nor destroyed but can be converted from one form to another. The main performance criteria required in this analysis are commonly power output and thermal efficiency. Using the steady state equation and neglecting the potential and kinetic energy changes, the energy balance of the system can be represented as (Ameri et al., 2009);

$$\dot{Q} - \dot{W} = \sum \dot{m}_e h_e - \sum \dot{m}_i h_i$$
$$\sum \dot{m}_i = \sum \dot{m}_e \quad \textbf{(1)}$$

Where m = mass flow rate
h=enthalpy
e and i = exit and inlet

The list of component in Egbin Power Plant in Figure 1 and their energy equation is shown in Table 1.

Table 1: Energy Performance Equation from the Components of Egbin Power Plant.

Component	Energy Equation
Boiler	Heat supplied ; $Q_B = (\dot{E}_1 + \dot{E}_{10}) - (\dot{E}_{43} + \dot{E}_{41} + \dot{E}_8)$
High pressure turbine	Turbine work ; $(\dot{W}_T)_{HPT} = \dot{E}_9 - (\dot{E}_2 + \dot{E}_3 + \dot{E}_4 + \dot{E}_5 + \dot{E}_6 + \dot{E}_{12})$
Intermediate pressure turbine	Turbine work ; $(\dot{W}_T)_{IPT} = (\dot{E}_{11} + \dot{E}_{12}) - (\dot{E}_{13} + \dot{E}_{14} + \dot{E}_{16} + \dot{E}_{18})$
Low pressure turbine	Turbine work ; $(\dot{W}_T)_{LPT} = \dot{E}_{18} - (\dot{E}_{22} + \dot{E}_{19} + \dot{E}_{20} + \dot{E}_{21})$
Condenser	Heat rejected ; $\dot{Q}_C = (\dot{E}_{23} + \dot{E}_{28} + \dot{E}_{49} + \dot{E}_{50} + \dot{E}_{51}) - (\dot{E}_{29} + \dot{E}_{30})$
Pump (CEP)	Pump Work ; $\dot{W}_{CEP} = \dot{E}_{31} - \dot{E}_{30}$
Pump (BFP)	Pump Work ; $\dot{W}_{BFP} = (\dot{E}_{39} - \dot{E}_{38})$
Low pressure Heater 1	$\dot{Q}_{add} = \dot{E}_{35} - \dot{E}_{34}$ $\dot{Q}_{rej} = \dot{E}_{19} + \dot{E}_{27} + \dot{E}_{47} - \dot{E}_{48}$
Low pressure Heater 2	$\dot{Q}_{add} = \dot{E}_{35} - \dot{E}_{36}$ $\dot{Q}_{rej} = \dot{E}_{21} + \dot{E}_{46} - \dot{E}_{47}$
Low Pressure heater 3	$\dot{Q}_{add} = \dot{E}_{36} - \dot{E}_{37}$ $\dot{Q}_{rej} = \dot{E}_{20} - \dot{E}_{46}$
Deaerator	$\dot{E}x_{37} + \dot{E}x_{15} + \dot{E}x_{45} - \dot{E}x_{38}$
High pressure heater 5	$(\dot{E}x_{40} + \dot{E}x_{13} + \dot{E}x_{44}) - (\dot{E}x_{42} + \dot{E}x_{45})$
High pressure heater 6	$(\dot{E}_7 + \dot{E}_{42}) - (\dot{E}_{44} + \dot{E}_{43})$
Air ejector	$(\dot{E}_{33} + \dot{E}_{48}) - (\dot{E}_{34} + \dot{E}_{49})$
Gland Condenser	$(\dot{E}_{31} + \dot{E}_{25}) - (\dot{E}_{51} + \dot{E}_{32})$
Generator	$\dot{W}_{Net} = \sum \dot{W}_T - \sum \dot{W}_p$

Heat Supplied by the Boiler: The total heat supplied by the boiler in kW to the fluid, can be represented by the equation below:

$$\dot{Q}_1 = \sum \dot{m} C_p \Delta T \qquad \textbf{(2)}$$

Where, m =mass flow rate of steam
C_p=specific heat capacity of water
ΔT=change in temperature

Heat Supplied to the Boiler: The major constituent of the total heat input to a boiler (in kW) is the LHV of the natural gas burned.

$$Q_{sp} = \dot{m}_f \times LHV \text{ (in kW)} \qquad \textbf{(3)}$$

$$Q_{sp} = LHV \text{ in kJ/kg or kJ/m}^3$$

Where, m_f = mass flow rate of fuel
LHV=lower heating value of the fuel

Fuel Cost by the Boiler:

$$C_{fuel}(\$) = 1000 \times \mu \times \Gamma \qquad \textbf{(4)}$$

Where, μ = to the total amount of natural gas required by the boiler per year in mmscf
Γ = to the unit price of natural gas in the energy market in (dollars/mcf)

Therefore the unit cost of heat supplied in $/kW is given the equation below:

$$\dot{C}_{fuel}(\$/kw) = \frac{C_{fuel}}{Q_1} \qquad \textbf{(5)}$$

Cost of Savings:

$$cost\ of\ saving\ achieved = ExistingUniyCost - ModifiedUnitCost \qquad \textbf{(6)}$$

Efficiency of the Boiler: The efficiency of the boiler is estimated from the ratio of the heat supplied by the boiler to heat supplied to the boiler. Mathematically:

$$\eta_{boiler} = \frac{Q_1}{Q_{sp}} \qquad \textbf{(7)}$$

Declination: The declination is the angular position of the sun at solar noon, with respect to the plane of the equator. Its value in degrees is given by Cooper's equation:

$$\delta = 23.45 \sin\left(2\pi \frac{284 + n}{365}\right) \qquad \textbf{(8)}$$

Where, *n*= day of year (i.e. *n* =1 for January 1, *n* =32 for February 1, etc.). Declination varies between -23.45° on December 21 and +23.45° on June 21.

Solar Hour Angle and Sunset Hour Angle: The sunset hour angle ω_s is the solar hour angle corresponding to the time when the sun sets. It is given by the following equation:

$$\cos\omega_s = -\tan\varphi\tan\delta \quad \textbf{(9)}$$

Where, ψ = the latitude of the site (Egbin), which is 6.16

Therefore sunrise hour (H_{sr}) is given by:

$$H_{sr} = 12 - \frac{\omega_s}{15} \quad \textbf{(10)}$$

and the sunset hour (H_{st}) is given by:

$$H_{st} = 12 + \frac{\omega_s}{15} \quad \textbf{(11)}$$

Hourly Extra-Terrestrial Radiation:

$$I_o = I_{sc}(1 + 0.033\cos(360n/365)[\sin\delta\sin\varphi + \cos\delta\cos\varphi\cos\omega) \quad \textbf{(12)}$$

Where I_{sc} = solar constant 1,367 W/m^2

Heat Demanded by the Collector: The heat demanded by the parabolic trough can be calculated from the formula (Eastop and McConkey, 2001):

$$Q_{load} = \dot{m}C_p\left(T_{out} - T_{in}\right) \quad \textbf{(13)}$$

Energy Supplied by the Sun:

$$\dot{E} = I_o A_T \quad \textbf{(14)}$$

Total Required Collector Area: The required collector area can be calculated from the formula:

$$A_T = \frac{Q_{load}}{I_o\eta_c} \quad \textbf{(15)}$$

Where, η_c=efficiency of the solar collector

Parabolic Trough Costing: The cost of a parabolic trough is around \$100/m^2 and the cost of thermal storage is with \$9/kWh-t (Price et al, 2002). Therefore, mathematically:

$$C_{pur}(\$) = 100 \times A_T \quad \textbf{(16)}$$

$$C_{therm}^{st}(\$) = 9 \times Q_{load} \times t \quad \textbf{(17)}$$

Where *t* is the total time required for thermal storage in hours.

Efficiency:

$$\eta_{sys} = 1 - \{(1 - \eta_{collector})(1 - \eta_{boiler})\} \quad \textbf{(18)}$$

Determining the Fuel Consumption:

$$Q_{NG} = \frac{100\dot{Q}_1}{(\eta_{boiler})(Q_{sp})} \quad \textbf{(19)}$$

Where, Q_{sp} = LHV in kJ/m^3 see Table 4.

$$\dot{m}_{NG} = \frac{100\dot{Q}_1}{(\eta_{boiler})(Q_{sp})} \quad \textbf{(20)}$$

Where, Q_{sp} = LHV in kJ/kg

Emission Rate:

$$\left(\dot{m}_{CO_2}\right)_{NG} = \rho_{CO_2^P} \times Q_{NG} \quad \textbf{(21)}$$

- For firing natural gas (m^3/s)

$$\left(Q_{CO_2}\right)_{NG} = \frac{\left(\dot{m}\right)_{NG}}{\rho_{CO_2^P}} \quad \textbf{(22)}$$

Where $\rho_{CO_2^P}$ =molar density of the CO_2 at the product end (kg/m^3)

Volume of CO_2 Emitted:

$$V = Q_{CO_2} \times t \qquad \textbf{(23)}$$

Where t is time in seconds

Reduction in CO_2 Emission:

$$\Delta V = ExixstingUnitEmission - ModifiedUnitEmission$$

$$\%\Delta V = \frac{ExistingUnitEmission - ModifiedUnitEmission}{ExistingUnitEmission} \times 100 \qquad \textbf{(24)}$$

RESULTS AND CALCULATION

The relevant modeling equations and performance criteria for each component of the plant and the overall plant has been discussed in details in the methodology. The relevant thermodynamic properties (flow rate, temperature and enthalpy) of water and steam at different nodes labeled in numbers on the flow diagram (Figure 1) as well as their corresponding energy is calculated using the equations on Table 1 the energy performance results were obtained and presented in the Table 2.

The base case of this research is the boiler system only, powered by natural gas which runs for 22 hours daily and 8000hours yearly. For the course of this research, three ISCCS models will be considered for comparison. Which are:

i) Solar collector or solar field in the morning, boiler used at night (no thermal storage)

ii) Solar collector or solar field preheats the steam to a certain temperature, and then the boiler heats it up to the required temperature. (Thermal storage required)

iii) Solar collector heats one stream, while the boiler heats another stream.

In Model 1 and Model 3 the solar collector and the boiler are connected in parallel, while in Model 2 they are connected in series to each other.

Table 2: Design Condition Properties of Streams of Water and their Energy. Energy Values at h_o=104.92 KJ/Kg), T_o = 298.15K.

Components	Work Output (KJ/s)	Work Input (KJ/s)	Heat Added (KJ/s)	Heat Rejected (KJ/s)
Turbines				
LPT	99883.0145			
IPT	65593.36542			
HPT	58179.52944			
Total	**223655.9094**			
Heaters				
LPH 1			11118.76504	
LPH 2			14786.95067	
LPH 3			15433.12839	
DEAERATOR			38987.5435	
HPH 5			23618.55333	
HPH 6			31479.784	
Total			**135424.7249**	
Pumps				
CEP		0.150841249		
BFP		3450.418141		
Total		**3450.568982**		
Others				
Drain Cooler			15915.50757	
Air Ejector			322.9410367	
Gland Condenser			773.9105278	
Total			**17012.35914**	
Condenser				186666.4224
Boiler			507331.8108	

THE BOILER

Below are the various streams that enter and leave the boiler as shown in Figure 1 and their parameters.

The heat input from boiler to water and the heat transferred to the boiler, given by Equations 2 and 3, respectively, is calculated. Table 3 shows the inlet and outlet stream of the boiler and hence the heat supplied by the boiler to the steam is calculated to be 359,974.39kW. Table 4 shows the constituents gases that makes up the fuel (natural gas) hence the heat supplied by the natural gas to the boiler is computed to be 574,611.14kW.Therefore the efficiency of the boiler given by Equation 7 is calculated to be 62.6%.

Table 3: Thermodynamics Properties of Stream Enter the Based Case Boiler.

Stream	Flow (kg/h)	T (°C)	P(KPa)	h (KJ/Kg)	Type
41	20000	165.5	709.5	707.2	Inlet
43	627504	236.6	3209	1023.3	Inlet
8	579724	351.2	3289	3114	Inlet
1	647504	541	12913	3446.6	Outlet
10	579724	541	3129	3546.7	Outlet

Table 4: Natural Gas Constituents and their Lower Heating Value (LHV).

Component	% volume	% Mass	Density (kg/m^3)	LHV (Kj/Kg)	LHV (Kj/m^3)
Methane (CH_4)	89.4	81.27	0.668	50050	33433.4
Ethane (C_2H_6)	8.6	14.65	1.264	47520	60065.2
Propane (C_3H_8)	0.4	1	1.882	46340	87211.8
Carbon IV Oxide(CO_2)	0.6	1.5	1.842	Nil	Nil
Nitrogen (N_2)	1	1.59	1.165	Nil	Nil
Total	100	100		48100.7	35377.9

The average volume of natural gas used in running the boiler per year is computed from to be 14050.54mmscf (Omokhode, 2010) for one unit of 220MW. Therefore the fuel cost by the boiler given by Equation 4 is $9,132,851 per year which is equivalent to N1, 369,927,650 (one billion, three hundred and six-nine million, nine hundred and twenty-seven thousand, six hundred and fifty naira) per annum.

The unit cost of fuel therefore given by Equation 5 is $1.15/kWh (N172.5/kWh) since the boiler runs for 22 hours daily.

The combustion of natural gas is given by the chemical equation:

$$0.894\, CH_4 + 0.086\, C_2H_6 + 0.004\, C_3H_8 + 0.006\, CO_2 + 0.01\, N_2 + \{2.109aO_2 + 7.930aN_2\}_{air} \longrightarrow$$

$$1.084\, CO_2 + 2.062\, H_2O + (a-1)\, 2.109\, O_2 + (7.930a + 0.01)\, N_2$$

The mole of CO_2 on the product end is 1.084, therefore the fuel consumption in (m^3/s) and (kg/s) given by Equations 19 and 20 is calculated to be $16.25m^3/s$ and 11.95kg/s, respectively.

Hence the emission of CO_2 in m^3/s given by Equation 21 is calculated to $5.98m^3/s$.

From the above analysis, the daily emission of CO_2 is calculated to be $473{,}616m^3$ per day and yearly emission is calculated to be $172{,}224{,}000m^3$ per year by Equation 23.

Summary of the Base Case

Name	Boiler (basecase)
Output	359,974.39kW
Vol. of natural used /yr	14040.54mmscf
Area required	Nil
C.O.P	$19,852,892.09
Cost of thermal storage	Nil
Cost of fuel	$9,132,851
O&M Cost	$10,959,421.2
Cost of saving	Nil
Fuel consumption	$16.25m^3$ /s
CO_2 emission rate	$5.98m^3/s$
Vol. of CO_2 emitted	$473{,}616m^3$/day
Efficiency	62.6 %

ISCCS

Egbin thermal station, located in Ikorodu Lagos is on Latitude 6.16. The solar declination, the sunrise and sunset hour, the average daily insolation and extra-terrestrial insolation is computed based on Equations 8-12.

For the course of this research the assumption that there is solar insolation from (8am in the morning till 5pm in the evening everyday), this was due to the fact that the assumed time still falls in between the calculated sun rise and sunset hour daily given by Equations 10 and 11.

In siting solar field the potential of the field must be evaluated. Among the siting potential factors which must be satisfied are the: direct normal solar resource level, land slope, environmental sensitivity, and contiguous area. Parabolic trough solar power plants require high Direct Normal Insolation (DNI), or beam radiation, for cost-effective operation; the required size of the solar field for a given power plant capacity is in general directly proportional to the DNI level. In course of this study, our design will be based on LS-3 design specification which is shown in Table 5.

MODEL 1

As stated above, for the first model (Figure 3), the solar collector (parabolic trough) is used in the morning (8am-5pm) for 9 hours, and the base case boiler system runs for the remaining 13 hours.

Table 5: Design Specification for LS-3 Parabolic Trough.

TYPE OF COLLECTOR	LS-3
STRUCTURE	V-truss framework
APERTURE WIDTH(M)	5.76
FOCAL LENGTH(M)	1.71
LENGTH/ELEMENT (m^2)	12
ABSORBER LENGTH (m)	99
COLLECTOR AREA (m^2)	547
ABSORBER OUTERDIAMETER (m)	0.07
ABSORBER INNER DIAMETER (m)	0.066
GLASS DIAMETER (m)	0.115
GLASS TUBE EMISSIVITY	0.9
ABSORBER THERMAL CONDUCTIVITY ($Wm^{-1}k^{-1}$)	54
GEOMETRIC CONCENTRATION	82:1
MIRROR TYPE	Silvered low iron float glass
HEAT TRANSFER FLUID	Water

Source: Solar Energy Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS for publication in the ASME JOURNAL OF SOLAR ENERGY ENGINEERING. Manuscript received by the ASME Solar Energy Division, July 2001; final revision, January 2002.

The heat supplied to the boiler and heat transferred from the boiler to the steam still remains 574,611.4kW and 359,974.39kW. However, since it's an ISCCS, the heat demanded by the collector given by the Equation 13 is equivalent to the heat transferred from the boiler which is 359,974.39kW. Therefore the required area of the solar parabolic trough given by Equation 15 is calculated to be 533,300.0m^2, with the efficiency of the parabolic trough (75%) as specified by the manufacturer.

Table 6 presents the cost of material and installation of the parabolic trough for Model 1. The table was generated from parabolic trough model software designed by NREL. It shows that the total cost of material, labor and overall cost are $41,821,253.90, $11,508,746.10, and $55,330,000, respectively.

As stated earlier, in the above model the boiler run for only 13 hours daily which is equivalent to 4727 hours in a year. This will definitely result in reduction in cost of fuel compared to the based case. The unit cost of fuel used was computed to be $1.15/kWh; hence the total cost of fuel for this model is $ 5,381,617.13. This means the volume of the natural gas used will drop from 14050.4mmscf to 8279.4 mmscf.

Table 6: Model 1 Cost of Purchase of Material and Labor.

	Material Cost	Labor Cost	Total
Solar Field Solar Collector Mirrors	$8,607,728.65		$8,607,728.65
Solar Field - Solar Collector Receiver Tubes & Fittings	$12,588,869.35		$12,588,869.35
Solar Field - Solar Collector Frame	$14,341,342.32		$14,341,342.32
Solar Field - Solar Collector Assembly Misc. Components	$361,806.34		$361,806.34
Solar Field - Foundations & Support Structures	$3,243,123.56		$3,243,123.56
Solar Field - Instrument & Controls	$1,438,752.13	$9,320.54	$1,448,072.67
Solar Field – Electrical	$387,861.48	$179,632.19	$567,493.66
Solar Field - Labour Installation	$0.00	$11,271,707.87	$11,271,707.87
Solar Field - Fabrication tent	$175,819.24	$48,085.50	$223,904.74
Solar Field - Empirical Sun Tracker	$675,950.83	$0.00	$675,950.83
TOTAL	$41,821,253.90	$11,508,746.10	$53,330,000.00

From the above analysis it can be estimated that the percentage reduction in the cost of fuel usage and the volume of fuel usage is 41% which is a great savings.

Summary of Model 1

Name	Model 1
Output	359,974.39kW
Area required	533,000m^2
C.O.P	$53,330,000 *
Cost of thermal storage	Nil
Cost of fuel	$5,381,617.13
O&M Cost	$6,457,940.55
Cost of saving	$4,501,480.64
Volume of CO_2 emitted	279,864m^3/day

From the combustion equation, the molarity of CO_2 on the product side could be seen to be 1.084, therefore the fuel consumption in (m^3/s) and (kg/s) given by Equations 19 and 20 is calculated to be 16.25m^3/s and 11.95kg/s respectively. Thus the emission of CO_2 in m^3/s is calculated to 5.98m^3/s. Hence the daily emission of CO_2 is 279,864m^3 per day which is equivalent to 101,762,856m^3 per annum. From the above analysis it can be estimated that the percentage reduction in the volume of CO_2 emitted is 40.9%. The solar collector and the boiler are both connected parallel to each other. Therefore the efficiency of this system can be estimated by Equation 18 to be 90.65%

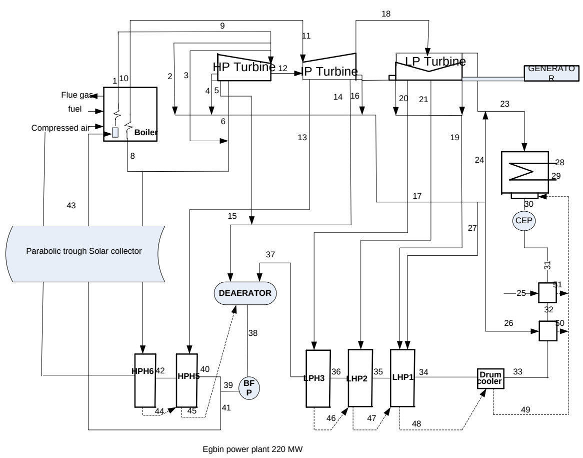

Figure 3: A Schematic Diagram of Model 1.

MODEL 2

In this model the solar collector heats up stream 43 to 500°C and the boiler heats it up to the required temperature alongside the remaining stream 41 and 8 (Figure 1). This will require a thermal storage of 13 hours because this will run for both night and day compared to Model 1.

The heat supplied to the boiler and the mass flow rate of fuel will reduce from 574,611 kW to 267,018.13kW and 11.84 kg/s to 5.5 kg/s respectively. The heat transferred from the boiler to the steam also changes from 359,974.39kW to 167,153.35kW. However the heat demanded by the collector given by the Equation 13 is equivalent to the heat transferred from the boiler which is 192,824.6kW. Therefore the required area of the solar parabolic trough given by Equation 15 is calculated to be 285,666.08m^2. The total cost of purchase of the parabolic trough given by Equation 16 is $28,566,608.

Table 7: Model 2 Cost of Purchase of Material and Labor.

	Material Cost	Labor Cost	Total
Solar Field Solar Collector Mirrors	$4,610,793.37		$4,610,793.37
Solar Field - Solar Collector Receiver Tubes & Fittings	$6,743,320.76		$6,743,320.76
Solar Field - Solar Collector Frame	$7,682,045.83		$7,682,045.83
Solar Field - Solar Collector Assembly Misc. Components	$193,804.23		$193,804.23
Solar Field - Foundations & Support Structures	$1,737,203.06		$1,737,203.06
Solar Field - Instrument & Controls	$770,678.20	$4,992.61	$775,670.81
Solar Field – Electrical	$207,760.86	$96,221.31	$303,982.17
Solar Field - Labour Installation	$0.00	$6,037,773.49	$6,037,773.49
Solar Field - Fabrication tent	$94,178.87	$25,757.35	$119,936.23
Solar Field - Empirical Sun Tracker	$362,078.05	$0.00	$362,078.05
TOTAL	$22,401,863.23	$6,164,744.77	$28,566,608.00

Table 7 presents the cost of material and installation of the parabolic trough for Model 1. The table was generated from parabolic trough model software designed by NREL. It shows that the total cost of material, labor and overall cost are $22,401,863.23, $6,164,744.77, and $28,566,608.00, respectively.

The unit cost of fuel used was computed to be $1.15/kWh; hence the total cost of fuel for this model is $ 4,228,979.76. This means the volume of the natural gas used will drop from 14050.4mmscf to 6505.05 mmscf per annum. From the above analysis it can be estimated that the percentage reduction in the cost of fuel usage and the volume of fuel usage is 53.8%.

Summary of Model 2

Name	Model 2
Output	192,824.6kW
Area required	285,666.08m^2
C.O.P	$28,566,608
Cost of thermal storage	$22,550,480
Cost of fuel	$4,228,979.76
O&M Cost	$5,074,775.71
Cost of saving	$5,884,645.5
Volume of CO_2 emitted	220,176m^3/day

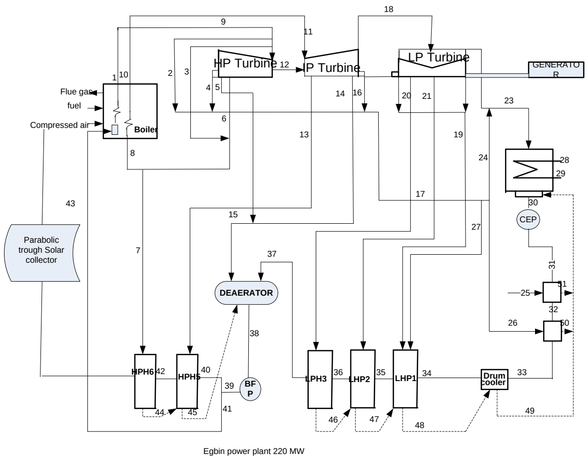

Figure 4: A Schematic Diagram of Model 2.

From Table 7 the purchase cost was found to be \$28,566,608 while the cost of 13 hours thermal storage given by Equation 17 is \$22,550,480. In this model, the fuel consumption is found to be 7.55m^3/s and 5.5kg/s respe8ctively. Hence the emission of CO_2 is calculated to be 2.78m^3/s. Thus, the daily emission of CO_2 is 220,176m^3 per day and 80,064,000m^3 per annum with a percentage reduction of 53.5%.

MODEL 3

In this model the solar collector heats up stream 43 to the required and the boiler heats up stream 41 and 8. This will require a thermal storage of 13 hours because this model will run for both night and day compared to Model 1.

The heat supplied to the boiler and the mass flow rate of fuel will reduce from 574,611 kW to 219,071.7kW and 11.84 kg/s to 4.55 kg/s respectively. The heat transferred from the boiler to the steam also changes from 359,974.39kW to 137,138.9kW. The heat demanded by the collector is 222,847.6kW. Therefore the required area of the solar parabolic trough is found to be 330,144.6m^2. The total cost of purchase of the parabolic trough given by Equation 16 is \$33,014,460.

Summary of Model 3

Name	Model 3
Output	222,847.6kW
Area required	330,144.6m^2
C.O.P	\$33,014,460
Cost of thermal storage	\$26,073,169.2
Cost of fuel	\$3,469,614.17
O&M Cost	\$4,163,536.97
Cost of saving	\$6,795,884.23
Volume of CO_2 emitted	180,576 m^3/day

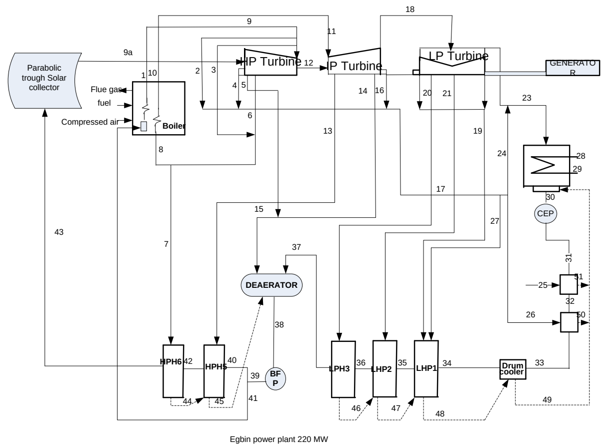

Figure 5: A Schematics of Model 3.

Criteria for the Comparison of Model

The models will be compared to the base case, based on the following criteria:

- Total Cost of purchase
- Required area
- Fuel cost
- Cost of saving
- Emission of carbon IV oxide

RELATIVE COMPARISON OF ALTERNATIVES

Table 8 shows the relative comparison of the various alternatives.

Table 8: Comparison of Alternative.

Criteria	Based Case	Modified case	Modified Case	Modified Case
	BOILER	MODEL 1	MODEL 2	MODEL 3
Required area of collector (m^2)	-	533,000	285,666.08	330,144.6
Total cost of purchase	-	\$53,330,000	\$28,566,608	\$33,014,460
Cost of thermal storage	-	-	\$22,550,480	\$26,073,169.2
Fuel Cost	\$9,132,651	\$5,381,617.13	\$4,228,979.76	\$3,469,614.17
Saving Cost/year*	-	\$4,501,480.63	\$\$5,884,645.5	\$6,795,884.23
%reduction in fuel cost	-	41%	53.7%	62%
CO_2 emission/day	473,616m^3	279,864m^3	220,176m^3	180,576
% CO_2 reduction	-	40.9%	53.7%	61.8

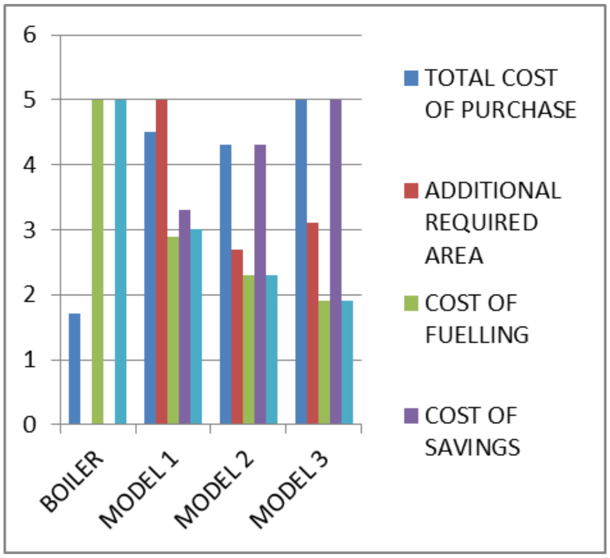

Figure 6: Relative Comparison of all of the Alternatives.

Table 9: Benefit – Cost Analysis

Mod	Total Cost ($)	N* Cost	Benefit ($)	N* Benefit	Benefit-Cost Analysis	Judg-ment
1	53,300,000	0.327	4,501,480	0.262	0.801	Not viable
2	50,817,088	0.311	5,884,645	0.342	1.100	Most viable
3	59,087,629	0.362	6,795,884	0.396	1.094	Viable

* N = Normalized

From the comparative study graph (Figure 6), it is seen that the best two alternatives are Model 2 and Model 3, this is because the two models requires lower cost of fuelling while invariably yielding higher cost of savings per annum; also for the two models there is lower emission of CO_2. In conclusion, judging from the benefit –cost analysis on Table 9, Model 2 is the most viable (*highest ratio greater than 1*).

From the above, it is seen that there is so much need for renewable source of energy in the nations' power plant. It is cheaper, clean, and environmental friendly.

REFERENCES

1. Omokhode G.E. 2010. "How Electricity is generated in Egbin Thermal Power Plant".

2. Price, H., D. Kearney, and E. Lupfert. 2002. "Advances in Parabolic Trough Solar Power Technology". *Journal of Solar Energy Engineering by ASME*. May 2002:124-109

3. Butti, K. and J. Perlins. 1980. *2500 Years of Solar Architecture and Technology*. Cheshire Books: London, UK.

4. Gee, R. 2011. ''Parabolic Trough Wind Tunnel Testing'.' Duke Solar, Presentation at Solar Forum 2001, Solar Energy: The Power to Choose, April 21-25. Washington, DC, 2011.

5. Odeh S.D, L. Morrison, and M. Behnia. 1998. "Modeling of Parabolic Trough Direct Steam Generation Solar Collectors". *Solar Energy*. 62(6):395–406. Elsevier Science Ltd.

6. Cohen, G., D. Kearney, and G. Kolb. 1994. ''Final Report on the Operation and Maintenance Improvement Program for CSP Plants'.' Report No. SAND 99-1290. SNL: Albuquerque, NM.

7. Spelling, J., D. Favrat, A. Martin, and G. Augsburger. 2011. "Thermo Economic Optimization of a Combined-Cycle Solar Tower Power Plant". Department of Energy Technology, Royal Institute of Technology, SE-100 44, 2011.

8. Steinmann, D. and M. Eck. 2001. "Direct Steam Generation In Parabolic Troughs: First Results of The DISS Project". *Proceedings of Solar Forum 2001*. Washington D.C.

9. Ameri. M, A. Pouria, and H. Armita. 2009. "Energy, Exergy, and Exergoeconomic Analysis of a Steam Power Plant: A Case Study". *Int. J. Energy Res*. 33:499–512.

10. Eastop, T.D. and A. McConkey. 2005. *Applied Thermodynamics for Engineering Technologists, Fifth Edition*. Pearson Education, Asia.

11. Egbin Thermal Power Plant. "Plant Layout".

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