PERFORMANCE MODELING OF SOLAR RETROFITS IN COMBINED CYCLE POWER PLANT

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CERTIFICATION

This is to certify that this project, **Analysis on Solar Retrofit in a Dual Pressure Combined Cycle Power Plant,** was carried out by **OTEJERE JOSEPHUS (ENG1704346), IMADEGBELO DANIEL (ENG1707195), OBENDE RITA (ENG1704313), UDEH KENNNETH (ENG1704515),** in the department of Mechanical Engineering, Faculty of Engineering, University of Benin.

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DEDICATION

This project is dedicated to God Almighty who made everything possible.

ACKNOWLEDGEMENT

First and foremost, we want to give thanks to God Almighty for the strength and wisdom to carry out this project.

Our sincere gratitude goes to our wonderful and highly esteemed supervisor Dr. Osarobo Ighodaro for his guidance, contributions, time, and disciplinary actions which inspired us to put more effort and ensure this project was a success, he is the reason this project is possible.

Our profound gratitude goes to our parents, whose love and support saw us through this project.

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NOMENCLATURE

M	Mass
h	Enthalpy
P	Pressure
η	Efficiency
v	Specific volume
$\dot{Q}_{Rad}$	Radiation losses
$\dot{Q}$	Energy in watts
ρ	Density
$\emptyset$	Relative humidity
ETAI/ETAIN	Variation in isentropic efficiency on compressor or turbine with regards to isentropic efficiency in nominal conditions
P(GT)	Power developed in the gas turbine
P(ST)	Power developed in the steam turbine
T1	Ambient temperature
T2	Inlet temperature to combustion chamber
T3	Inlet temperature to turbine section
T4	Exhaust temperature from gas turbine
P3	Pressure at entry to turbine section of the gas turbine

ABSTRACT

The aim of this project is the modelling and simulation of GT13E2 combined cycle gas turbine with the aid of the software EBSILON PROFESSIONAL, and carrying out analysis on solar retrofit. The design mode was modeled using guaranteed performance data from the plant, in the off design, temperature variation at inlet to compressor and other analysis were carried out. The model results were validated by comparing the actual operating data using error percentage analysis. The validation results obtained ranged from -0.0038% to 0% in design condition, while the results varied from -0.9202% to 10.24%.

From the research, we can conclude that as ambient temperature increases, the mass flow rate of air reduces and as such this reduces the power that can be developed in the gas turbine. Also, since the energy available in the flue gas from the gas turbine is reduced at higher ambient temperature, the power developed in the steam turbine reduces also. At higher ambient temperature, the overall cycle efficiency decreases.

In order to maintain the design exhaust temperature, extra fuel has to be burned to extend the combustion process.

The results achieved from the simulation of solar boosting revealed that as mass flow of solar steam increases, power developed in the steam turbine increases. However since the HRSG is a heat sensitive component, the limit to the amount of solar steam that can be added is 3Kg/s. If extra mass of water is added, issues will arise in the most critical part of the HRSG which is the evaporator. If the energy available at the location of the evaporator is not enough, steam would not be generated hence the steam cycle would fail.

From the analysis and simulation of the High pressure solar boosting and the Low pressure solar boosting, we can conclude that the highest extra power generated in the high pressure solar boosting is 3.2 MW while that of low pressure solar boosting is 1.7 MW, hence high pressure solar boosting is the best configuration.

CHAPTER ONE

INTRODUCTION

1.1 BACKGROUND TO THE STUDY

In today's modern world, with the different inventions of various devices and machines which improves lives, it is impossible to do without electricity. Energy has an established positive correlation with economic growth. Providing adequate, affordable and clean energy is a prerequisite for eradicating poverty and improving productivity in any nation (Jayakumar, 2009).

With increasing population comes increase in demand for energy and as a consequence, increased cost of electricity, increase in cost of living and reduced nonrenewable energy (Kabeyi et al, 2020). As such there is need for more studies on optimum energy application and minimizing of energy consumption. Hence to provide for this increase in power, new power plants with greater efficiency and the optimization of existing power plants has been considered.

In recent times, fossil fuels have dominated the global energy mix as most of the global electricity generation comes from combustion of fossil fuels. Fossil fuels are not renewable and as such have limited availability, also fossil fuels have negative impact on the environment such as the global warming, waste disposal and flue gas emissions (Kabeyi et al, 2020). Due to these reasons, there has been increased attention by researchers on ways to improve the design and operation of energy system and mitigate its negative effect on the environment.

Nigeria has a total installed power generation capacity of 16,384MW, power generation in Nigeria is based mainly from hydro and gas-fired thermal power plant.

Year	Current Power Generated (MW)	Expected Power Generated (MW)
2023	4616	22,000

Table 1.1. Energy Resources in Nigeria (source Nigeria Electricity Sector on energypedia.info)

As of 2023, the peak generating capacity is 4,616 MW, this is significantly lower than the installed capacity and is caused by factors including transmission and distribution losses, gas shortages, and plant maintenance. This power is insufficient for households and businesses as they are compelled to independent power generation at additional cost (Mudathir, 2017).

In recent times in Nigeria, gas turbines has been the preferred choice for power generation due to their high performance, good reliability, low capital cost, and short start-up time (Bolatturk, 2007). However, in a bid

to increase the generating capacity in the country, new configurations of this gas turbine has been considered and one of this configurations include the combined cycle.
Some of the combined cycle plants in Nigeria are shown in the table below;

POWER STATION	**CAPACITY**	**STATUS**
Afam VI Power Station	650 MW	Operational
Alaoji Power Station (NIPP)	1074 MW	Partially operational (225MW)
Okpai Power Station	480 MW	Operational
Olorunsogo II Power Station (NIPP)	675 MW NDPHC	Partially Operational

Table 1.2. Combined cycle power plants in Nigeria

The advantages of combined cycle is as follows; it can easily adapt to electricity demand by operating in part loads and hence avoid production surpluses. Combined cycle also have increased efficiency compared to conventional steam or gas turbine power plant and as such they utilize the energy available to a greater extend when compared to their counterparts. This simply means combined cycle consumes less fuel compared to conventional power plants. Combined cycle can use natural gas as their source of fuel and as such they have lower emissions of pollutants such as carbon dioxide, nitrogen oxide, and sulfur dioxide as natural gas is a cleaner fuel compared to coal fuel oil or any other petroleum derivatives. Finally, combined cycle do not require as much water as any conventional steam plant as the water required for the condensation of steam is less as the mass of steam required in the steam turbine is less for the equivalent power output.
Nigeria is richly blessed with natural gas and crude oil, however due to its adverse effect on the environment and with the western world shifting their attention to renewable energy, there is need to improve on the existing gas turbine technology.
In improving this technology, the term called efficiency is key as it affects other aspects of sustainability which includes cost effectiveness, effective utilization of resources, better design and analysis, energy security and better environment.
With modern technology, we can now model and analyze power generation systems. This modeling is of great engineering interest and is essential for the efficiency utilization of energy resources. Modelling provides an easy and efficient analysis of various power plant configurations without any need to carry out any actual tests as the calculations from these models are quite reliable. The most used method for this

analysis of energy conversion is the first law of thermodynamics or energy analysis, however, this method is limited because it doesn't provide a full measure of efficiency and losses.

One means of improving the efficiency of these various power plant is the concentrated solar power (CSP) technology and this method will be considered in this project. CSP systems concentrate a broad region of sunlight into a narrow beam by using lenses, mirrors, and tracking devices. Afterwards, a conventional power plant (solar thermoelectricity) uses the focused light as heat or as a heat source (Jayakumar, 2009).

Concentrated solar power technology provides clean energy source as the sun is a renewable energy, it has the ability to store energy and can generate electricity even when the sun is not shining. However some of its challenges can be its cost and land use as CSP plant requires a large amount of land which can compete with other land uses such as agriculture.

In light of the fact that CSP is a promising renewable energy technology that has the potential to significantly contribute to meeting our energy demands in the future, we suggest in this study a solar retrofit to increase the steam cycle's input energy from the combined cycle gas turbine.

1.1.1 AFAM VI POWER PLANT

The AFAM VI power station in Afam, Okoloma, Rivers State, Nigeria, is home to the gas turbine-based combined cycle that is the subject of this project's modeling. Three 150 MW gas turbines and one 200 MW steam turbine make up the power plant. All three gas turbines combine to create steam for the steam turbine, and each is connected to a separate heat recovery steam generator. The power plant produces 650MW in total. 2008 saw the plant's commissioning. It is one of Rivers State's four Afam gas plants, together with the 360 MW Afam (FPL) power station, the 450 MW Afam IV power station, and the 276 MW Afam V power station.

The Nigerian Electricity Regulatory Commission extended the Afam VI Power Plant's power producing license in November 2017. The plant supplied the grid with more than 25.97 million Megawatt-hours (MWh) of power between 2008 and 2017.

1.2 STATEMENT OF PROBLEM

Power plant components which have been manufactured in western nations and have been shipped to Nigeria operate in conditions different from which they were originally designed to operate and as such this affects the efficiency and power developed or consumed by such component and as such the question arises; How can we improve the efficiency and power output of the plant while giving consideration to cleaner energy sources?

1.3 AIMS AND OBJECTIVES

1.3.1 AIM

This study aims to model and analyze the off design performance of the combined cycle in AFAM VI Combined Cycle Power Plant.

1.3.2 OBJECTIVES

The objectives of the study are;

1. Obtain data for design and part load performance of Afam VI (31/03/23 – 10/4/23)
2. Development of model (10/4/23 – 21/04/23)
3. Validation of model(14/04/23 – 21/04/23)
4. Energy analysis of the power plant (21/04/23 – 23/06/23)
   i. Variation in ambient temperature
   ii. Variation of fuel and air flow
   iii. Variation of part load conditions
5. Incorporate the solar retrofit(23/06/23 – 14/07/23)
   i. Solar boosting with saturated high pressure steam
   ii. Solar boosting with cold reheated steam
6. Best configuration to optimize solar boosting (14/07/23 – 21/07/23)
7. Study the impact of solar steam injection on temperature and pressure (21/07/23 – 30/07/23)

1.4 SIGNIFICANCE OF STUDY

Gas Turbines are volumetric machines hence the volume flow rate of the compressor is constant once it has been designed. At higher ambient temperatures, the mass flow of air decreases due to lower air density. This means that a gas turbine designed and built in countries where the ambient temperature is close to ISO conditions being shipped to Nigeria would under-perform, producing lower power than it was originally designed. To make up for this reduction in input energy, a solar retrofit is proposed to supply extra energy to the steam cycle of the combined cycle gas turbine.

1.5 SCOPE AND LIMITATIONS

The scope of this project is we would be modeling AFAM VI power plant and we would be focusing on the major components in the plant.

CHAPTER TWO
LITERATURE REVIEW

2.0 OVERVIEW

Combined cycle, which consists of a toping cycle at a higher maximum temperature and a bottoming cycle at a lower maximum temperature, is a combination of two thermodynamic cycles combined for the purpose of improving maximum efficiency. The Brayton cycle is the toping cycle and the Rankine cycle is the bottoming cycle in a combined cycle gas turbine (Ibrahim and Rahman, 2011). The Brayton cycle rejects heat at a temperature that is convenient for using as the Rankine cycle's energy source because of its high source temperature. According to Saravamutto et al. (2009), the working fluid for the Rankine cycle is steam or water, whereas the working fluid for the Brayton cycle is air.

The gas turbine's combustion chamber burns liquid or natural gas to produce a steady pressure that turns the turbine or rotor connected to the generator to produce electricity. The waste heat from the gas turbine exhaust is recovered and used at the heat recovery steam generator to produce superheated steam, which powers a steam turbine that turns another generator.

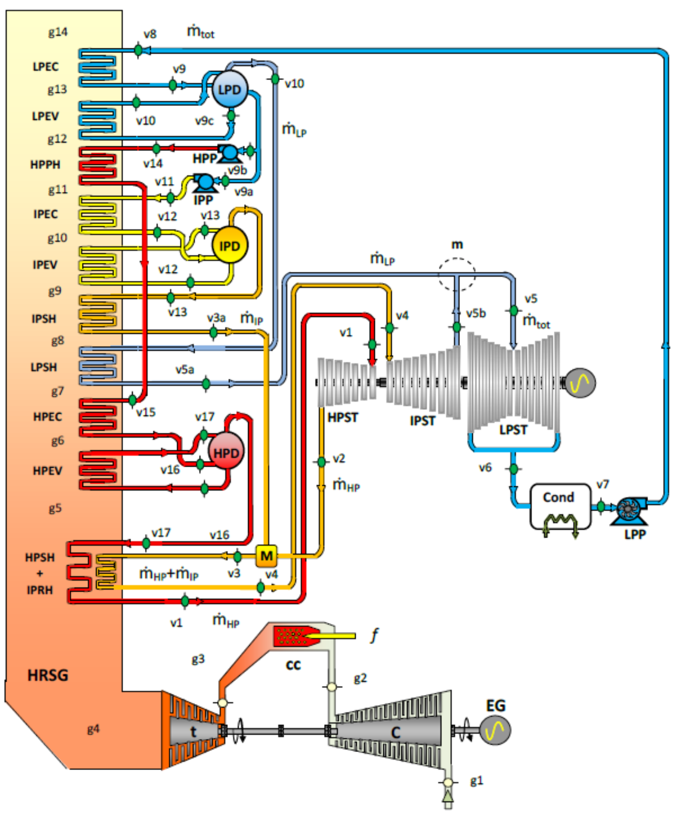


Figure 2.1 Schematic diagram of a combined cycle (Edgar Torres et al, 2017)

2.1. MAJOR COMPONENTS OF THE COMBINED CYCLE POWER PLANTS

2.1.1 GAS TURBINE

A gas turbine, alternatively known as a combustion turbine, is a continuous flow internal combustion engine. All gas turbine engines share common components in their power-producing section, which include:

- A rotating gas compressor.
- A combustor.
- A turbine that drives the compressor.

Using air as its working fluid, a gas turbine's basic operation is based on the Brayton cycle. In the beginning, ambient air is compressed, increasing its pressure. The compressed air is then mixed with fuel and ignited, resulting in combustion and the production of a high-temperature, high-pressure gas stream. Following its passage through a turbine, the high-temperature gas generates shaft work that powers the compressor. Any excess energy is released as exhaust gases, which can be used for other purposes. For example, they can be used to power a power turbine, which is a secondary turbine that can be used to power a fan, propeller, or electrical generator, or they can be directed toward other tasks. The design of a gas turbine is dictated by its specific function, which aims to maximize the energy distribution between shaft work and thrust. Notably, because gas turbines run as open systems and do not recycle the same air, the fourth stage of the Brayton cycle-cooling the working fluid is skipped.

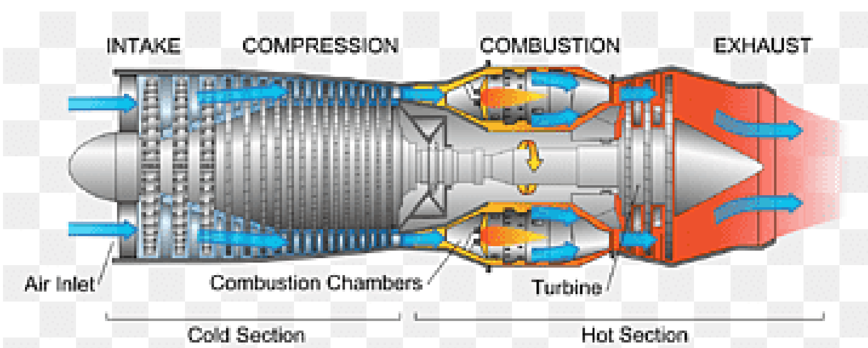


Figure 2.2. Gas Turbine

2.1.2. COMBUSTION CHAMBER

Combustion chamber, is where combustion reaction between fuel and oxygen takes place. In gas turbines, the combustor chamber gets high-pressure air from the compression system, heating it as fuel burns. This heated mix expands and goes to the turbine. Combustors must maintain stable combustion despite high airflow rates. They mix and ignite air and fuel, ensuring complete combustion. Combustors greatly impact an engine's efficiency, emissions, and response to changing conditions

In a gas turbine, the combustor's goal is to provide energy for the turbines, achieving this involves several design considerations, some of them include:

- Efficiently burn fuel to prevent waste and emissions.
- Maintain low pressure loss to aid turbine efficiency.
- Contain combustion within the combustor to prevent overheating further downstream.
- Ensure the ability to reignite at high altitudes if the flame goes out.
- Create a uniform temperature profile to avoid damaging the turbine.
- Allow for a wide range of operating conditions.

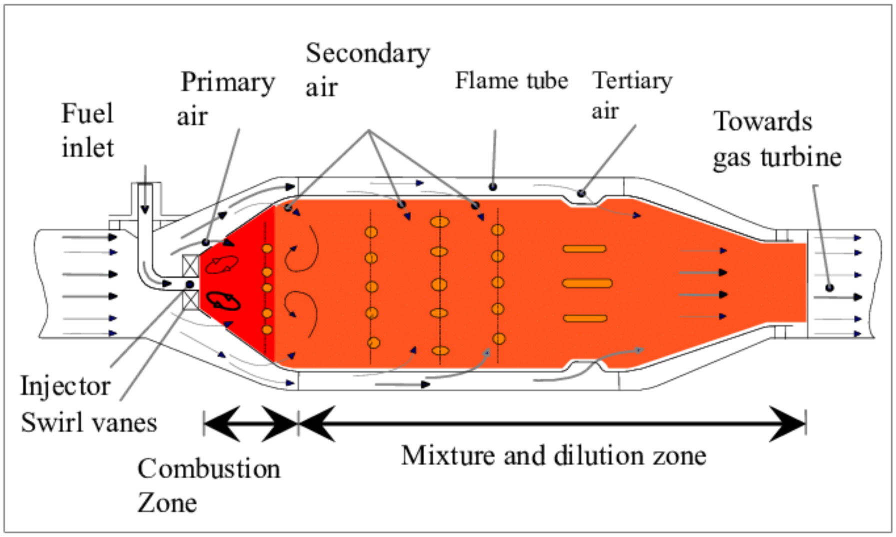


Figure 2.3. Combustion Chamber

2.1.3. HRSG (HEAT RECOVERY STEAM GENERATOR)

HRSGs (Heat Recovery Steam Generators) have four main components; economizer, evaporator, superheater, and water preheater. HRSG can be classified based on;

Exhaust Gas Flow: vertical and horizontal. In horizontal HRSGs, gas moves horizontally over vertical tubes, while in vertical HRSGs, gas moves vertically over horizontal tubes.

Pressure: there are single pressure (one steam drum) and multi-pressure (two or three steam drums) HRSGs. Multi-pressure ones have low, intermediate, and high-pressure sections, each with a steam drum and evaporator. Steam goes through superheaters to raise its temperature.

Some HRSG use additional burners called supplemental or duct firing. These burners add more energy, producing extra steam to boost the steam turbine's output, usually at a lower cost. This is often used for peak electricity needs. HRSGs may have diverter valves to control gas turbine input when there's no steam demand or the HRSG needs maintenance.

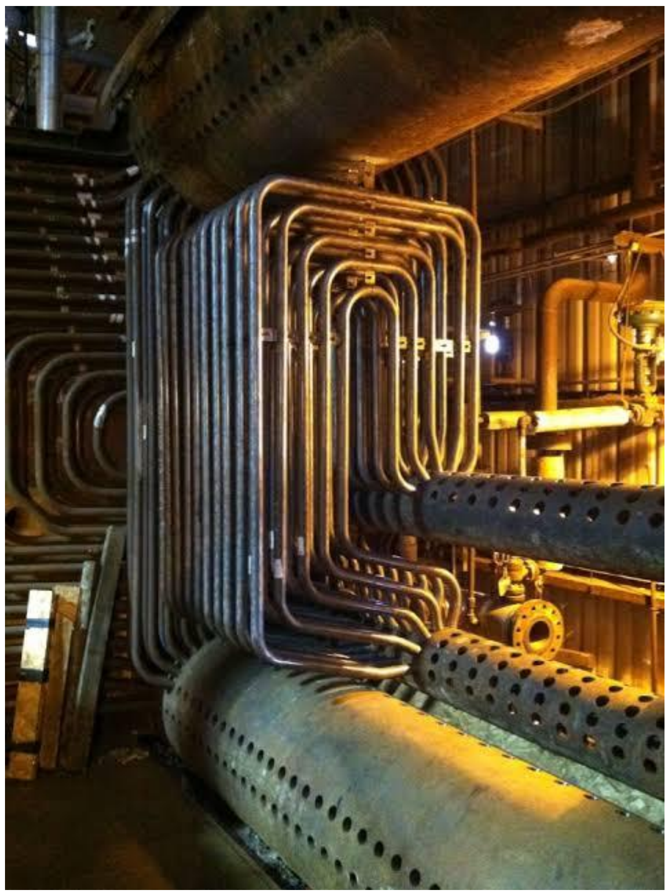


Figure 2.4. Super heater tubes

2.1.4. GENERATOR

A generator is a machine that converts fuel (derived from chemical energy) or motion (derived from potential or kinetic energy) into electrical energy for use in an external circuit. Steam turbines, gas turbines, water turbines, internal combustion engines, wind turbines, and hand cranks are examples of mechanical energy sources. Electrical grids rely heavily on generators to supply the majority of their electricity.

Apart from designs based on electricity and motion, photovoltaic generators utilize solar power, while fuel cell generators use hydrogen-based fuels to produce electricity.

The opposite process, converting electrical energy into mechanical energy, is carried out by electric motors, which are quite similar to generators. In fact, many motors have the capability to generate electricity from mechanical energy.

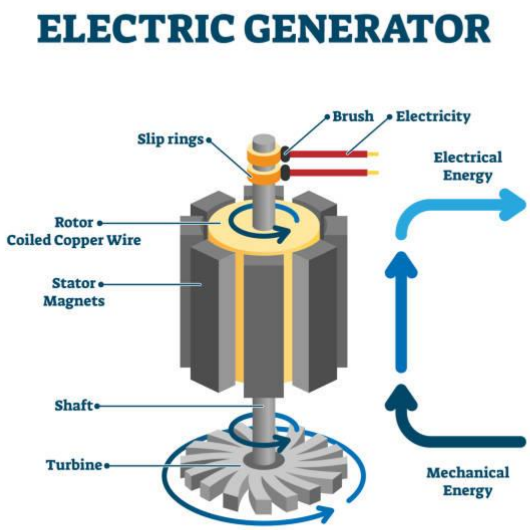


Figure 2.5. Generator

2.1.5. COMPRESSOR AND PUMP

Compressors and pumps are alike in that they both raise fluid pressure and move it through pipes. However, compressors mainly change the density or volume of gases, as gases are compressible. Liquids, being hard to compress, aren't typically compressed using compressors. On the other hand, pumps are designed to pressurize and move liquids.

Many compressors use multiple stages to gradually increase pressure. The second stage is often smaller to handle already compressed gas without reducing its pressure. Each stage boosts gas pressure and temperature unless intercooling is used between stages.

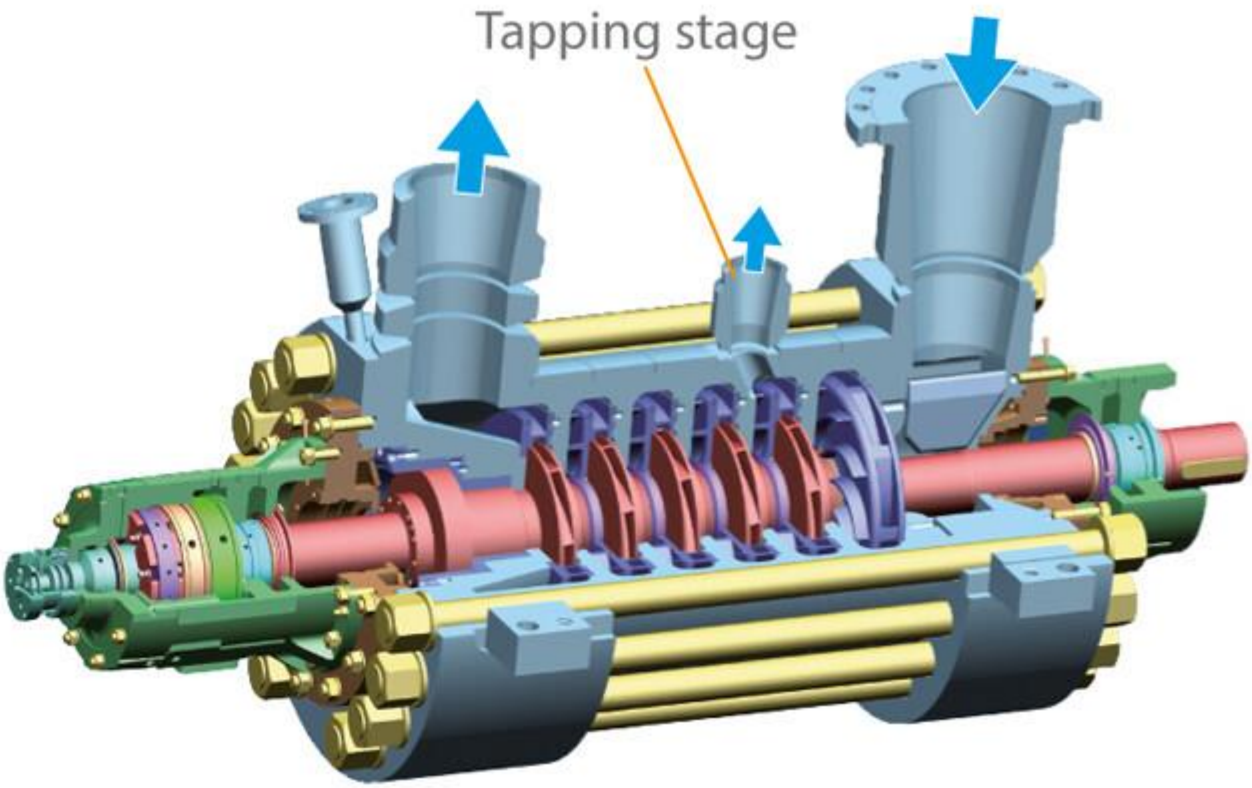


Figure 2.6. Boiler Feed Pump

2.1.6. STEAM TURBINE

The steam turbine is a type of heat engine that achieves significant enhancements in thermodynamic efficiency through a unique approach: employing multiple stages during the expansion of steam. This multi-stage design allows it to closely approximate the ideal reversible expansion process, thus optimizing its performance.

One of the notable advantages of a steam turbine is its ability to convert heat energy into rotary motion. By connecting it to a generator, this rotary motion can be harnessed to produce electricity. These combined systems, known as turbogenerators, form the backbone of thermal power stations. These power stations can be powered by a range of energy sources, including fossil fuels, nuclear fuels, geothermal energy, or solar energy.

However, the application of steam turbines comes with technical challenges. These include addressing issues like rotor imbalance, vibrations, wear and tear on bearings, and handling uneven expansion, which can lead to various forms of thermal shock. In larger installations, even robust turbines can face problems if they operate outside their optimal conditions, potentially causing mechanical issues or structural damage.

Therefore, precise engineering and maintenance are essential to ensure the reliable and efficient operation of steam turbines in power generation systems.

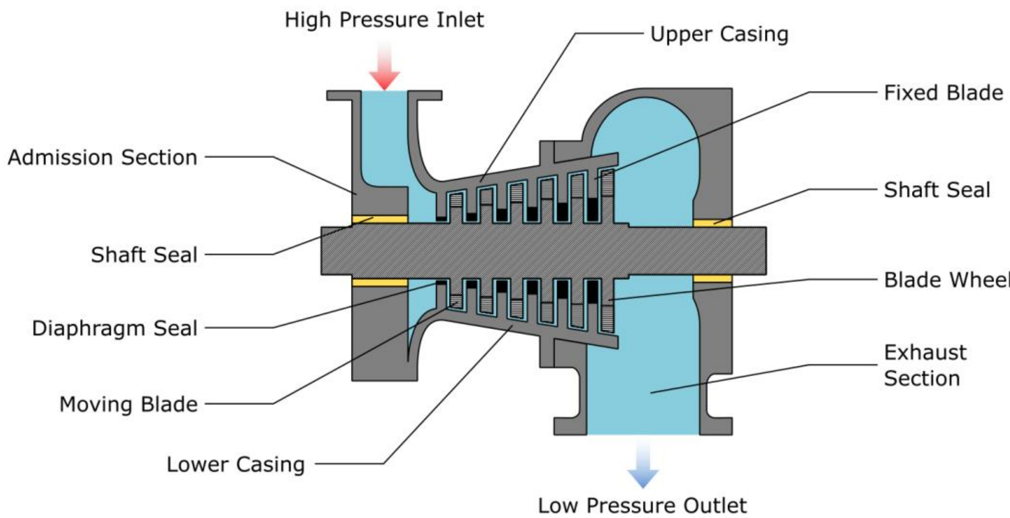


Figure 2.7. Steam Turbine

2.2 EARLY DEVELOPMENT

Most coal-fired and natural-gas fueled power cycles have efficiencies in the range of 30%. As such most of the heat energy from combustion process is released to the atmosphere as waste. The concept of utilizing this large waste heat from the gas turbine was realized in a Korneuburg-A plant in Austria in 1961. The power plant consists of 2 Brown Boveri & Cie 25MW steam turbine, with an overall efficiency around 32.5%. Though this was a small improvement from the regular 30% efficiency at the time, the Korneuburg-A proved that this concept could work (Daniel Green et al, 2021).

The critical component of the combined cycle which makes this improvement of efficiency possible is the Heat Recovery Steam Generator (HRSG) which is a heat exchanger. The HRSG transfers heat from hot flue gas from the exhaust of the gas turbine to water.

El-Wakil, (1984); Studied low power combined cycle, and in his research concluded that the power output from the steam turbine was about 50% lower than the power output from the gas turbine and the cycle usually consists of one deaerator and closed feed water heater as other component. The power output can be increased for a short period of time during peak hours by supplementary burning of fuel.

Although the characteristic compactness of the gas turbine is sacrificed when compared to binary cycle plants, the efficiency is much higher when compared to simple cycle and as such combined cycle is now used widely for large scale generation of electricity (Saravanamutto et al 2009, Ighodaro and Aburine 2011).

Early CCPP had the limitation of poor performance and the reasons for this includes;

- Steam turbines at the time were not optimized for combined cycle operations. It may have been adapted from existing designs for different applications, which didn't harness the waste heat from the gas turbines effectively

- The gas turbine used in korneuburg-A plant were based on early designs, characterized by lower compression ratios and less advanced combustion technology. These limitations resulted in lower thermal efficiencies as they couldn't extract as much energy from the fuel.

- Early combine cycle power plant faced difficulties in fully integrating the gas and steam cycles. The engineering and technology for efficient heat recovery and transfer were less sophisticated.

2.3 RECENT DEVELOPMENT

Combined cycle power plants have become increasingly significant in the world's electricity production in recent years. This power plant's primary goal is to increase efficiency, or ascertain the correlation between efficiency and electricity expenses. However, the environmental effects of burning fossil fuels to generate electricity have put the world in danger in a number of ways, including depletion of the ozone layer, global warming, and air pollution. Utilizing clean, renewable energy sources (like solar and wind) to generate electricity is one way to lessen the negative effects of fossil fuel electricity generation. Another is to reorganize already-existing fossil fuel power plants to increase generation efficiency and cut greenhouse gas emissions (Xiaoli et al., 2022).

Just roughly 23% of the world's generation capacity is currently accounted for by electricity generation based on clean and renewable energy (IRENA 2022). The majority of power plants still burn fossil fuels. It is crucial to reconfigure fossil fuel power plants to increase their thermal efficiency and power generation rate in order to account for this circumstance.

Combined cycle power plants (CCPPs) have a high capacity in addition to partially low emissions. CCPP produces 32% of waste and 68% of electricity. On the other hand, only 33% of electricity can be produced by other kinds of power plants. (2008) Niu and Liu.

The aforementioned benefits make it evident that, as CCPP usage rises, power plant forecasting model prediction and interpretation have gained popularity and importance among researchers.

To accurately predict a power plant's power output, we need to know the influencing factors and how they interact. Maximizing income from available megawatt hours (MW/h) and having a highly efficient, dependable, and sustainable power plant are also advantageous.

2.3.1 EFFECTS OF AMBIENT CONDITIONS

The ambient conditions, which include ambient temperature (AT), atmospheric pressure (AP), and relative humidity (RH), as well as the turbine inlet temperature (TIT) and exhaust steam pressure (also known as vacuum, V), are important factors that impact the performance of the CCPP. The system's target variable is the electrical power produced by the gas and steam turbines, and these parameters are its input variables. (2014) Tüfekci.

According to Thamir et al. (2013), the ambient temperature, compression ratio, and turbine inlet temperature all affect a combined cycle gas turbine's overall performance. These variables have an impact on power output, heat rate, and overall thermal efficiency. THERMOFLEX software was used in his study to create a thermodynamic model on an actual combined cycle gas turbine. The findings showed that when ambient temperature increased, a combined cycle gas turbine's overall efficiency and power output decreased because the gas turbine's air compressor required more power as a result. While the overall efficiency of a combined cycle gas turbine increases with an increase in compression ratio up to 21, after which it decreases, the combined cycle's total power decreases as the compression rate increases.

2.3.2 CONSIDERATIONS FOR BETTER BLADE COOLING TECHNOLOGIES

Better blade cooling technology in the topping cycle and improved heat recovery in the bottoming cycle are two ways that combined cycle performance is being improved. It may be possible to raise the gas turbine and steam turbine's turbine inlet temperatures (TITs) with advancements in blade cooling technologies.

Senthil and colleagues (2012) examined a combined cycle gas turbine that used transpiration cooling and varied the gas/steam turbine inlet temperatures (TITs). In this study, the model simulation for TIT varied from 1,600 to 1,900 K, and the performance of a combined cycle with dual pressure heat recovery steam generator was assessed for various Cycle Pressure Ratios (CPRs) varying from 11 to 23. The findings showed that for each pressure ratio, TIT increases cycle efficiency as well as specific work. An ideal pressure ratio for cycle efficiency and particular work exists for every TIT. At a TIT of 1,900 K and a maximum steam temperature of 570 ºC, the CPR of 23 exhibits the best cycle performance, yielding a cycle efficiency of 60.9% and a net specific work of 909 Kj/Kg.

2.3.3 OPTIMUM HEAT RECOVERY CONSIDERATIONS

Making the best use of gas turbine exhaust heat in the steam cycle to produce the most steam turbine output is one of the major challenges in designing a combined cycle power plant. The majority of combined cycle developers overlook the importance of the heat recovery steam generator, which has a significant impact on the combined cycle power plant's overall performance, in favor of the gas turbine output.

A study by Kumar et al. (2007) examined the best ways to use recovered heat from flue gas in various configurations of single- and dual-pressure heat recovery steam generators. A parametric analysis of the combined cycle efficiency was conducted using the first and second laws of thermodynamics. The article comes to the conclusion that for optimal heat recovery from the heat recovery steam generator in a dual cycle system, high pressure steam turbine pressure must be high and low pressure steam turbine pressure must be low.

2.3.4 EFFECTS OF PART LOAD CONDITIONS

Since gas turbines typically operate at part load conditions for a significant portion of their lifetimes, attention must also be paid to the off-design performance of these machines in addition to their design efficiency. Power reduction typically results in a decline in performance, regardless of the specific configuration of the gas turbine. In order to increase overall fuel economy, attention must be paid to the task of improving part load performance.

An intermediate recuperated combined cycle gas turbine (IRCCGT) system was taken into consideration by Yongyi et al. (2020) in an effort to enhance the performance of the CCGT under part load conditions. The goal of the design was to prevent the turbine inlet temperature from falling as the load is reduced. Between the third and fourth turbine stages, a heat exchanger was erected, and compressed air was heated using the exhaust from the third turbine stage. The findings showed that until the gas turbine part load was reduced to 49% of the design value, the TIT could be kept at that value. The minimum recuperative mass flow rate had little effect on system performance, according to the results, but the pressure drop in the recuperator had a significant impact.

In 2009, Miroslav examined the part load conditions of a recently constructed 80 MW combined cycle in Slovakia to determine potential fuel savings. Three potential solutions were identified based on the plant's online monitoring data: condensate preheating by turning on the hot water section that is currently idle, altering the steam condensing pressure regulation strategy, and—most importantly—gas turbine inlet air preheating. According to his research, higher inlet air temperatures do not impact gas turbines operating at part load; instead, they raise the amount of high pressure steam in the HRSG, increasing the power output of the steam turbine and requiring less fuel to burn in the combustion chamber to produce the same amount of power. The study also shows that there is a chance to reduce the fuel consumption of the power plant by 2% by implementing all three recommendations at the same time, albeit at a small initial cost in capital.

2.3.5 THERMODYNAMICS AND EXEGY ANALYSIS

The design of power plants primarily relies on the first law of thermodynamics, which emphasizes energy efficiency. The quality of energy produced is a major concern in power plant analysis nowadays. We call this quality "exergy." According to Ighodaro and Aburime (2011), energy is the maximum amount of work that can be obtained from the movement of materials and energy that are brought into equilibrium with the environment. Since energy in a system is typically destroyed rather than conserved, optimization is at the core of an energy analysis. This destruction is a measure of irreversibility. Points where plants lose the most energy can be identified when energy analysis is done correctly.

Khaldi and Adouane (2011) carried out exergy analysis on a gas turbine power plant using a flowsheet program called “Cycle-Tempo”. Their research revealed that the performance of the gas turbine decreased under high ambient temperatures which agrees with the investigations carried out by Thamir et al (2013). The exergy analysis indicated high losses occurring in the combustion chamber and the losses increased with an increase in ambient temperature. The analysis also revealed that preheating the air before reintroducing into the combustion chamber as with a regenerative gas turbine had a considerable higher impact on the plant efficiency. Hence exergy destruction in a gas turbine is important when considering improving the efficiency of the gas turbine power plant.

A proposed trigeneration, which combines a gas turbine, a heat recovery steam generator (HRSG), and a vapor absorption refrigeration system, was the subject of analysis by Khaliq (2009).

According to his research, the pressure ratio and turbine inlet temperature (TIT) had a major impact on the amount of energy destroyed in the combustion chamber and HRSG, but the pressure drop and evaporator temperature had no effect at all. Significant exergy destruction occurs in several vapor absorption refrigeration cycle and HRSG components due to process heat pressure and evaporator temperature. According to study by Khaliq (2009), the combustion and steam generation processes destroy the most exergy, accounting for more than 80% of the total exergy destroyed in the system as a whole.

2.4 REVISION OF PAST RELATED LITERATION ON COMBINED CYCLE INTEGRATION WITH RENEWABLE ENERGY

As mentioned in section 1.1, due to the adverse nature of fossil fuels, more research has been made on how to increase the amount of power that can be obtained from these natural resources. Some of these natural resources include solar, carbon capture, wind, hydro, biomass, geothermal, tidal energy etc.

For locations with high average direct normal radiation (DNI), these are suitable locations for concentrated solar thermal (CSP) technology. Integrated solar combined cycle system (ISCC) is a technology that integrates solar thermal energy into a combined cycle power plant system. One of the challenges of integrated solar thermal is solar fluctuations, which often lead to low thermal performance of ISCC systems when DNI is low or non-existent. To meet this challenge, ISCC is often equipped with thermal energy storage systems (Zhen and Liqiang, 2020).

Under various DNI circumstances, Zhen et al. (2020) investigated the ISCC system using a novel operating strategy with a variable integration mode. EBSILON software was used in the development of the ISCC system model. An extensive analysis has been conducted on the ISCC system's performance when utilizing the new operating strategy. The findings indicate that, compared to the conventional ISCC system, the ISCC system employing the new operating strategy has an annual solar power efficiency of 19.08 percent, which is 1.1% higher. The larger the low value DNI distribution ratio, the greater the thermodynamic performance advantage of the ISCC system with the optimization strategy, because of the varying DNI distribution throughout the year.

Wadah et al. (2023) investigated the integration of the solar power tower and the waste heat recovery process of the gas turbine cycle in the Al Qayara power plant in Iraq. To support the installation of an Integrated Solar Combined Cycle (ISCC), which makes use of concentrated solar tower technology, a thermos-economic analysis was conducted. The findings showed that the investigated power plant has a 561.5 MW total generating capacity, of which 68 MW comes from CSP and 130.4 MW from gas turbine waste heat recovery. The gas turbine cycle's waste heat recovery increased thermal and energy by 10.9 and 10.61%, respectively, and the total production cost was 11.43$/MWh.

For ISCC, the total unit cost of production is 12.39$/MWh, with increases in thermal and exergy efficiencies of 17.96 and 17.34%, respectively. September had the lowest monthly capacity of the integrated solar combined cycle, which was approximately 539 MW. April had the highest monthly capacity, which was approximately 574.6 MW.

In 2014, Nipu Datta conducted an experiment on a miniaturized solar power station to improve the efficiency of a solar power plant using a combine cycle system. The study established a model for a solar panel system based on this combine cycle system. The goal of the technique is to increase the solar power plant's overall efficiency by introducing a combine cycle, which uses concentrated solar radiation heat to produce electricity. This was studied, and a miniaturized solar power station with six solar panels and individual sun tracking systems was designed and used. According to the paper's conclusion, the solar power plant was designed with a solar tracker to track the sun's position in order to provide enough sunlight for the solar panels. It also has three microcontrollers to regulate each of the six solar panels using solar light. The combined cycle heats up diethyl ether, which evaporates at 34.6oC and flows through a turbine to generate electricity, increasing the solar station's efficiency by 8.48%. The model is useful for producing more electricity at the same location by merging engines and solar systems.

Despite the fact that natural gas is seen as a cleaner fuel than coal, it is important to remember that the Natural Gas Combined Cycle (NGCC) produces a significant amount of CO2 at the plant location, which contributes to global warming. Abdulla (2012) suggested that the use of amine in CO2 Capture and Sequestration (CCS) reduce global warming because there is enough CO2 in the atmosphere. He noted that the CCS system's implementation results in a 15-20% increase in energy consumption and came to the conclusion that waste utilization and the integration of processes can be used to increase energy efficiency by revealing and contrasting a parameter that is defined as the power gain to waste heat ratio. Additionally, a new integrated CCS configuration is proposed, which involves the NGCC removing CO2 and the CO2 cycle, using parameters like the Organic Rankine Cycle (ORC), NGCC, CO2 compression cycle, and CO2 liquefaction cycle. One of the suggested CCS configurations that makes use of the waste heat that is already available in absorption chillers is the efficiency improvement.

One of the most significant issues is that noncombustible gases with low calorific value make up a significant portion of the contents of biogas, depending on the feedstock and the generational cycle of the gas. Researchers Sayed and Mazian (2014) looked into potential avenues for the growth of biogas combustion in combined heat and power generation. They came to the conclusion that, because biogas conventional combustion is not sufficiently sustained, adding hydrogen to the biogas components could improve combustion stability and make it more affordable to use modification techniques like cryogenic and membrane.

2.5 CONCENTRATED SOLAR POWER

Nigeria is a good location for Concentrated Solar thermal Power (CSP) technology as there is high average Direct Normal Irradiation (DNI), and as such has been considered as a retrofit for boosting of power in the combined cycle which will be analyzed. CSP uses mirrors or lenses and in some situations tracking systems to concentrate a large area of sunlight onto a smaller area usually tubular receiver to heat a fluid in a collector at high temperature.

There are four available CSP technologies:

1. Parabolic Through collector

2. Parabolic dishes
3. Solar towers
4. Linear Fresnel collectors

Of all the available CSP technologies, Parabolic through collectors are the most popular, developed and commercialized. Currently, about 90% of CSP power plants around the world use parabolic through concentrators technology (Shahin M.S et al., 2016).

Mirrors, tube heat receivers, and structural support make up the parabolic through concentrators, as seen in fig. 1 below. A parabolic-shaped flat sheet of reflective material is used to create parabolic mirrors, which focus sunlight onto a focal receiver line. The absorber tube within the glass envelope makes up the receiver line. In order to convert solar energy into steam or thermal energy storage, heat transfer fluid is pumped through the receiver tubes. The majority of parabolic trough concentrators on the market today use synthetic oil in the tubular receiver for heat transfer. Up to 400°C can be operated at with synthetic oil (Sorour et al., 2020).

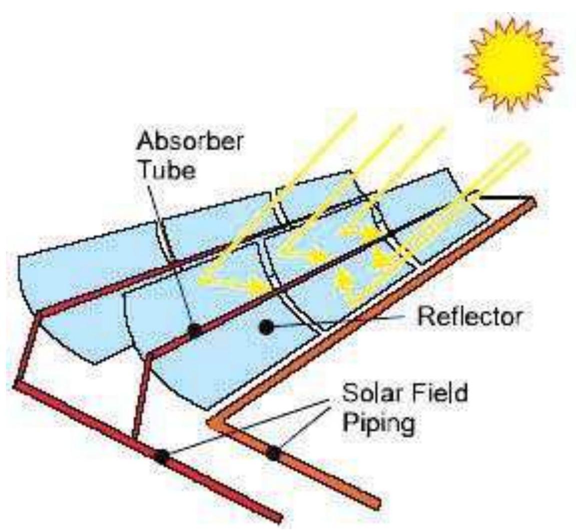


Figure 2.8. Parabolic Through Collector (source www.hkengineer.org.hk)

Andrea (2010) carried out analysis on solar retrofit in combined cycle power plant, in his paper, he modeled single, dual and triple pressure combined cycle power plant, then introduced solar boosting by modeling the parabolic through collector. Results revealed that for a 9FA three pressure CCPP design strictly limited by the ISO case, approximately 7 additional MW can be obtained at 30oC. Also Parabolic Through Concentrators (PTC) technology has a better optical efficiency when comparing the amount of radiation absorbed per unit of surface to Parabolic Pre-stressed Concentrators (PPC).

Paul Breeze (2016) wrote a paper on adding supplementary heating to a Concentrated solar power plant to create a hybrid plant with a configuration of using Parabolic trough and Linear Fresnel collectors systems in which the heat collected from both systems were used to raise steam and drive the steam turbine generator. He also pointed out that using Solar towers the plants can achieve concentration ratios of up to 1000.

According to J.D. Nikon (2010), in his publication on Solar Thermal collection technology for electricity generation in North West India, Parabolic Dish Reflectors (PDRs) have the highest potential. In fact, one PDR currently holds the world record for solar to electrical efficiency at 31.25%. It should be noted that this technology achieves optical efficiencies of up to 94% and concentration ratios of 500–1000 while enabling the maximum capture of solar energy.

2.6 MODELLING SOFTWARE

In an attempt to ascertain parameters such as mass flow rates of fuel and air, compressor discharge temperature and pressure, turbine inlet pressure and temperature for both design and off design conditions, numerous researchers have looked into laborious iterations and thermodynamic model equations. Since modeling software simplifies these laborious iterations, modeling and simulation are considered preliminary steps towards idea approval. To precisely evaluate the plant's performance in relation to changes in part load and ambient temperature, a detailed model must be executed for the combined cycle under investigation. (2020, Najja).

For modeling analysis and power plant simulations, there are plenty of modeling software options available. APROS, Autodynamics, EBSILON professional, HYSYS, gProms, SIMODIS, PowerSim, MMS, ProTRAX Aspen, Autodynamics, and TRNSYS are a few of these modeling programs. For this project, professional software from EBSILON has been selected. Power plant process cycle usage for steady-state and semi-dynamic simulation, as well as plant parameter advancement measures, is extensive when using Ebsilon Professional. For steady-state estimation, the Ebsilon professional programming conditions hold true (Dahash, 2019).

One of the most widely used mass balance and energy computation programs in European countries is EBSILON Professional (Gülen, 2019). The modeling software is compatible with Microsoft environments and exhibits high intermingling solidity and computation speed. Every prerequisite required for the analysis is present in EBSILON. The power plant analysis research projects that have made use of EBSILON software include the following: Andrea Miguez's (2010) study on solar retrofit for combined cycles, Investigation of the thermodynamics of a combined combined cycle gas turbine (CCGT) and high temperature nuclear system by Jaszczur and Dudek (2019); Wallentinen (2016) on concentrated solar power gas turbine with thermal storage; and Wojcik and Wang (2018) on steam power plant optimization for energy for an efficient process of a post-combustion of CO2 capture plant. We used it in our research because it is the best software available for modeling and simulating power plants, as other researchers have demonstrated its effectiveness in doing so.

A real power plant, the SGT5-2000E Gas Turbine in Azura, Benin City, was modeled by Egware (2021). The error percentage ranged from -3.13% to 0.88% for design conditions, while -3.24% to 2.66% for off-design conditions, according to validation results using design and actual operating data by applying the error percentage analysis. The results were also consistent with the actual installed operating data of the plant.

On the French island of La Reunion, the 60 Mw twin thermal power plant of Albioma Bois Rouge was modeled, calibrated, and simulated by Romuald et al. in 2023. Utilizing the EBSILON software to compare the primary process parameters for the plant's conversion from coal to biomass in order to determine plant performances in terms of boiler efficiency, overall power efficiency, and specific fuel consumption.

EBSILON professional is the modeling software of choice for this study because it has been used numerous times in the past and has continuously shown to be an accurate, precise, and powerful tool for the design, modification, or retrofitting of power plants with successful applications and further development (Matjanov, 2020).

2.7. SUMMARY OF LITERATURE

In summary the literature review reveals that

- The combined cycle power plants improves the output for power generation compared to simple power plant such Brayton cycle and Rankine cycle which is majorly in use in Nigeria.
- Ambient conditions such as Temperature, pressure and humidity affect thermal efficiency, power output and the heat rate of the combined cycle
- Higher compression ratio increases the overall efficiency of the CCGT
- Exergy destroyed in combustion chamber and the HRSG are affected by the pressure ratio and Turbine inlet temperature

- Natural gas is preferred for being a cleaner fuel however due to global warming and pollution released from plant site into the atmosphere adoption and integration of renewable energy has been considered

Nigeria is a good location for Concentrated Solar thermal power due to the presence of abundant sunlight throughout the year considering its location near the equator and as such is suitable for the use of solar retrofits for the boosting of power in the CCPP. Solar retrofits offers a solution as a clean renewable energy source and as a means of increasing the power generated in Nigeria and as such will be the analyzed in this research paper.

CHAPTER THREE
METHODOLOGY

The purpose of this research is to see how a combined cycle gas turbine reacts to changes in surrounding conditions, including temperature, relative humidity, mass flow, and part load variations. Next, watch how it reacts when the sun boosts.

The gas turbine model description based on the simple cycle performance guarantee, thermodynamic equations, modeling and simulation of the GT13E2 model, model validation, and simulation analysis form the basis of this project's methodology.

3.1 DATA COLLECTION AND OVERVIEW OF AFAM VI POWER STATION

Data collected from AFAM VI power station was through direct observation from the monitoring screen of the human machine interface (HMI), log books and manufacturer's manual.

The gas turbine combined cycle modeled in this project is located in AFAM VI power station in Afam, Okoloma, Rivers State, Nigeria. The power plant consists of 3 Gas turbine rated 150MW and a steam turbine rated 200MW. Each of the gas turbine is attached to its respective heat recovery steam generator and all three combines and provides steam for steam turbine. The power plant generates a total of 650MW.

The plant was commissioned in 2008. It is one of the four Afam gas plants in Rivers State, including Afam (FPL) power station, Afam IV power station, and Afam V power station.

3.2 THERMODYNAMIC OPERATIONAL PRINCIPLE OF COMBINED POWER PLANT

The installed GT13E2 gas turbine power plant works under the principle of the Joule Brayton cycle. The schematic and temperature-specific entropy (T-S) diagrams for the combined power plants are illustrated in Figure 1 and 2 respectively. Ambient air is taken into the compressor at **1**, after the air has been compressed it then enters the combustion chamber at **2** where it is mixed with fuel and combustion takes place, the exhaust gas at **3** then enters the turbine and is expanded. The expansion process rotates the shaft which provides the mechanical energy required to generate electricity in the generator. It should also be noted that considerable part of this mechanical energy generated by the turbine is used to run the compressor and this compressor takes up about 60% of the generated mechanical energy hence the efficiency of the gas turbine operating in simple cycle will always be less than 40%.

After expansion in the turbine, the flue gas is exhausted at **4** where it enters into the Heat Recovery Steam Generator which consists of a high pressure and low pressure evaporator, a low pressure economizer, 3 high pressure economizers, a low pressure superheater and 3 high pressure superheaters. After the flue gas supplies the steam with heat, it is then exhausted to the atmosphere through the stack. The steam picks up heat from the flue gas in the heat recovery steam generator and then drives the High pressure steam turbine at **5** and the low pressure steam turbine at **7** which in turn drives the generator providing electricity. The steam is then exhausted into the condenser at **8,** the steam condenses into water at **9** and is then pumped back into the Heat recovery Steam Generator repeating the cycle.

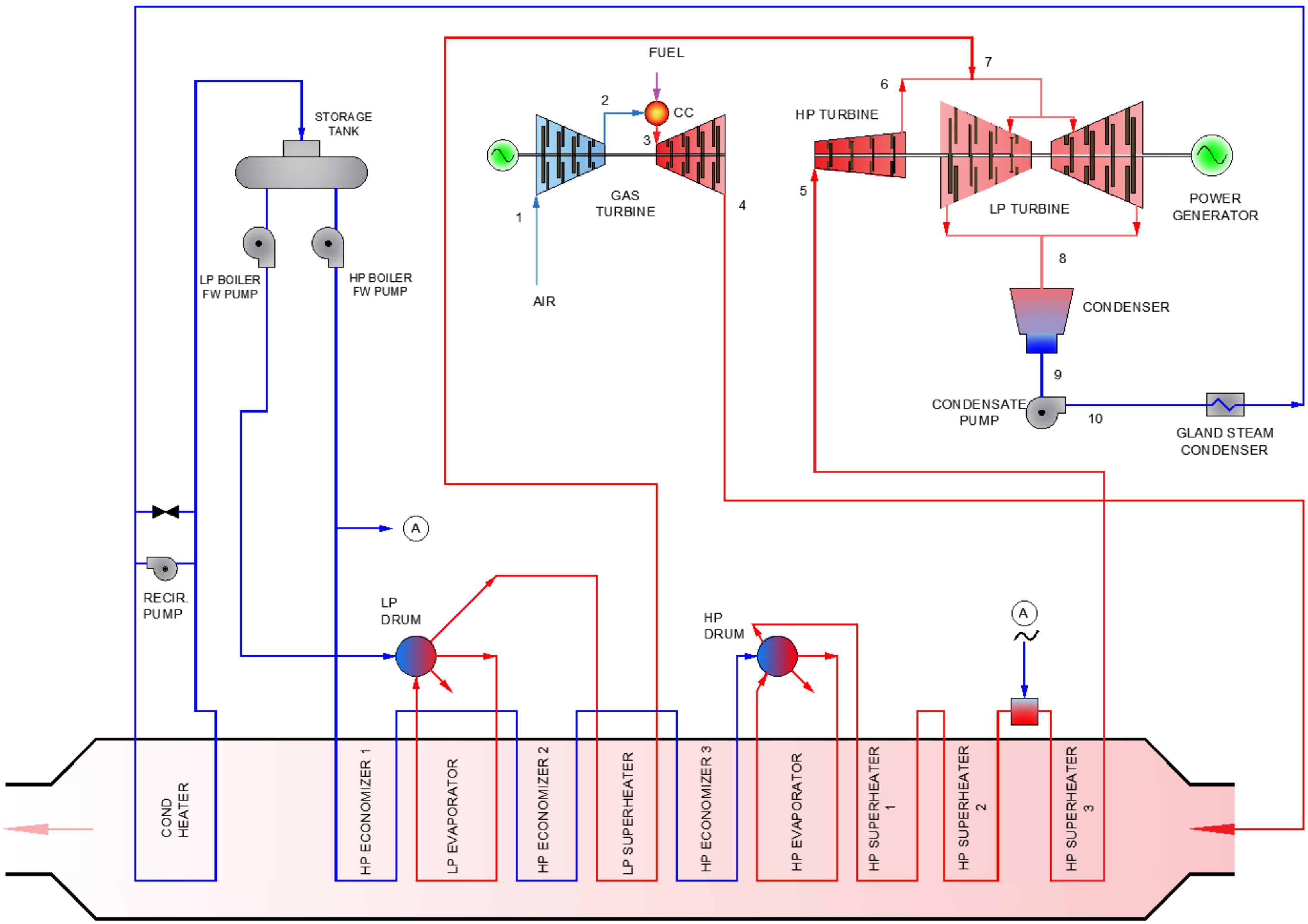


Figure 3.1. Schematic Diagram AFAM VI GT13E2 Combined cycle power plant

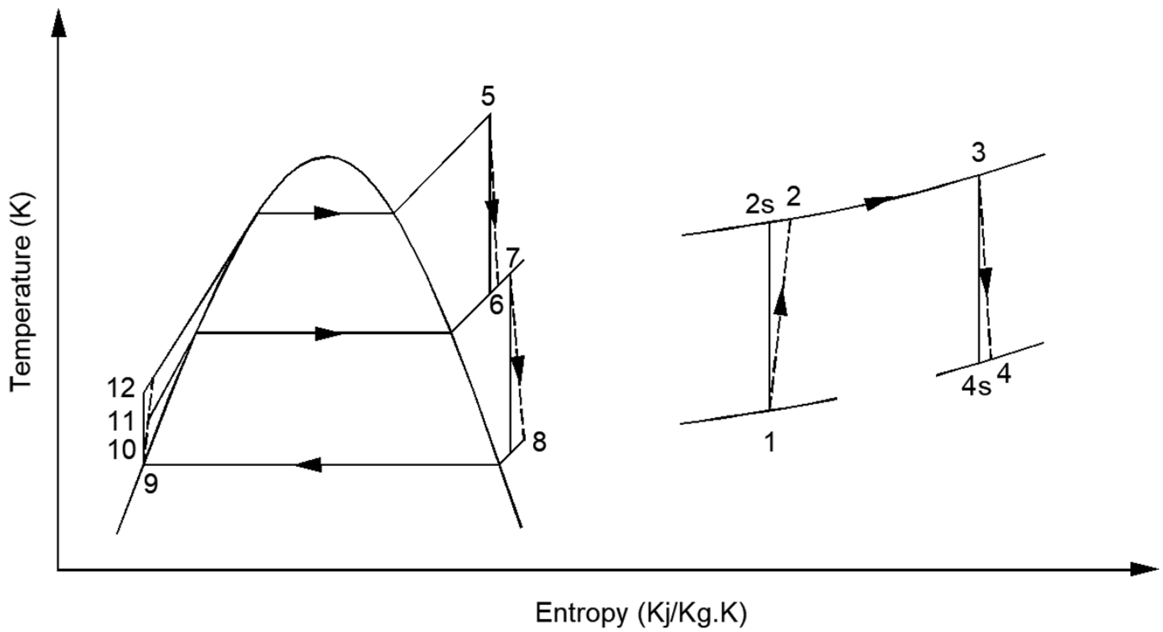


Figure 3.2. Simple Brayton cycle and Rankine cycle TS diagram

3.3 MODELLING EQUATIONS

3.3.1. Compressor and Pump

Compressors and pumps operate using the same physics. The fluid is brought in from a low pressure and delivered at a high level.

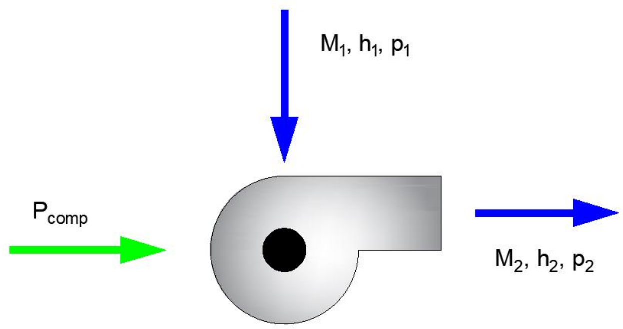


Figure 3.3. Fluid properties of pump and compressor

- Energy balance: $M_2h_2 - M_1h_1 = P_{comp}$ (1)
- Outlet enthalpy h_2: $h_2 = h_1 + \frac{1}{\eta}.(h_{2s} - h_1)$ (2)
- Mass balance: $M_1 - M_2 = 0$ (3)

3.3.2. Steam Turbine and Gas Turbine

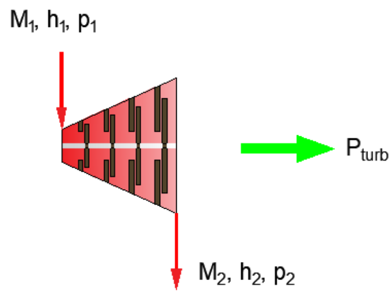


Figure 3.4. Steam turbine and gas turbine fluid properties

- Energy balance: $M_1 h_1 - M_2 h_2 = P_{turb}$ (1)
- Outlet enthalpy h2: $h_2 = h_1 - \eta.(h_1 - h_{2s})$ (2)
- The relationship between inlet and outlet pressure in off-design is described by the law of ellipses' of Stodola:

$$\left(\frac{m_1}{m_{1N}}\right)^2 = \frac{p_1}{p_{1N}}.\frac{v_{1N}}{v_1}.\frac{1-\left(\frac{p_2}{p_1}\right)^2}{1-\left(\frac{p_{2N}}{p_{2N}}\right)^2} \tag{3}$$

3.3.3. Combustion chamber

For the furnace a stoichiometric combustion calculation is done. For the energy balance the lower heating value h_{HW} of the fuel and its composition must be known.

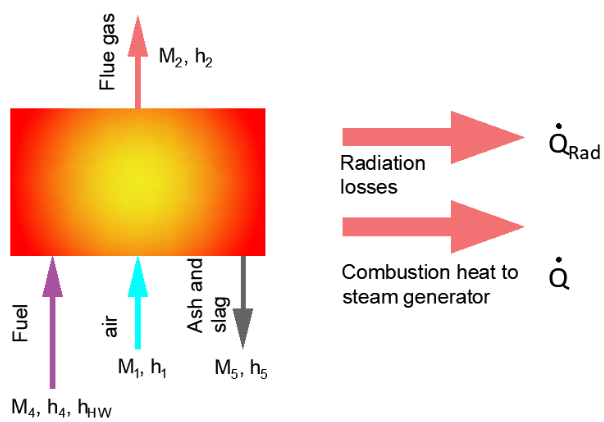


Figure 3.4. Fluid properties of the combustion area

- Mass balance: $M_2 - M_4 - M_1 + M_5 = 0$ (8)

- Energy balance: $\dot{Q} = M_1h_1 + M_4h_4 + M_4h_{HW} - M_2h_2 - M_5h_5 - \dot{Q}_{rad}$ (9)

3.3.4. Heat Exchanger

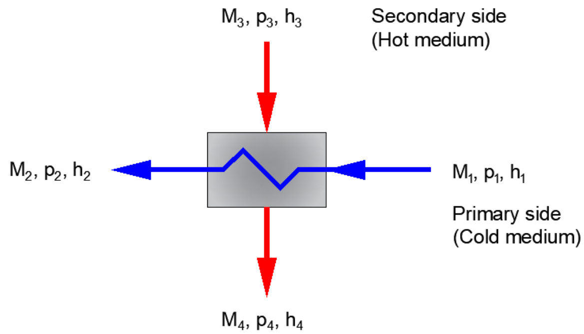


Figure 3.5. Fluid properties of a heat exchanger

Enthalpy: $M_2h_2 - M_1h_1 = Q$ (5)

Enthalpy: $M_3h_3 - M_4h_4 = Q$ (6)

The heat rate of the gas turbine was computed using Equation (7)

HR = 3600/ η_{net} (7)

Where η_{net} is cycle efficiency for gas turbine in simple cycle.

3.4 MODELING AND SIMULATION WITH EBSILON PROFESSIONAL

EBSILON Professional is the modeling software, as was indicated in section 2.5. "Energy Balance and Simulation of the Load Response Power Generating or Process Controlling Network Structures" is a term that can be shortened to "EBSILON." This tool is useful for stationary simulation of various thermodynamic power cycles and processes.

A program for calculating mass and energy balance in thermodynamic cycles is called EBSILON. The performance of a combined cycle power plant under design and partload conditions can be simulated using EBSILON (Dr. Hans-Peter, 2012).

Standard components, which are used to model typical power plant components, form the basis of the EBSILON model framework. Also, programmable parts with user-defined behavior for simulating intricate power plant operations are useful tools.

The International Association for the Properties of Water and Steam, or IFC67 steam table, is the source of data used by Ebsilon along with cp-polinomial for flue gas and air.

EBSILON is a variable program system that uses a closed solution based on a sequential solution method to balance all power plant circuits that occur (Andrea, 2010). Objects are used to construct the cycles. The many object kinds include: Components, Text fields, Pipes, Macros (like the gas turbines in the library), Value crosses; and Graphical elements

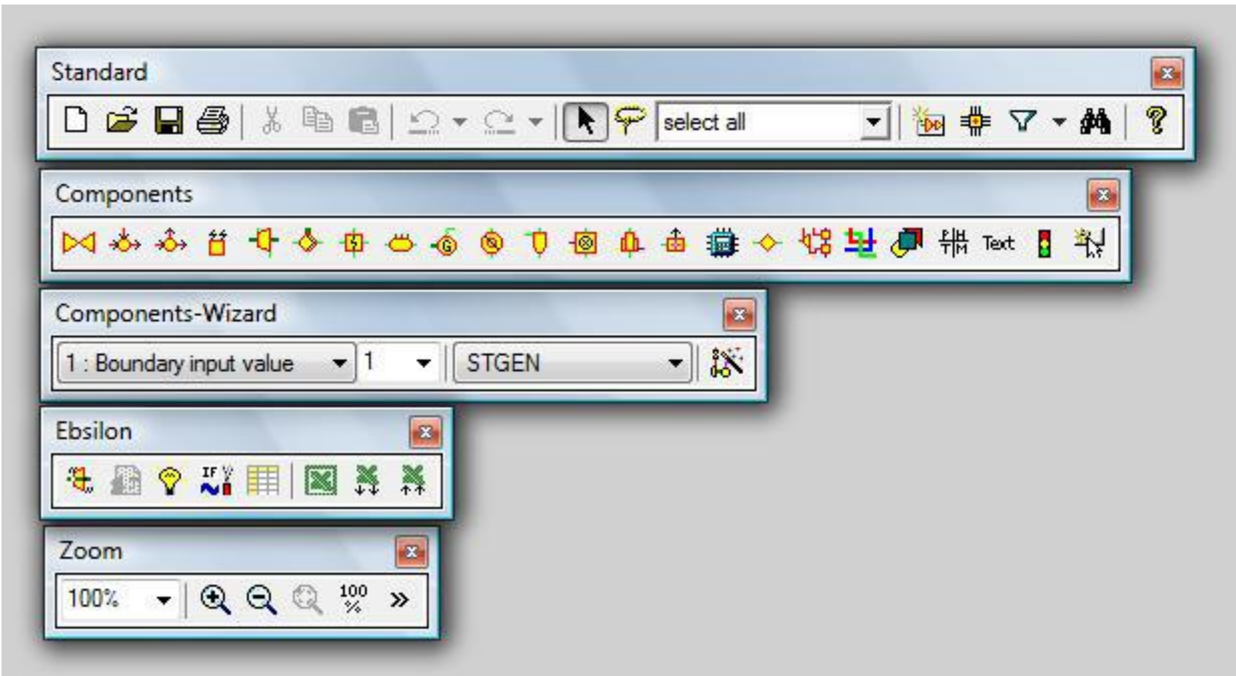


Figure 3.7. Basic control elements and tool bars of EBSILON software (Andrea, 2010)

OPERATING PARAMETERS OF THE AFAM VI POWER PLANT

In order to model the AFAM VI power plant, we first start from the modelling of the gas turbine. The performance guarantee for simple cycle of the GT13E2 as obtained from AFAM VI power plant is provided in the table below.

NET ELECTRICAL OUTPUT GUARANTEE		**Unit**
Guaranteed Unit Net MCR Output Simple Cycle	146.882	MW
NET HEAT RATE GUARANTEE		
Guaranteed Unit Net Heat Rate (Simple Cycle)	10,509.30	KJ/kWh
AMBIENT CONDITIONS		
Ambient air temperature	30.9	ºC
Ambient air pressure	1.013	Bar
Relative air humidity	70	%

OPERATING CONDITION		
GT load	100	% (Base load)
Gas Turbine Exhaust Temperature	531.3	ºC
Exhaust mass flow rate	503.1	Kg/s
Pressure Ratio	23	-
Lower Heating Value of fuel	48462	KJ/Kg

Table 3.1. Guaranteed performance for GT13E2

The following presumptions were made when modeling the power plant:

i) The simulations were run at steady state.

ii) The momentary effects of starting and stopping during operation were disregarded

iii) The models' various component pressure drops were taken into account as the nominal pressure drop.

The air compressor, combustion chamber, gas turbine, generator, steam turbine, and dual pressure heat recovery steam generator are the parts of the GT13E2 combined cycle model. The components of the dual pressure heat recovery steam generator are as follows: three high pressure economizers, one high pressure evaporator, one high pressure superheater, an economizer ahead of a storage tank, and one low pressure evaporator. EBSILON professional's component bar and component wizard were utilized to model the GT13E2 combined cycle using different components.

The data available in Table 1 from the power plant were used to model the design performance of the GT13E2 gas turbine. Component 1 (boundary input value) was selected to set the ambient conditions of air and the temperature of fuel, component 46 (measured value input) was selected to set the turbine exit mass flow, temperature and pressure, component 46 was again selected to specify the relative humidity of air at inlet to compressor.

In the combustion chamber, under method for specification of air and fuel flow (FALAM), set M1, M4 and M5 was selected so as to calculate and obtain the mass flow of air under the design consideration. The pressure ratio was set in the gas turbine and under type of characteristics (FCHR), power specification was selected and a component 33 (general input value) was selected to specify the design power.

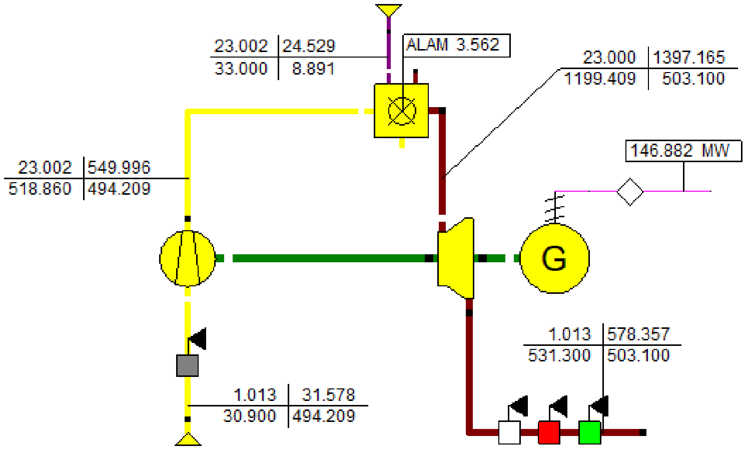


Figure 3.8. GT13E2 gas turbine model

Once the mass of air was obtained, in the combustion chamber under FALAM, “set M1 and calculate the other flow using ALAMN” was selected. In the gas turbine under FCHR, “ETAI/ETAIN” was selected and the component 33 selected to specify power was removed. Component 46 was selected to input the mass flow of air and the temperature and mass flow at exit of turbine were removed and the simulation was carried out again. Once the initial results for the exit were obtained then the model is ready attachment of the heat recovery steam generator.

Moving on to modeling of the combined cycle, in the design mode, all the temperatures and mass flows were specified the respective positions as shown in figure 11 below by selecting component 46, they were all set to “on in design mode and off in off design mode” under FFU. The respective pressures in the low pressure turbine and the high pressure turbine were specified in their respective turbines. The evaporators were set to “steam production M2 (externally)” under FSPECD, while FTAPPN was set to “T1 given externally”. In the heat economizers and superheaters, under FSPECD “Both cold and one hot stream temperature given” was selected. Due to the fact that the plant consists of 3 gas turbine and 3 HRSG’s, the multiplier component was used to model this fact. A multiplier of 3 was used to indicate where the fluid or power line mixed and a multiplier of 1/3 was used to indicate where the fluid were spitted to the various HRSG. The model was simulated and nominal values were obtained.

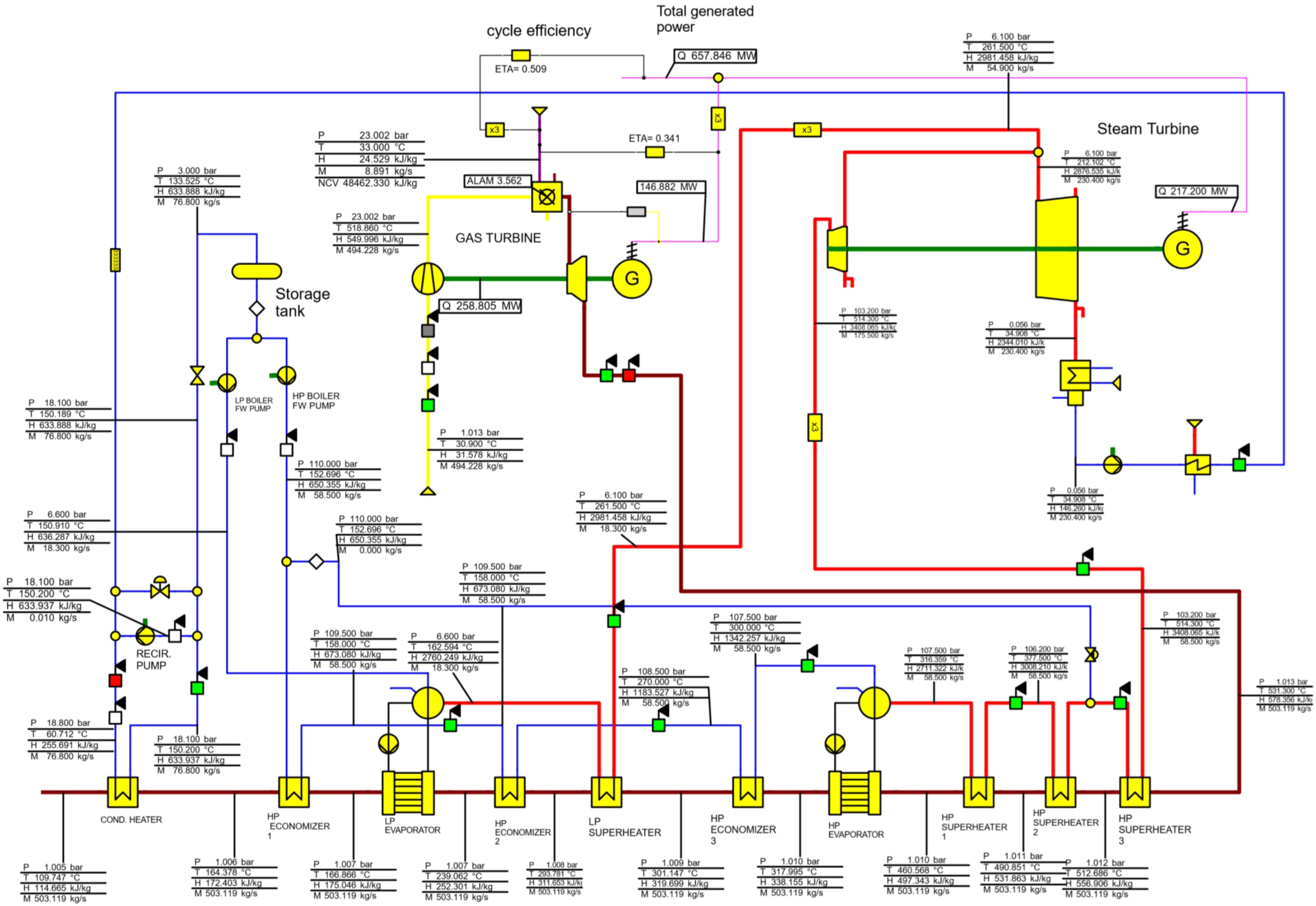


Figure 3.9. Combined Cycle Gas Turbine (Design mode)

3.5 GT13E2 COMBINED CYCLE GAS TURBINE OFF DESIGN MODELLING

After the design mode was confirmed to be working properly without any errors, the off designs calculations were then modeled, a new profile was then created and renamed to off design. In this new profile, first, in all the super heaters, economizers and evaporators, under FSPECD they were set to "effectiveness given". Doing this would enable the effectiveness to be specified based on nominal values obtained from design calculations.

In off design mode, the plant operates at conditions different from design specification, for example, there can be changes in ambient air conditions such as temperature and humidity, in other situations, the gas turbine may be operating in part load. All these situations are modeled as the situation requires.

3.5.1 VARIATION OF AMBIENT AIR TEMPERATURE

In the gas turbine plant, at various times of the year, ambient temperature changes, this change in ambient temperature is important to take note of because when ambient temperature changes, so does its relative humidity and mass of air. This is because as temperature increases, the density of air decreases and hence the mass flow into the compressor of the gas turbine reduces. The volume flow rate of the compressor is constant once it has been designed, the mass of air that can enter the gas turbine is constrained to this volume flow rate. (Egware, 2020).

The various mass flow rate, density and relative humidity is calculated using the following relations;

$$\rho_i = \rho_N \frac{T_N}{T_i} \tag{1}$$

$$\dot{m}_i = \dot{m}_N \frac{\rho_i}{\rho_N} \tag{2}$$

Where $T_N, \rho_N, \dot{m}_N$ are the absolute temperature, density and mass flow rate in design conditions or nominal values respectively. $T_i, \rho_i, \dot{m}_i$ are absolute temperature, density and mass flow rate in off design conditions. Equations (1) and (2) was used to calculate the air density and mass flow rate respectively for different variations of ambient temperatures.

The table today shows the temperature and its respective humidity obtained from data available from the power plant.

Ambient temperature (°C)	Relative Humidity (%)
13	83
30.9	70
41	70

Table 3.2. Actual data for relative humidity and ambient temperature

$$\emptyset = \frac{ti-13}{30.9-13}(70-83)+83 \qquad (3)$$

Where t_i and Ø are the off design temperature in Celsius and relative humidity respectively.

Equation (3) is the interpolation equation used to calculate the relative humidity for temperatures between 13ºC and30.9ºC. The results of the calculation is displayed in the table below.

	T (ºC)	T (K)	Density (kg/m^3)	$\dot{m}$ air (kg/s)	Relative Humidity (%)
Design	13	286.15	1.219	525.144	83.00
	20	293.15	1.190	512.605	77.92
	22	295.15	1.182	509.131	76.46
	24	297.15	1.174	505.704	75.01
	26	299.15	1.166	502.323	73.56
	28	301.15	1.158	498.987	72.11
	30	303.15	1.150	495.695	70.65
	32	305.15	1.143	492.446	70.00
	34	307.15	1.135	489.240	70.00
	36	309.15	1.128	486.075	70.00
	38	311.15	1.121	482.950	70.00
	40	313.15	1.114	479.866	70.00
Guaranteed performance	30.9	304.05	1.147	494.228	70.00

Table 3.3. Temperature variation and its effect

3.5.2 VARIATION OF AIR FUEL RATIO

Air ratio called ALAM in EBSILON is a parameter in the combustion chamber that determines the ratio of mass of air and mass of fuel that can take part in a combustion. ALAM is the ratio of the mass of air entering the combustion chamber to the stoichiometric mass of air for the fuel taking part in the combustion. This is important because this ratio affects the maximum temperature in the combustion chamber. By increasing the air ratio, the mass of air that takes part in the combustion process increases while that of fuel reduces, this would lead to a reduction in the maximum temperature at the exhaust in the combustion chamber and subsequent reduction in the exhaust temperature of the gas turbine. The reverse occurs when the air ratio is reduced.

In part load conditions, the power required from the gas turbine is less than the design power. The modeled gas turbine operates in combined mode hence the air fuel ratio has to be adjusted such that at different part loads, the flue gas from the gas turbine is kept at the operating temperature from actual data of the power plant. This is done to ensure the HRSG operates properly as the HRSG performance is dependent on the temperature of flue gas from gas turbine.

To do this variation in the model, component 39 (controller) was used along with component 46 (measured value input) to set the exhaust temperature to the value from actual performance data. The combustion chamber under method for specification of air and fuel flow (FALAM), “set M1, M4 and M5” was selected and the controller was used to vary the mass flow rate. A range of 50% to 110% was selected and calculations were done in off design mode.

It should be noted that in peak load condition (110% of base load), since the mass of air cannot exceed the design mass flow rate hence the component 46 for exhaust temperature was turned off and the mass flow rate at entry to compressor was turned on and set to design mass flow rate while the mass of fuel was varied with the aid of component 39.

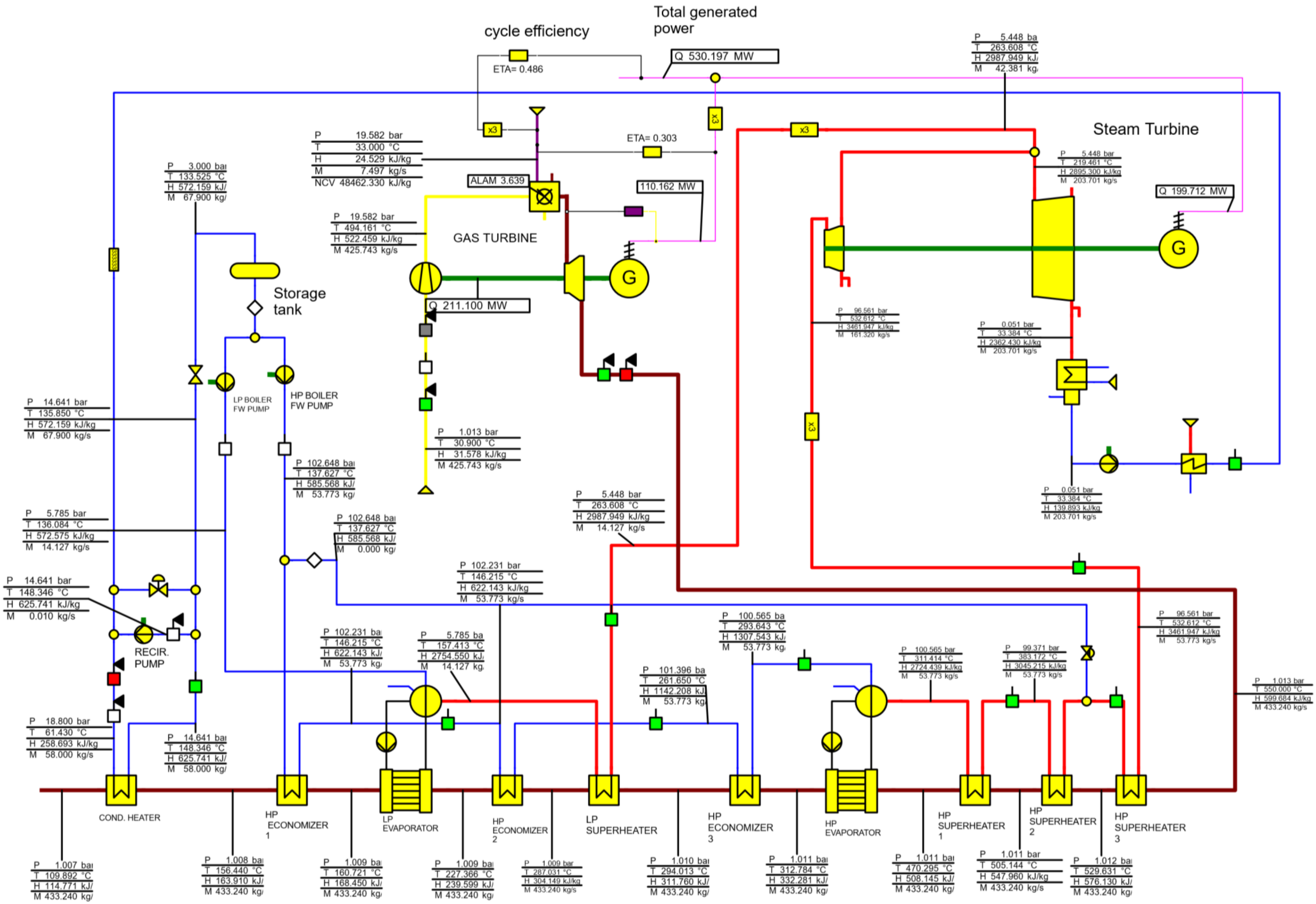


Figure 3.10. Combined Cycle Gas Turbine (75% of base load, off design mode)

3.6 VARIATION OF EFFICIENCY IN COMPONENTS

When the combined cycle operates in off design mode, the efficiencies of the various components changes due to difference in the nature of fluid during design mode and off design mode. EBSILON provides a characteristics curve to show how these efficiencies would vary due to these different conditions.

Index	M1/M1N	ETAI/ETAIN
1	0	0.5
2	0.4	0.9
3	1	1
4	1.2	1.1

Table 3.4. Variation of efficiency in Compressor with mass flow

Index	M1/M1N	ETAI/ETAIN
1	0	0.85
2	0.4	0.9
3	0.7	0.95
4	1	1
5	1.2	1.1

Table 3.5. Variation of efficiency in Gas turbine with mass flow

Index	FCHR	ETAI/ETAIN
1	0	0.98
2	0.5	0.991
3	0.6	0.993
4	0.7	0.995
5	0.8	0.9975
6	0.9	0.999
7	0.95	0.9998
8	1	1
9	1.05	0.9995
10	1.1	0.998

Table 3.6. Variation of efficiency in Steam turbine with mass flow

Index	M1/M1N	(k*A)1/(K*A)N
1	0.01	0.018463
2	0.04	0.075067
3	0.07	0.133570
4	0.11	0.191619
5	0.15	0.243225
6	0.2	0.303098
7	0.25	0.359180
8	0.3	0.412325
9	0.35	0.463079
10	0.4	0.511816

Table 3.7. Variation of Heat transfer in Steam turbine condenser with mass flow

Table 4 to 6 shows the correction curves for the relationship between the mass flow and the efficiency while Table 7 shows relationship between heat transfer and mass flow rate.

3.7 VALIDATION OF MODEL

After building the model, a comparison is made between the model and the actual installed GT13E2 combined cycle power plant performance data. Equation (4) is used to compute the percentage error in the model (Wallentinen, 2016).

$$\%\text{Model Error} = \frac{(Actual\ data - Model\ data)}{Actual\ data} * 100\% \tag{4}$$

3.8 INCOPORATION OF SOLAR BOOSTING

When the ambient temperature rises, the mass flow through the gas turbine decreases and hence the amount of energy which is transferred from the flue gas of the gas turbine to the HRSG reduces as was shown earlier in section 3.4.1. This implies that at higher temperatures the amount of steam generated will be less at higher temperatures.

Actual data from the power plant is shown in the table below

% base load	T1 (°C)	T4 (°C)	$\dot{m}_{gas}$ (kg/s)	$T_{HPinlet}$ (°C)	$T_{LP\ super\text{-}heater}$ (°C)	$\dot{m}_{HPinlet}$ (kg/s)	$\dot{m}_{LPsuper\text{-}heater}$ (kg/s)	Total steam mass flow in 3 HRSG (Kg/s)
100	13	519.4	534.7	503.1	263.2	58.9	19.3	234.6
75		550	435.6	514.3	256	54.4	14.9	207.9
50		498	399.7	490.2	250.5	42.1	13.8	167.7
100	30.9	531.3	503.1	514.3	261.5	58.5	18.3	230.4
75		550	423.3	514.3	255.4	53.5	14.8	204.9
50		511.4	381.5	502.8	248.8	42.7	13.1	167.4

Table 3.8. Actual Performance data for GT13E2

From the table above, it can be observed that the mass flow at design condition of the power plant, the total steam mass flow is 234.6 Kg/s while at the guaranteed performance; the steam flow is 230.4 Kg/s. This proves that at higher temperatures, the mass flow of steam reduces and as such the steam turbine under performs. To make up for this loss of power, we introduce a solar retrofit such that it takes in water with the mass flow of the difference in mass flow of steam at design and at 30°C and converts the water to saturated steam then this steam is mixed with the steam in the HRSG for superheating.

To model the solar boosting, an extra circuit is designed as shown in Figure 6. The water after leaving the condenser is pumped to an economizer and evaporator which takes their heating source from the parabolic through collectors. The fluid in the parabolic through concentrators is Therminol VP-1. The saturated steam is then taken to the HRSG where it mixes with the steam in the HRSG and is superheated.

The solar boosting components consists of PTC, a storage tank, pumps and a duct burner. When the power plant is approaching the hot part of the day, the duct burner raises the temperature of the fluid in the solar boosting cycle to about 380°C then is switched off. This temperature is maintained by the storage tank and the heat from the sun as the storage tank acts as a thermal storage.

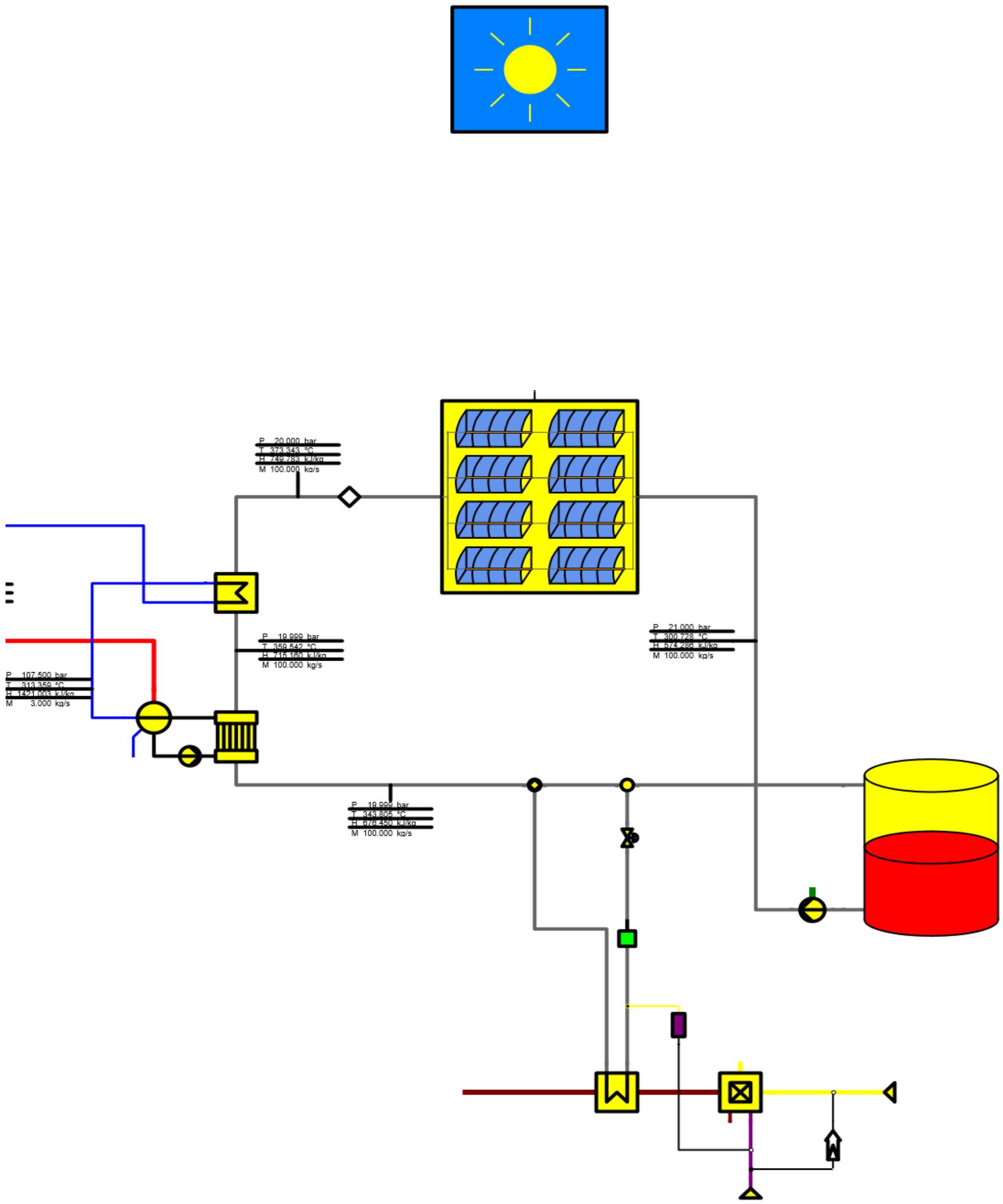


Figure 3.11. Solar boosting components.

The injection of saturated steam back into the Rankine cycle will be considered from 2 points;

1. Saturated High Pressure steam; The solar field generates saturated steam at high pressure which is injected after the high pressure evaporator as shown in figure 7.
2. Saturated Low pressure steam; The solar field generates saturated steam at low pressure which is injected after the low pressure evaporator as shown in figure 8.

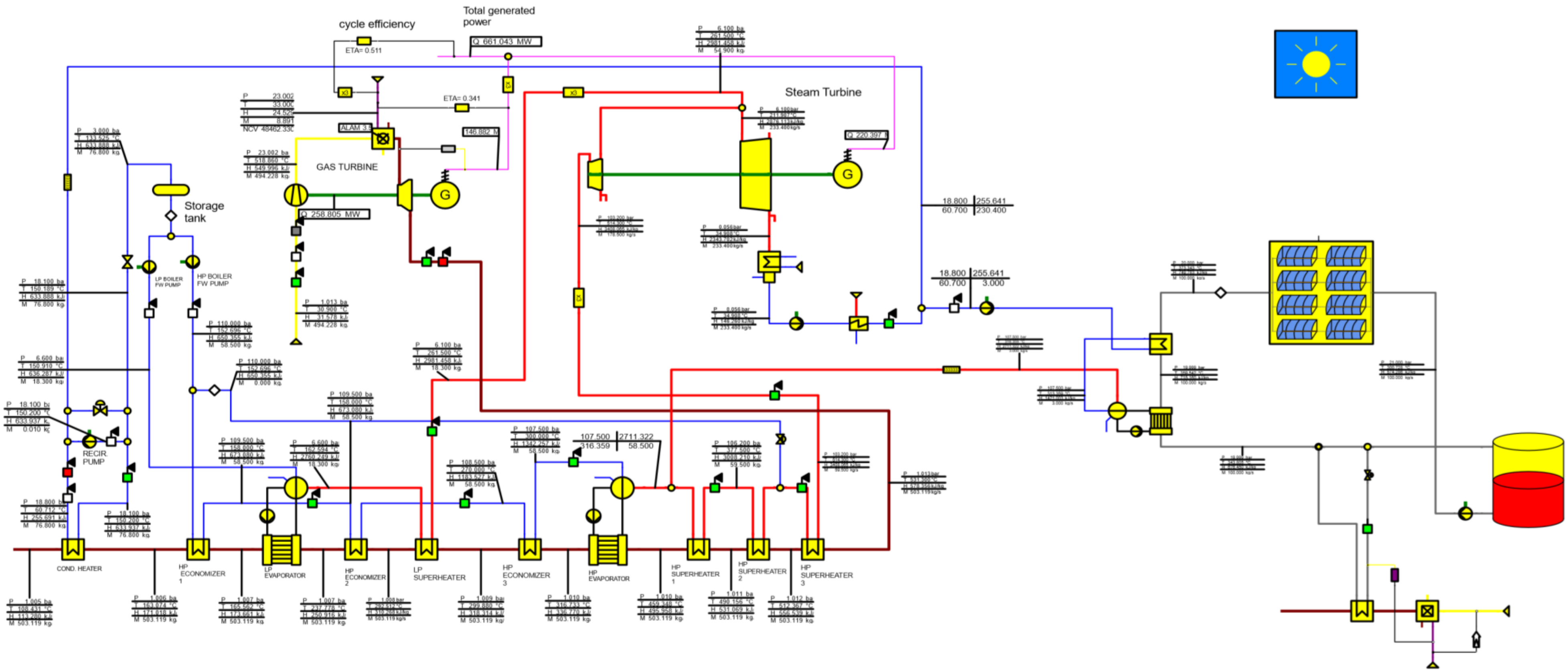


Figure 3.12. Solar boosting with saturated high pressure steam

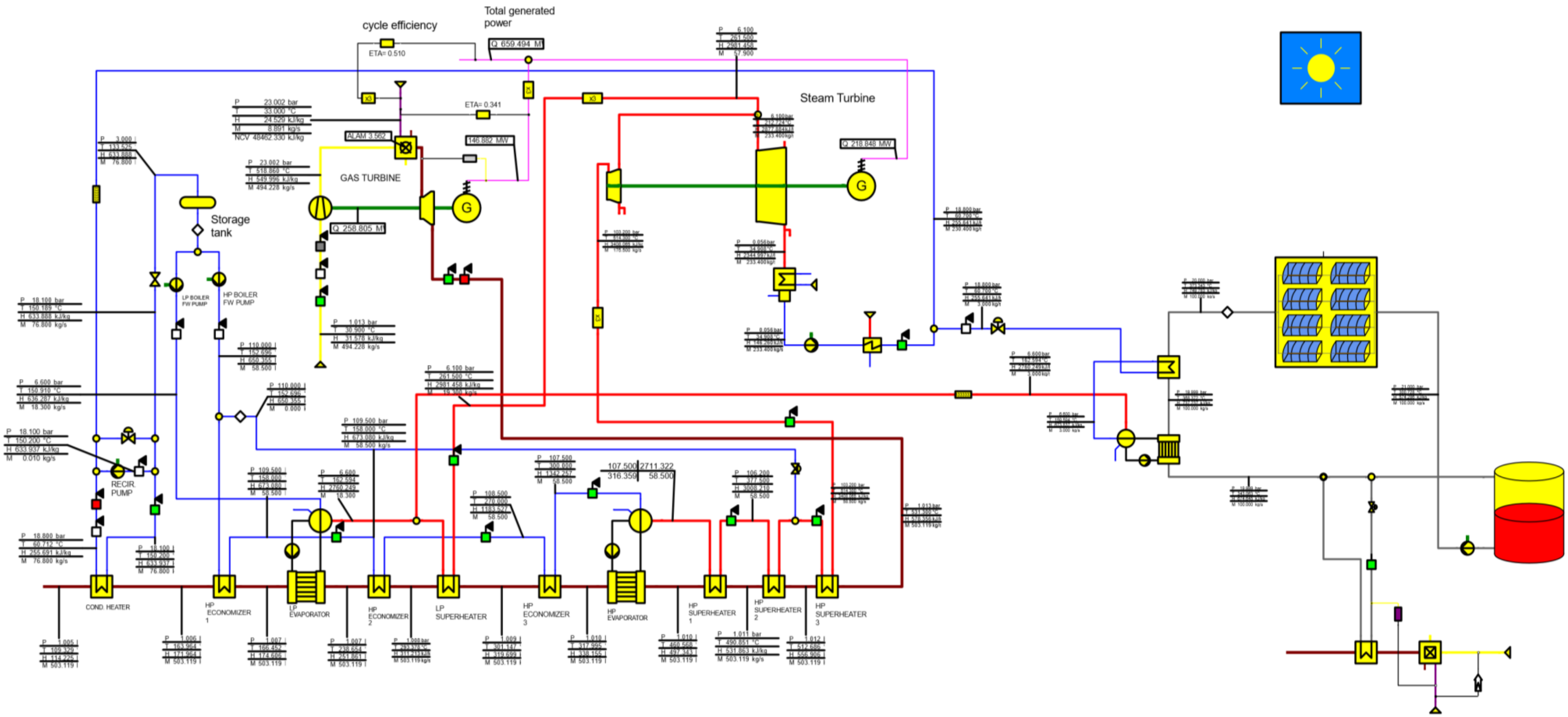


Figure 3.13. Solar boosting with saturated low pressure steam

CHAPTER FOUR

RESULTS AND DISCUSSION

The analysis of results for the design and off design conditions for GT13E2 combined cycle are presented in the following section

4.1 MODEL VALIDATION

The guaranteed performance and the design data for the GT13E2 were compared in the table below;

% base load	T1 (ºC)	T4 (ºC)	$\dot{m}$ gas (kg/s)	$T_{HPinlet}$ (ºC)	$T_{LP\ super\text{-}heater}$ (ºC)	$\dot{m}$ Hp inlet (kg/s)	$\dot{m}$ LP super-heater (kg/s)	Power GT (MW)
100	13	519.4	534.7	503.1	263.2	58.9	19.3	200
75		550	435.6	514.3	256	54.4	14.9	151.4843
50		498	399.7	490.2	250.5	42.1	13.8	100.9895
100	30.9	531.3	503.1	514.3	261.5	58.5	18.3	146.882
75		550	423.3	514.3	255.4	53.5	14.8	110.161
50		511.4	381.5	502.8	248.8	42.7	13.1	73.441

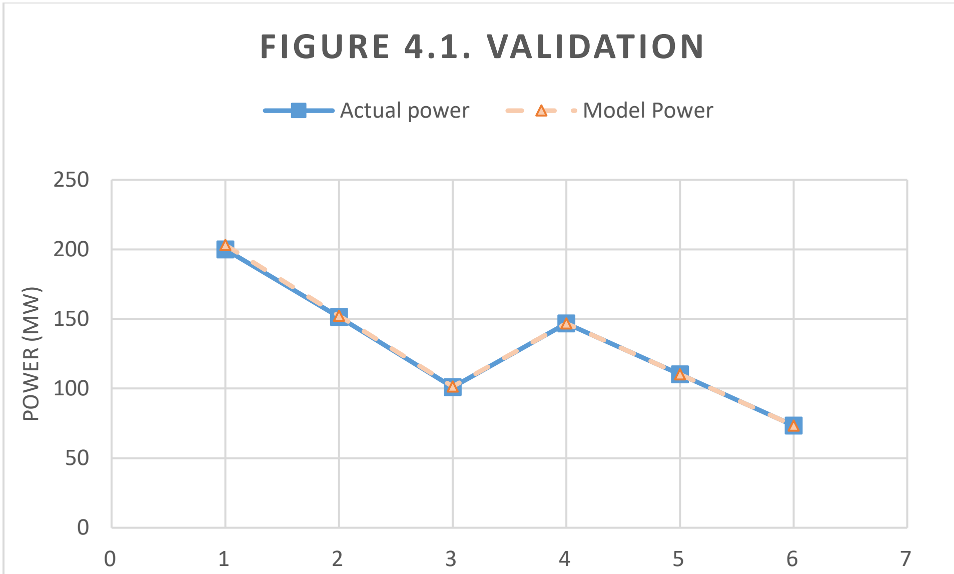
Table 4.1. Guaranteed Performance and Design Data Obtained From the Power Plant

4.1.1 COMPARISON OF MODEL DATA WITH ACTUAL DATA

The table below shows the model data;

% base load	T1 (ºC)	T4 (ºC)	$\dot{m}_{gas}$ (kg/s)	$T_{HPinlet}$ (ºC)	$T_{LP\ super\text{-}heater}$ (ºC)	$\dot{m}_{HPinlet}$ (kg/s)	$\dot{m}_{LPsuper\text{-}heater}$ (kg/s)	Power (MW)
100	13	519.4	535.6	504	260.8	59.2	19.7	203.146
75		550	479.4	530	264	58.7	16.6	152.359
50		498	434.2	490.5	254.2	45.5	15	101.573
100	30.9	531.3	503.1	514.3	261.5	58.5	18.3	146.882
75		550	433.2	532.5	260.9	53.8	15.1	110.161
50		511.4	388.5	503.19	253.5	43.3	13.2	73.441

Table 4.2. Model data



The table below shows a comparison of actual data (Table 9) and model data (Table 10)

% base load	T1 (%)	T4 (%)	$\dot{m}_{gas}$ (%)	$T_{HPinlet}$ (%)	$T_{LP\ super\text{-}heater}$ (%)	$\dot{m}_{HPinlet}$ (%)	$\dot{m}_{LPsuper\text{-}heater}$ (%)	Power (%)
100	-	-	0.1680	0.1785	-0.9202	0.5067	2.0304	1.5486
75		-	9.1364	2.9622	3.0303	7.3253	10.240	0.5744
50		-	7.9486	0.0611	1.4555	7.4725	8	0.5745
100	-	-	-0.0038	0	0	0	0	0
75		-	2.2853	3.4178	2.1080	0.5576	1.9867	0
50		-	1.8018	0.0783	1.8540	1.3856	0.7575	0

Table 4.3. Percentage error between the actual data and model

By observation, the highest error occurs at 75% baseload at 13ºC. This error can be justified by considering the inaccuracy of the air ratio used as the information was not available. Regardless, the model performs to a good extent as the actual plant. The results obtained from the calculations above agree with the works of Egware et al (2021), Andrea Miguez (2010) and Wallentinen (2016), all of whom used Ebsilon Professional to carry out their analysis. Since the results are consistent, further simulation ensues.

4.2 AMBIENT TEMPERATURE VARIATION

The ambient temperature was varied from 13ºC to 40ºC with the respective relative humidity obtained by calculations from Table 3 in the off design model for this simulation, the air ratio was kept constant and the simulations were carried out. The table below shows the results of the analysis.

T1 (ºC)	T4 (ºC)	P3 (bar)	$\dot{m}_{air}$ (kg/s)	$\dot{m}_{gas}$ (kg/s)	$T_{HPinlet}$ (ºC)	$T_{LPinlet}$ (ºC)	$\dot{m}_{HPinlet}$ (kg/s)	$\dot{m}_{LPinlet}$ (kg/s)	P(GT) (MW)	P(ST) (MW)	cycle efficiency (net)
13	485.95	24.26	525.14	534.71	477.39	198.42	52.53	73.05	182.18	64.46	53.20
20	504.74	23.76	512.60	521.90	493.10	204.05	55.11	74.70	167.99	67.81	52.30
22	509.87	23.62	509.13	518.36	497.29	205.60	55.79	75.13	164.04	68.71	52.00
24	514.89	23.48	505.70	514.85	501.34	207.12	56.44	75.53	160.12	69.59	51.80
26	519.80	23.34	502.32	511.40	519.80	208.61	57.07	75.92	156.25	70.44	51.50
28	524.58	23.20	498.99	507.99	509.05	210.06	57.67	76.29	152.40	71.26	51.30
30	529.25	23.06	495.70	504.62	512.70	211.48	58.25	76.65	148.59	72.05	51.00
32	532.48	22.92	492.45	501.29	515.22	212.46	58.58	76.80	145.39	72.52	50.80
34	534.48	22.76	489.24	498.01	516.78	213.08	58.70	76.78	142.67	72.70	50.70
36	543.15	22.70	486.03	494.81	522.13	216.65	53.71	71.61	141.88	80.00	50.04
38	546.76	22.58	482.91	491.63	524.92	217.85	53.98	71.73	139.86	71.73	49.99
40	550.54	22.47	479.83	488.49	527.81	219.12	54.26	71.87	137.93	71.90	49.95

Table 4.4. Variation in Ambient air Temperature results

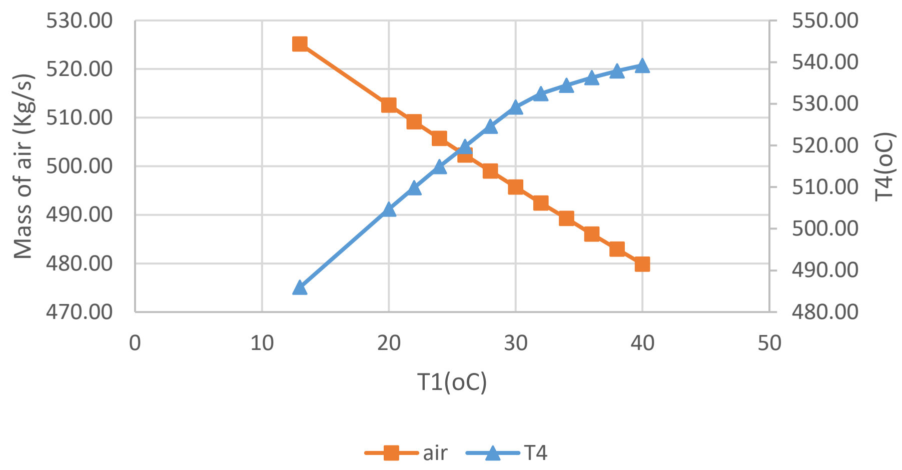


Figure 4.1. Inlet temperature vs mass of air and exhaust temperature

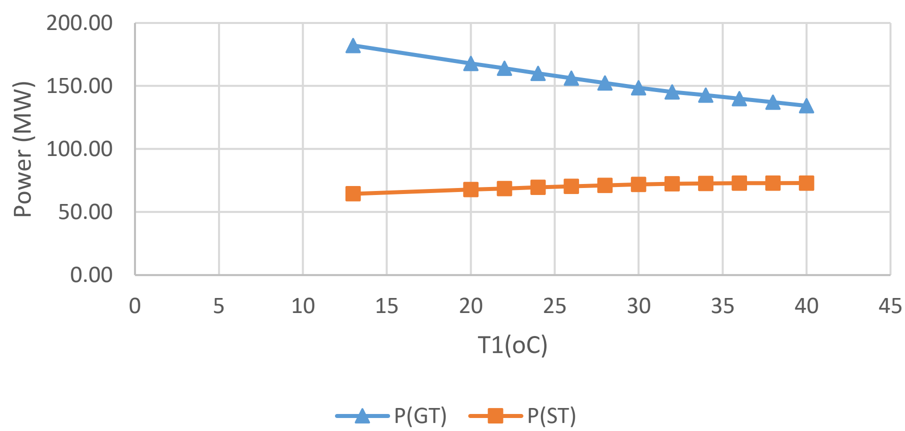


Figure4.2. Inlet Temperature vs GT Power and ST Power

From Figure 6 and Figure 7, the following can be observed;

- As ambient temperature increases, the mass flow rate of air reduces and as such this reduces the power that can be developed in the gas turbine
- As ambient temperature increases, the turbine exhaust temperature of the gas turbine increases and as such, the energy supplied to the steam turbine is increased therefore, more energy is supplied to the steam turbine.
- From Table 10, as the ambient temperature increases, the overall cycle efficiency decreases.

However, in actual power plant, the exhaust temperature of the gas turbine is kept constant by varying the air ratio. Considering that as ambient temperature increases, the mass flow rate of air decreases and as such the power generated in the steam turbine would therefore decrease with increase in ambient temperature.

4.3 FUEL AND AIRFLOW VARIATION

The HRSG is a temperature sensitive component, if there is a considerable change in the energy supplied to the HRSG, abnormal conditions would occur in the HRSG for example, if the temperature is not high enough, steam would not be liberated at the operating pressure in the evaporator component of the HRSG. As such, the exhaust temperature of the gas turbine is kept constant. This condition is made possible by burning extra fuel or reducing the fuel condition where necessary.

As explained earlier in section 3.4.2, to model this condition in EBSILON, a parameter called air ratio is used. The table below shows the results of the simulation.

T1 (°C)	T4 (°C)	$\dot{m}_{air}$ (kg/s)	$\dot{m}_{gas}$ (kg/s)	$\dot{m}_{fuel}$ (kg/s)	$T_{LPinlet}$ (°C)	$\dot{m}_{HPinlet}$ (kg/s)	$\dot{m}_{LPinlet}$ (kg/s)	P(GT) (MW)	P(ST) (MW)	Air ratio λ
13	530.1	525.14	535.88	10.74	209.15	183.77	236.75	209.94	74.85	3.17
20	530.1	512.60	522.55	9.95	209.64	179.85	231.63	182.96	73.33	3.33
22	530.1	509.13	518.87	9.74	209.77	178.81	230.26	175.76	72.92	3.38
24	530.1	505.70	515.23	9.53	209.89	177.79	228.93	168.78	72.52	3.42
26	530.1	502.32	511.65	9.33	210.00	176.80	227.64	162.01	72.13	3.47
28	530.1	498.99	508.12	9.13	210.11	175.85	226.39	155.44	71.76	3.51
30	530.1	495.70	504.64	8.94	210.21	174.93	225.19	149.05	71.39	3.55
32	530.1	492.45	501.24	8.79	210.28	174.09	224.10	144.15	71.06	3.59
34	530.1	489.24	497.91	8.66	210.32	173.35	223.12	140.41	70.76	3.60
36	530.1	486.03	494.62	8.59	210.36	172.64	222.21	136.77	70.48	3.62
38	530.1	482.91	491.37	8.46	210.38	171.99	221.36	133.22	70.21	3.64
40	530.1	479.83	488.17	8.34	210.39	171.39	220.58	129.77	69.97	3.65

Table 4.5. Variation of air fuel ratio with ambient temperature

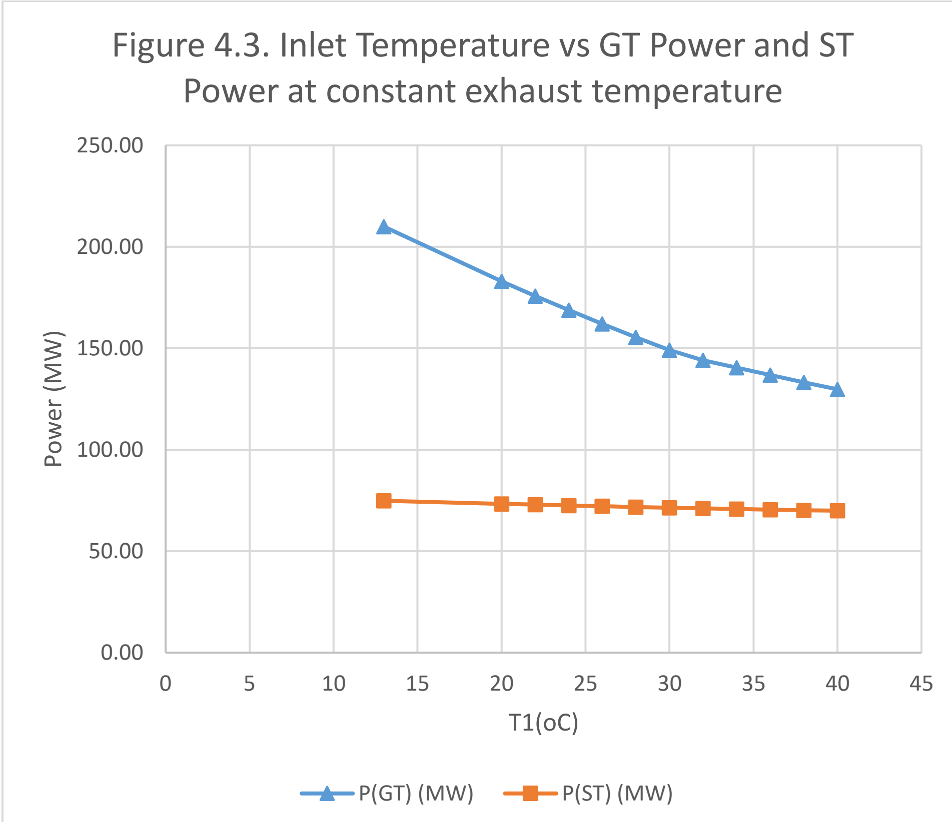


Figure 4.3. Inlet Temperature vs GT Power and ST Power at constant exhaust temperature
250.00
200.00
150.00
100.00
50.00
0.00
Power (MW)
0
5
10
15
20
25
30
35
40
45
T1(oC)
P(GT) (MW)
P(ST) (MW)

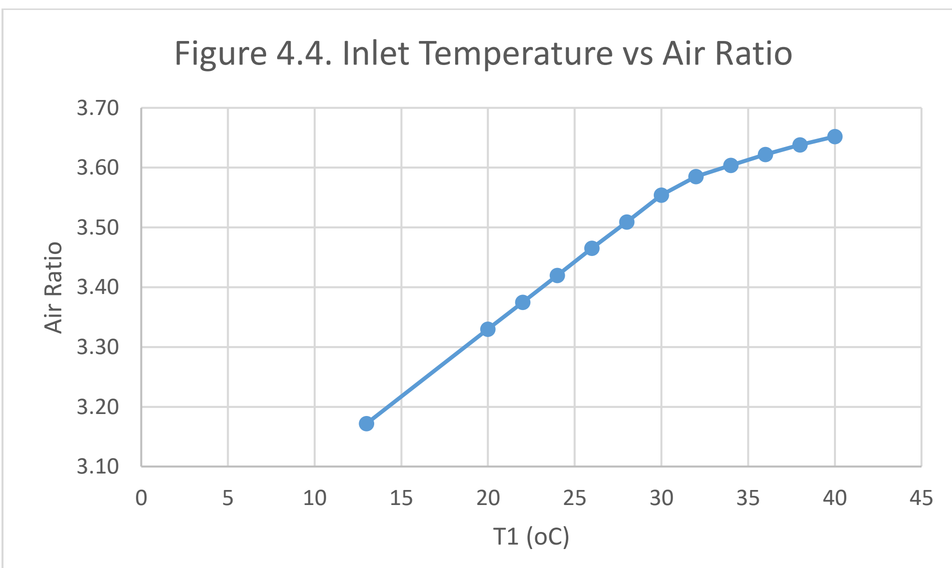


Figure 4.4. Inlet Temperature vs Air Ratio
3.70
3.60
3.50
3.40
3.30
3.20
3.10
Air Ratio
0
5
10
15
20
25
30
35
40
45
T1 (oC)

From Table 11 and Figure 8 and 9, it is observed that:

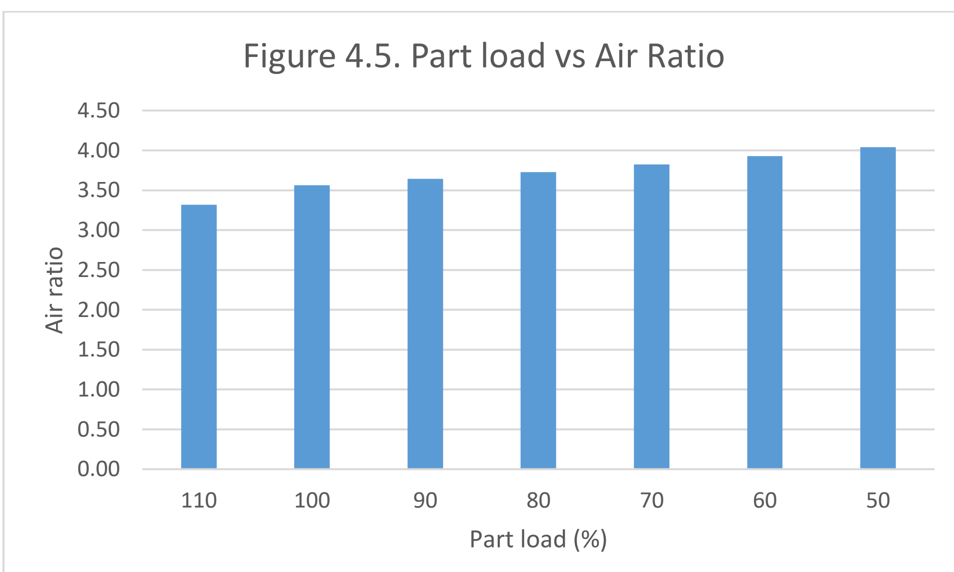
- As ambient temperature increases, the power developed in both the gas turbine and the steam turbine decreases when the exhaust temperature from the gas turbine is kept constant
- As ambient temperature increases, in order to maintain exhaust gas temperature at a constant value, the air ratio increases.

From Table 11, we can see that as ambient temperature increases, less mass goes through the turbine. This means that the air flow and the fuel flow consequently decreases. Although both the mass of air and the amount of fuel is decreasing, the fuel decreases way less compared to the air flow, hence at higher ambient temperatures, more fuel is required for combustion.

The table below shows the simulation for part load conditions from 110% to 50%.

Part load (%)	T4 (ºC)	$\dot{m}_{air}$ (kg/s)	$\dot{m}_{gas}$ (kg/s)	m fuel (Kg/s)	$T_{LPinlet}$ (ºC)	$\dot{m}_{HPinlet}$ (kg/s)	$\dot{m}_{LPinlet}$ (kg/s)	P(GT) (MW)	P(ST) (MW)	Air ratio λ
110	558.4	494.23	503.78	9.55	218.73	190.63	240.23	161.57	78.59	3.32
100	531.3	494.23	503.12	8.89	210.79	175.23	225.88	146.88	71.63	3.56
90	531.3	473.31	481.64	8.33	211.63	168.32	216.87	132.19	68.89	3.64
80	531.3	450.73	458.47	7.75	212.49	160.85	207.14	117.51	65.89	3.73
70	531.3	426.11	433.25	7.14	213.35	152.69	196.52	102.82	62.59	3.82
60	531.3	398.96	405.47	6.51	214.22	143.67	184.77	88.13	58.91	3.93
50	531.3	368.53	374.36	5.84	212.39	133.25	165.13	73.44	53.61	4.04

Table 4.6. Variation of air fuel ration with part load



Once again we see the same trend that occurred during the variation of ambient temperature. The figure below shows the TS diagram when less mass goes through the turbine.

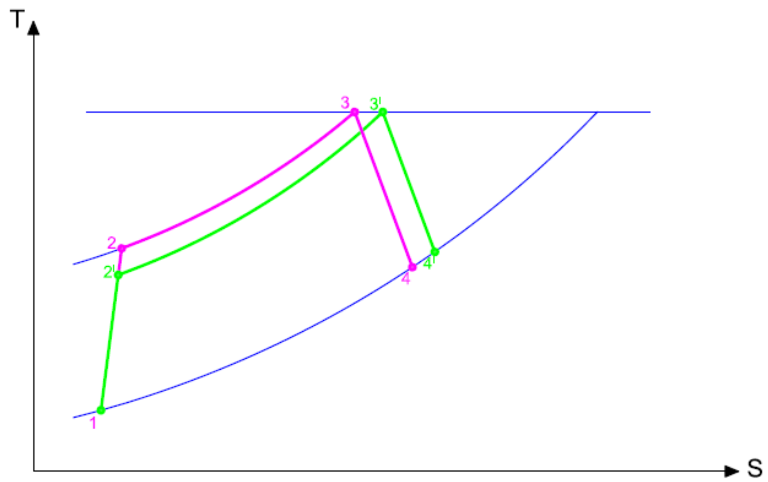


Figure 4.6. TS diagram for gas turbine accepting less air and receiving air at design condition

From Figure 21, 1-2-3-4 represents the gas turbine cycle working in design conditions while 1-2'-3'-4' represents the gas turbine operating with less air. It can be observed that when the airflow is reduced, the compression process is shorter and the subsequent combustion process is longer in order to bring the exhaust temperature to design condition. In real gas turbines, the mass of air entering the compressor is controlled by changing the orientation of the blades at the entrance (inlet guide vanes).

4.4 SOLAR BOOSTING

From the last section, we can summarize that when the gas turbine operates at higher ambient temperatures, there will be losses in power hence to reduce this losses we introduce a solar retrofit as described in Section 3.7.

4.4.1 HIGH PRESSURE SOLAR BOOSTING

The results from the simulation of high pressure solar is shown in the table below:

Solar Steam added (Kg/s)	Ambient Temperature (°C)	POWER IN STEAM TURBINE (MW)	Power added by Solar (MW)

0	13	224.55	
	30.9	217.20	
	40	210.00	
1	30	218.27	1.07
2		219.33	2.13
3		220.40	3.20
1	40	211.28	1.28
2		212.47	2.47
3		213.11	3.11

Table 4.7. Comparing performance with high pressure solar heat added.

From Table 13, it is observed that as the mass flow of water increases, the power developed in the steam turbine increases. However as mentioned earlier in Section 4.3, The HRSG is a heat sensitive component hence there is a limit to how much mass flow of solar steam can be added to the cycle. The results from the simulation shows that the maximum safe amount of solar steam that can be added is 3 Kg/s.

4.4.2 LOW PRESSURE SOLAR BOOSTING

The results from the simulation of low pressure solar is shown in the table below:

Solar Steam added (Kg/s)	Ambient Temperature (°C)	POWER IN STEAM TURBINE (MW)	Power added by Solar (MW)
0	13	224.55	
	30.9	217.20	
	40	210.00	
1	30	217.75	0.55
2		218.30	1.10
3		218.85	1.65
1	40	210.50	0.50

2		211.05	1.05
3		211.70	1.70

Table 4.8. Comparing performance with low pressure solar heat added.

From Table 13, the results of simulation is similar to the high pressure solar heat added except that the extra power developed by the low pressure solar heating is considerably lower than the power generated by high pressure solar boosting. Hence we can safely conclude that the best configuration for solar boosting is high pressure solar boosting.

CHAPTER FIVE

CONCLUSION AND RECOMMEDATION

In this research project, the importance of modeling using computer engineering soft wares have been illustrated. The modeling and simulation of GT13E2 combined cycle gas turbine has been done with the aid of the software EBSILON PROFESSIONAL. The design mode was modeled using guaranteed performance data from the power plant, in the off design, temperature variation of ambient air and fogging (spray water at inlet to compressor) was simulated. The fogging was simulated under two different modes; constant fuel consumption and constant turbine exhaust temperature .The model results were validated using actual operating data by applying error percentage analysis. The validation results obtained ranged from -0.0038% to 0% in design condition while the results varied from -0.9202% to 10.24%

From the research, we can conclude that as ambient temperature increases, the mass flow rate of air reduces and as such this reduces the power that can be developed in the gas turbine. Also, since the energy available in the flue gas from the gas turbine is reduced at higher ambient temperature, the power developed in the steam turbine reduces also. At higher ambient temperature, the overall cycle efficiency decreases.

In order to maintain the design exhaust temperature, extra fuel has to be burned to extend the combustion process.

The results achieved from the simulation of solar boosting revealed that as mass flow of solar steam increases, power developed in the steam turbine increases. However since the HRSG is a heat sensitive component, the limit to the amount of solar steam that can be added is 3Kg/s. If extra mass of water is added, issues will arise in the most critical part of the HRSG which is the evaporator. If the energy available at the location of the evaporator is not enough, steam would not be generated hence the steam cycle would fail.

From the analysis and simulation of the High pressure solar boosting and the Low pressure solar boosting, we can conclude that the highest extra power generated in the high pressure solar boosting is 3.2 MW while that of low pressure solar boosting is 1.7 MW, hence high pressure solar boosting is the best configuration.

RECOMMENDATIONS:

After carrying this research works, the following are recommended:

1. Nigerians should learn the technology of building turbines so as to ensure that the turbines are built using weather conditions present here in the country.

2. Supplementary firing can be considered to increase the mass flow of solar steam that is super heated in the HRSG.
3. A better and more efficient combustion chamber should be introduced to reduce the fuel consumption.
4. Regular maintenance of the plant as this guarantees a better life of the gas turbine thus reducing the failure rate and as such increasing the units' reliability.

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