Technical and Economic Analysis of Parabolic Trough Concentrating Solar Thermal Power Plant

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Key Words

Renewable Energy, Concentrating Solar thermal Power (CSTP), Heat Transfer Fluid (HTF), Wet Cooling, Capacity Factor, Water Consumption, Dry Cooling, Levelized Cost of Electricity (LCOE), Thermal Energy Storage (TES), Parabolic Trough, Solar Multiple, System Advisor Model (SAM), Solar Field, Thermal Block, Power Block, Steam Rankine Cycle, Power system stability

Declaration

I, Samuel Kariuki Kibaara, submit this thesis in the fulfilment of the requirements for the degree of Master Science in Electrical Engineering.

I certify that all ideas, designs and conclusions are my own work. All the materials reproduced have been appropriately referenced.

This thesis has not been submitted to this, or any other University, for degree purposes.

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Abstract

Population growth rate and economic development are leading to a continuous increase in energy demand all over the world. At the same time conventional energy sources are being depleted. The emissions of the green house gases are also a major concern in the energy industry. These factors have ultimately lead to an increasing implementation of renewable energy sources, particularly Concentrating Solar Thermal Power (CSTP).The common CSTP technologies in the market today are parabolic troughs, power tower, Linear Fresnel and the parabolic dish. CSTP plants have the ability to provide electricity, refrigeration and water purification in one unit. These technologies will be extremely helpful in improving the quality of life for many people around the world who lack the energy needed to live a healthy life.

Presently parabolic trough CSTP is the most efficient and economic technology to generate electricity from solar energy. It has been deployed in large scale installations for more than 20 years and has proved to be the best technology in terms of energy conversion efficiency compared to the other CSTP plants. Parabolic trough power plant collects heat from the sun using collectors. The collected heat is reflected and concentrated onto the absorber tubes. The absorber tubes contain a heat transfer fluid (HTF) flowing inside them which absorb the reflected heat from the collectors. The heated HTF flows into the steam Rankine cycle where heat exchange between HTF and water occurs. The boiled water is used for steam production which runs a turbine for electricity generation. Once the steam has passed the turbine it is cooled back into water using condensers before being reused.

Cooling of CSTP plants require a large volume of water. The availability of water required for cooling is becoming more limited especially in the arid areas where CSTP plants are located. This has prompted the use of dry (air) cooling to condense the steam turbine exhaust vapour. However the capital costs of a dry cooled CSTP plant are approximately 5% higher than wet cooled CSTP plant. The cost of electricity from a dry cooled parabolic trough plant is over 10% higher that of a wet cooled plant. The losses of energy that occur in dry cooling make it less efficient compared to wet cooling. The net energy production of a dry cooled plant is low and hence achieves lower capacity values.

This thesis reports on the technical and economic analysis of wet and dry cooling technologies of parabolic trough CSTP plant. This was done through modelling and simulation of a standalone and grid connected parabolic trough using the System Advisor Model (SAM). The location of the

proposed study is Lodwar, Kenya which is a semi arid area. The data for simulation is past data available in literature. The thesis took into account the available solar potential in Lodwar, Kenya. The main objectives in this thesis, therefore, are to study the economic, technical and environmental analysis of dry and wet cooling systems applied in the solar parabolic CSTP plant.

The economic analysis will present the different factors that affect the levelized cost of electricity (LCOE) of parabolic CSTP plants. This will include the impacts of variation of initial temperature difference (ITD), impacts of approach temperature variation, effects of thermal energy storage size, effects of variation of the value added tax (VAT) and effects of variation of inflation rate.

The technical analysis done compares the two cooling technologies in the following dimensions i.e. energy production, efficiency of thermal energy collection, solar to thermal conversion efficiency, capacity factor and the thermal energy storage.

The environmental analysis compares the two cooling in terms of land usage and water consumption. The amount of carbon dioxide avoided by installation of a 50MWe parabolic CSTP and the impacts of parabolic trough on flora and fauna will also be discussed.

Stability analysis of the micro grid is done. It will used to investigate how long the system takes to respond to disturbances in the system. In this case the synchronous generator is modelled as a standalone and grid connected. Stability analysis is done using DIg SILENT power factory 14.0. The voltage, frequency, speed, real and reactive power will be monitored.

List of Acronyms

ACC:	Air Cooled Condenser
BOP:	Balance of plant
CSTP:	Concentrating Solar Thermal Power
DNI:	Direct Normal Irradiance
GWh:	Gigawatt hour
HTF:	Heat Transfer Fluid
HCE:	Heat Collecting Element
HRSG:	Heat Recovery Steam Generator
ISCC:	Integrated Solar Combined Cycle
ITD:	Initial Temperature Difference
kWh:	Kilo watt hour
LCOE:	Levelized Cost of Electricity
MW:	Megawatt
MWe:	Megawatt electric
NREL:	National Renewable Energy Laboratory
NPV:	Net Present Value
PCU:	Power Conversion Unit
PV:	Photo Voltaic
REFIT:	Renewable Energy Feed in Tariff
R&D:	Research and Development
SAM:	System Advisor Model
SF:	Solar Field
SRC:	Steam Rankine cycle
SEGS:	Solar Energy Generating Station
SCA:	Solar Collector Assembly
SM:	Solar Multiple
TES:	Thermal Energy Storage
W/m^2:	watt per square metre

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1 Introduction

1.1 Background to the Study

Energy is a basic requirement in both developed and developing countries for improving the living standard of the people. Africa is endowed with a wide range of renewable and non-renewable energy sources. The non-renewable energy resources found in Africa include coal, oil and natural gas. The renewable energy sources of energy found in Africa include wind, solar, hydroelectricity, geothermal, biomass, tidal and wave power. However due to low capital investment, majority of these energy resources have not been exploited. Africa is reported to have the lowest level of electrification in the world. Of the total population in Africa only 26% have access to electricity. This means that a total of 547 million people are not able to access electricity [11].

The energy demand growth in Kenya is continuously increasing because of the high population growth at a rate of 2.462% per year (2011). As a result, the energy utilities are now rationing electricity and have increased the cost of electricity per kWh to 0.285$cents/kWh (2012) to maintain its energy reserve. This has led to under-production of goods and services by some commercial and industrial companies. This has therefore resulted in laying off workers thereby increasing the rate of unemployment [2, 4].

Kenya has plenty of renewable energy sources the most important amongst them being solar energy, wind energy, energy from biomass, hydroelectricity and geothermal energy. Currently, hydroelectricity is the most exploited renewable energy source in Kenya [2]. Kenya has a net renewable energy potential of over 14,000MW as shown in Table 1-1. From this potential, only 1536MW has been exploited. The exploitation of the installed energy capacity has been contributed by the energy utilities in Kenya in the following ratios as shown in Table 1-2.

Nonetheless, this installed energy capacity is not enough to meet its population of about 40 million people whose energy demand growth rate is estimated at 12% per annum [4].

Table 1-1 Renewable Energy Exploitation in Kenya [4]

Energy Resource	Potential	Exploited capacity
Geothermal	10000MW	198MW
Hydro	3000MW	764.5MW
Cogeneration	118MW	26MW
Wind	3-10m/s^2	5.1MW
CSTP	4-6KWh/m^2	-

Table 1-2 Energy Utilities and their % Energy Production in Kenya [4]

Utility	Thermal (MWe) (Fossil fuels)	Hydro (MWe)	Geo-thermal (MWe)	Wind (MWe)	Biogas & solar (MWe)	Total (MWe)	% exploitation
Kengen	390	764	185	5.1	0	1344	87.5
KPLC	5.1	0	0	0	0	5.1	0.3
IPPS	148	0	13	0	26	187	12.17
Totals	525	764	198	5.1	26	**1536**	0.1
%	34.17	49.73	12.9	0.3	1.7	-	100

The government of Kenya has proposed to increase its installed capacity in the next five years by 1250MWe. These plans include importing 250MW from Ethiopia and building a 1000MWe nuclear plant by 2017. However implementing this proposal is costly in terms of capital equipment investment and on environmental safety, particularly the management of the effluent from the nuclear plant [4].

In order to stop depending on other countries for energy, the energy stake holders in Kenya should explore harnessing its indigenous renewable energy sources. CSTP is one of the possible solutions to supplement this energy deficit. Furthermore, as shown in Table 1-2 above the main source of electricity comes from hydroelectric plants. The rivers and dam levels are prone to the dry season. In this dry season, rationing of electricity becomes inevitable in Kenya since the volume of rivers and dam levels decrease causing insufficient water availability for generating enough energy to meet the demand.

It is proposed in this thesis that during dry seasons, the vast dry waste land in Eastern Kenya is adequate space for placement of the bulky CSTP mirrors. The average Direct Normal Irradiance (DNI) for Lodwar, Kenya is 1836kWh/m^2/year. This weather condition is potentially capable of generating electricity via CSTP plants. Spain for example is a giant in CSTP and has a DNI of 2136 kWh/ m^2/year. Examples of CSTP plants in Spain are Andasol 1 and Andasol 2. CSTP plants in USA include Bakersfield and Kimberlina found in Mojave Desert, California. Mojave Desert receives annual DNI of 1965.2kWh/m^2/year [3].

CSTP generation utilizes solar irradiance as a source of heat energy. The heat energy is concentrated to produce steam which is used to drive a steam turbine drives. In this regard, CSTP generation is similar to other thermal power plants such as coal-fired, gas-fired or nuclear. Some CSTP plants also employ heat engines to convert heat to electrical energy.

The first energy generation using CSTP principle took place in California, USA in 1980 as a result of the oil crisis of the 1970s. However, the deployment of CSTP was not very successful with the discovery of cheap natural gas in developed countries. Currently again due to the rising awareness on global warming and due to the volatility of gas prices, interest in CSTP deployment is growing rapidly [3].

Africa has high solar energy resource potential equivalent to 100 million barrels of oil per annum. However, lack of large-scale investment on CSTP has delayed its deployment in Africa. With sufficient investment, CSTP plants can provide an economic source of clean electricity without creating any environmental pollution hazard or global warming. It is reported that investment in solar energy using CSTP in Africa can meet the total energy demand and still export the surplus to Europe. Several CSTP projects have been announced while others are under construction and some entering into service. An example is Desertec, a German project in North Africa, which has announced 100GW of electricity production using CSTP plants. South Africa and Botswana are conducting a feasibility plan to erect a 100MW and 50MW capacity CSTP plants respectively. This therefore shows a promising future for CSTP technologies as key generating sources along with wind energy, hydroelectricity, solar Photo Voltaic (PV) and biomass energy [5, 11].

Water is a scarce resource in many African countries with arid and semi-arid regions. Economic development in these African countries and in many other regions in the world has been hampered by limited resource of and limited access to water and energy. It has been forecasted that by 2025 over 60% of the world population will live in regions with imbalances of water accessibility and energy. This will especially be more prevalent in Asia, Africa and Latin America. In 2000 the use of water for cooling thermoelectric plants such as coal, nuclear, oil, natural gas and CSTP plants in USA was reported to draw 39% (136 billion gallons per day) of the total fresh water consumption, which ranks second after irrigation. Generation of each kWh thus requires about 25 gallons of water (which is the weighted average for all thermal plants) mainly for cooling purposes. Thermal power plants also use water for washing of mirrors in CSTP plants, ash handling in coal plants and flue gas desulfurization. Desulfurization is a process of removing sulphur dioxide from boiler and furnace exhaust gases. In this process water absorbs sulphur dioxide. In USA water usage by thermal power plants is projected to increase by 18% by 2030 with rise in energy demand [115].

The amount of water used for cooling a thermal power plant depends on size of the plant, cooling type used and the ambient temperatures encountered in the region. There are two types of cooling water system design: once through (open loop) and recirculation (closed loop). In once-through cooling, water is withdrawn from nearby water bodies e.g. a lake, river or ocean. The water is re-circulated in the cooling tower of the thermal power plants after which it is discharged back to the water body. Thermal power plants equipped with this type of cooling mechanism withdraw large volumes of water but the water consumed in cooling is little. Closed loop or recirculation systems are divided into three types: These are wet cooling towers (wet recirculation); cooling ponds (wet recirculation) and dry cooling (air recirculation). Wet cooling towers and cooling ponds use water to dissipate heat on the condensers of thermal power plants. Air cooled systems operate by circulating the ambient air on the condensers to dissipate the hot exhaust steam from the turbine. The exhaust steam from the turbine circulates inside the dry condenser tubes. The tubes carrying the exhaust steam are blown with ambient air drawn by the fans. The exhaust steam is therefore cooled by conduction heat transfer [65,115].

Cooling of CSTP plants is seen as a major problem contributing to its slow deployment as compared to other renewable energy sources such as wind, PV and biomass. CSTP plants are cooled either by water, air or a mixture of water and air. Almost 90% of the total water consumed in CSTP plants is used for cooling the exhaust heat from the turbine. For example, in wet cooled parabolic trough CSTP a plant, cooling the exhaust steam is provided by mechanical draft wet cooling towers. The cooling towers consume 800 gallons of water for each MWh generated. Dry cooling/air cooling on the other hand uses air for cooling the exhaust steam. It saves up to 90% of the total water used in CSTP plants. A dry cooled parabolic trough uses 80 gallons of water for each MWh generated. The water is mainly used for mirror washing. The fans used for dry cooling are large and expensive. This increases the capital cost of dry cooled plants by about 5% and the cost of electricity by 10% [65].

In this simulation parabolic trough plant has been considered. This is because based on the literature review conducted it is the most common CSTP technology and most of its parameters have been standardised. The purpose of the research presented in this thesis therefore, is to investigate the technical and economic feasibility of dry and wet cooled parabolic trough CSTP plant in Lodwar, Kenya. The study takes into account the available solar potential in Lodwar and other data concerning the design of parabolic trough. The two cooling types of parabolic troughs plants are analysed in terms of:

- Performance: Includes energy production, capacity factor, solar electric efficiency and solar thermal heat collection efficiency
- Effects of solar multiple (SM) on energy production
- Effects of initial temperature difference (ITD) variation on LCOE for dry cooled plant
- Effects of approach temperature variation on LCOE for a wet cooled plant
- Thermal energy storage (TES) and its effects on energy production, LCOE and capacity factor
- Water consumption and land usage
- Sensitivity analysis of parabolic trough on NPV and LCOE due o inflation and tax.
- After tax cash flow

1.2 Objectives of Research

1.2.1 Problems to be investigated

The objectives of this study are to:

- Define the Concentrating Solar Thermal Power technologies (CSTP) by doing the following:
    - Conducting a thorough literature review of the existing CSTP technologies in the current market.
    - Identify the cooling types applicable for CSTPs technologies.
- Study the economic analysis of dry and wet cooling:
    - The economic analysis presented includes sensitivity analysis of the parabolic trough CSTP on LCOE due to variation in inflation rate, impacts of value added tax(VAT) on LCOE, effects of thermal energy storage size on LCOE, effects of variation of the on initial temperature difference on LCOE, effects of variation of approach temperature on LCOE and after tax cash flow.
- Study the technical analysis of wet and dry cooling by comparing the following ways:
    - The efficiency of thermal energy collection, energy production, capacity factor, solar multiple and thermal energy storage
- Study the impacts on the system stability when the CSTP plant is operating in grid-connected and standalone modes:
- Study the environmental impacts of both dry and wet cooling in terms of:

- Land usage, water consumption, impacts on flora and fauna
- Amount of carbon dioxide avoided by installation of a 50MWe parabolic trough

➢ Identify areas of research directed towards improving the performance and reduction in the operation and maintenance costs of parabolic trough CSTP plants:

1.3 Research Methodology

Research methodology followed in this thesis is discussed in the following section with the sole aim of ensuring that the methods followed are clear.

i. Literature review

The scope of the present research work is on the technical and economic analysis of cooling methods applied in parabolic troughs CSTP Plants in Lodwar, Kenya. This has been done through conducting a literature review of the current CSTP technologies. The intention of the literature review is to gather all the work previously done on CSTP. Academic papers, reports and textbooks will be used to complement the theories put forward where possible. The main aim of the literature review conducted on this thesis is to identify the main CSTP technologies and their level of penetration in the market currently. The working principle, advantages and disadvantages of each technology is done. The factors considered during CSTP deployment are discussed alongside the factors hindering its wide deployment in many parts of the world.

ii. Simulation work

Simulation of the parabolic trough CSTP plant forms the main body of this research work. The location of the parabolic trough CSTP plant is Lodwar Kenya. The main blocks making up the parabolic trough are discussed and their operating parameters defined. These blocks include the solar field, power cycle, thermal energy storage and the power block. The cooling methods of the parabolic trough i.e. dry and wet cooling are discussed and their parameters defined. These cooling methods form the main case studies which are discussed in the following dimensions viz; Economic analysis, technical analysis and environmental impacts.

The economic analysis will include the impacts of tax variation on LCOE, impacts of increasing the hours of thermal energy storage (TES) on LCOE, impacts of inflation rate on LCOE, impacts of varying initial temperature difference (ITD) of a dry cooled plant on LCOE, impacts of varying approach temperature of a wet cooled plant on LCOE and the cash flow.

The technical analysis will comprise of energy production, efficiency of energy production, capacity factor and solar multiple analysis. The environmental analysis will deal with water usage, land usage, impacts of the two cooling types on flora and fauna and the amount of carbon dioxide avoided by installation of a parabolic trough. The simulation package used is the System Advisor Model (SAM) which is an open source software made by the National Renewable Energy Laboratories (NREL), USA.

Power system stability will be done to evaluate the impacts on the stability of voltage, active and reactive power and speed when the CSTP plant is operating in grid-connected and standalone modes. Power system stability is done using DIg SILENT power factory 14.0.

1.4 Limitations of the study

This research had certain limitations which include:

- The weather data for DNI used here is based on weather data recorded in 2009. Having more recent recorded data would have produced more accurate results. This is because the weather patterns have changed since the recorded date.
- The model assumed water in the vicinity but Lodwar is a semi-arid region with little or no water. Therefore, for the wet cooling analysis to be accurate, water must be imported to the plant site.

The first chapter is the **Introduction**

It provides the general background information of CSTP plants regarding the purpose, motivation, scope of work, limitations and the project's plan of development. This chapter discusses the energy poverty in Kenya and the need to deploy more renewable energy technologies to add to the already existing ones. The potential and level of exploitation of the different renewable energy sources is also discussed. Parabolic trough CSTP is suggested for deployment owing to the big stretches of land in Lodwar and the availability of solar radiation in this region. The chapter also discusses on the different cooling technologies applicable in parabolic trough plant.

Second chapter is the **Literature Review**

It provides the historic background, theoretical and technical evaluations of CSTP technologies which serve as a guideline for the Model Development and Simulation. The working principle of the four main CSTP technologies is discussed and their level of exploitation in the world. The different types of thermal energy storage and cooling types applicable in CSTP plants are also analysed. The environmental factors, economic impacts and challenges influencing CSTP deployment in the world are also discussed.

Third chapter is **Model Development and Simulation**

In this chapter, all the relevant theories are explained and simulations are presented to verify the theory. This chapter introduces the software used in the modelling of the parabolic trough CSTP plant. The design of the parabolic trough blocks is done in this chapter and the factors influencing its location in a given area. The two cooling types i.e. dry and wet cooling are discussed and designed. Power system stability is defined and discussed.

Fourth chapter are the **results and analysis**

This chapter presents the results of economic, environmental and technical analysis. The economic analysis presented includes the effects of initial temperature difference on LCOE, effects of approach temperature on LCOE, sensitivity analysis of parabolic trough on LCOE and NPV due to variation of inflation rate, sensitivity analysis of parabolic trough on LCOE and NPV due to variation of value added tax (VAT) and the effects of thermal energy storage size on LCOE.

The technical analysis presented includes the energy production, efficiency of thermal energy collection and energy capacity factor on each cooling method. Also presented is the power system stability analysis when the CSTP is operated as a standalone unit and when connected to the grid.
The environmental analysis includes the impacts of CSTP on water, land usage, flora and fauna.

Fifth Chapter is **Conclusions and Recommendations**

In this chapter a summary of the analysed results and recommendations for future work regarding parabolic trough CSTP is presented.

2 Literature Review

2.1 Motivation of the Study

The environmental friendliness of renewable energy sources and their lifetime sustainability are facilitating their rapid deployment all over the world. In order to overcome economic and regulatory barriers associated with the deployment of renewable energy sources, the power and energy sector is seeking novel renewable energy technologies with environment friendliness, that is easily controlled and has high energy potential at the most acceptable cost possible.

Deployment of CSTP plants for power generation is growing rapidly especially in areas where solar energy resource is plentiful. It might be mentioned in this context that at the beginning of the 21^{st} century, the world started encountering three major problems: (a) global warming and climate change caused by release of greenhouse gases as a result of industrialization and economic development; (b) Energy shortage and rise in gas and oil prices due to the depletion of fossil fuels, population and economic growth and (c) shortage of fresh water in most parts of the world caused by deforestation to obtain energy by burning wood. Some solutions being applied to counter the aforesaid problems are: Introduction of energy efficient systems and energy saving; introduction of renewable energy sources; coal gasification and carbon dioxide sequestration and the introduction of nuclear power. The global energy crisis of 1970 brought to light the fact that many countries need to depend on other countries for fossil fuels to meet their energy demands. Harnessing indigenous renewable energy can provide an achievable solution to all these country-specific problems [1, 5].

Solar Energy is a form of renewable energy that does not diminish with time. Renewable energy resources produce a clean and cheap form of indigenous energy [7]. It is a fact that almost 80% of the energy used today comes from natural gas, coal and oil which are gradually running out with population and demand growth. Renewable energy is moreover, able to alleviate environmental pollution that leads to climate change and global warming by minimising emissions [8]. Renewable energy sources include, geothermal energy, hydro-electric power, solar energy, wind energy, biomass, tidal and wave power [9, 10].

Energy poverty has diminished Africa's productive capacity. Africa has huge energy demands especially in the sub Saharan Africa which is reported to have the lowest rate of electrification in the world of about 30%. These regions mostly depend on traditional biomass used mainly for cooking and water heating for households. Traditional biomass accounts for 80% of their total energy needs. Africa is well endowed with renewable energy resources that exceed its demand requirements for the next century. However many African countries are still struggling for energy access which is a major reflection of the low income and economic under development [11].

Kenya is currently suffering from energy crisis. Almost 75% of the Kenyans living in the rural areas still have no access to electricity today. The utility companies in Kenya (Kenya Power and Lighting Company (KPLC) and KenGen) are currently exploring augmentation of generation capacity with nuclear power but this is likely to introduce more health risks from radiation. The radiation from nuclear power consists of subatomic particles which travel at a velocity of 186,000 miles per second, which can easily penetrate into the human body and cause some biological disorder [12].

The main power generation plants, the geothermal power generation station in Naivasha and the Seven Forks Hydro projects along Tana River in Kenya, have not been sufficient to meet the national electricity demand since 2009. Increasing the capacity of hydro-electric stations is almost impossible because rivers are drying up as a result of population increase and other factors. In the search for energy from wood the population is cutting down trees which extract water from the ground and release it to the atmosphere.

When parts of these forests are cleared the region cannot hold as much water and this result in a much drier land which results to rivers drying up. Majority of the rural areas in Kenya has the potential of generating energy from the sun, wind and biogas.

The main motivation behind this research is to investigate how the solar energy resource in Eastern Kenya in particular Lodwar, can be tapped using parabolic trough CSTP plants and the most suitable cooling technology.

It is believed that by harnessing indigenous solar energy, Kenya will be able to meet its prime objectives of the development plan dubbed '*Vision 2030*'. The *Vision 2030* has identified energy as one of the key pillars in the continual economic growth of Kenya. The vision is committed to supplying clean, adequate, reliable and affordable energy to Kenyan citizens.

The vision is expected to help reduce poverty by half, by the year 2015 as per the Millennium Development Goals (MDGs).

The Ministry of Energy has pledged to fulfil the *Vision 2030* by doing the following:

- Enhancing power generation capacity
- Increasing access to electricity
- Developing new renewable sources of energy
- Ensuring security of fossil fuels resources
- Capacity building of the energy sector
- Improving quality and reliability of electricity supply throughout the country.
- Providing electricity to areas that are currently not supplied from the national grid
- Providing the power link to the neighbouring countries to enhance energy exchange in the region
- Reducing transmission losses that currently are costing the country up to US$17 million per year
- Reducing the cost of electricity to the consumer by absorbing the capital cost of transmission lines since they will be fully funded by the government [14]

This project would serve as a stepping stone towards the implementation of a green city fully powered by renewable energy sources. This implementation acts towards the fulfilment of the Kenya's *Vision 2030*.

2.2 History of Solar Thermal Power Plants

Solar energy concentration for heat production dates back to early human civilization. Mirrors were used to concentrate heat for lighting fire in Greece and China. They used this technique of concentration of solar energy for putting the enemy ships on fire during warfare. In the early 20th century, scientists had built machines that could concentrate the solar energy for pumping water for irrigation purposes.

The first thermal plant was built in Meadi, Egypt in 1913. The mirrors had the shape of a parabola and hence the technology was named the parabolic trough. The heat of the sun was focused on the parabolic mirror which was 62m long. The concentrated heat was focused on a tube full of water running in it. The steam thus produced powered an engine which was coupled to a pump and used for delivering 6000 gallons of water from River Nile.

Modern development of CSTP began after the oil crisis of 1970 which prompted many countries to investigate further options on renewable energy technologies as an alternative source of energy. Luz International Company of Egypt built the first CSTP in Mojave Desert, California, USA in 1985. The company employed parabolic trough CSTP to focus the heat on the pipes filled with synthetic oil as the heat transfer fluid (HTF). Once the fluid reached a temperature of 375^0C, it was pumped to a power block (also called the power conversion unit, PCU) where it went through a series of heat exchangers, which turned water into steam and powered a steam turbine coupled to a generator for electricity production. Since then many countries have installed CSTP plants to add to their renewable energy generation mix [15].

Spain for example has considerably been involved in the search of alternative sources of energy to replace dependence on the fossil fuels which are running to depletion. Spanish government guarantees a feed in tariff of 26 Eurocents/kWh for 25 years in the endeavour to promote renewable energy technologies. As a result Spain has become the number one country in the world in embracing renewable energy sources from CSTP which is mainly owed to the amount of DNI received of close to 2000kWh/year. It is reported that six Commercial CSTP plants are in operation and about 27 plants still under construction for a total estimated capacity of 1037MWe [16].

2.3 Concentrating Solar Thermal Power (CSTP)

CSTP is a technology of generating electricity from the sun. The sun is a source of renewable energy which is freely accessible. CSTP is one of the technologies best suited for the climate change mitigation in a very affordable way and heavily reduces the dependency on the fossil fuels. CSTP plants utilize the direct sun rays called beam radiation or DNI. DNI is the amount of solar irradiance received on a unit area (Wh/m^2). Most suitable sites viable for CSTP deployment receive an average DNI of 1800kWh/m^2 annually. These areas include South West United States, Central and South America, North and South Africa, Middle East countries, former Soviet Union, China and Australia[17,18].

CSTP can be designed to be solar only or hybridized with other forms of renewable energy when the solar irradiation is low. There are several CSTP projects with different capacities in countries like Greece, Spain, Egypt, Morocco, India, Iran and Mexico. These plants utilize their potential in the Sun Belt regions of the earth. Sunbelt is a region stretching from South to South west of the United States. The main defining feature of this region is its extended summers and mild winters [20, 22].

It is reported that one square kilometer in most arid areas of the world is enough to generate 100-120 GWh of electricity per year which can be equated to a coal plant with a capacity of 50MW operated at mid loads. Mid load operation occurs when the load goes higher than usual but does not reach the peak consumption [18].

CSTP plants are better suited for large scale deployment because the steam cycle applied in CSTPs is more familiar to the engineers. In March 2009, electricity from CSTP in USA was estimated to cost $0.1/kWh over the facility's life while it was $0.26 for PV. Deployment of CSTP is expected to rise as more plants are being installed and technologies are improving [23].
In a CSTP plant, power is generated by tracking radiation from the sun and using this heat to run a steam turbine-generator to produce electricity as shown in Figure 2-1.

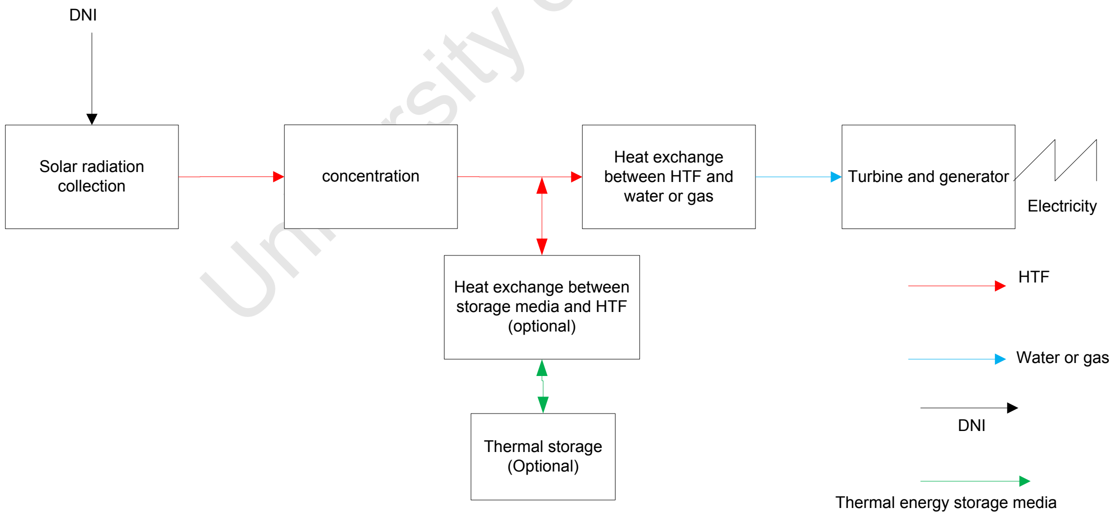

Figure 2-1 Schematic diagram of a CSTP plant [20]

The DNI is trapped by mirrors and then concentrated to heat the Heat transfer Fluid (HTF) running in a tube to the receivers. The HTF can either be oil, molten salts, air or water.
Water or gas can also be used for cooling of the turbine exhaust heat. However, mostly water is preferred as it is economic and has very good thermal efficiency. Water is also used for

mirror washing, condensing of vapour and providing make up water to compensate for any water loss. Different engines are applicable for CSTP such as Stirling engines, gas turbines and steam turbines [19].

CSTP plants can be integrated with Thermal Energy Storage (TES) unit which stores heat energy for later use. The thermal storage enables a smooth electricity production and eliminates fluctuations of energy generation caused by intermittent nature of the sun such as passing clouds. Thermal storage can also be used for generation shifting to hours of peak demands.

The thermal storage medium for CSTP plants is a fluid with high boiling point and has ability to retain heat for quite some time of at least two hours. Mostly molten salts, ceramics or concrete are used as the thermal storage materials. The stored heat is usually used for electricity generation at night or hours of no sun. The TES fluid is stored in the tanks. The energy stored therefore enables CSTP plants to run as base load plants. A base load plant provides a uniform amount of energy to meet the energy demand [17, 19, 20]

2.3.1 Categorization of CSTP plants Technologies

There are four different types of CSTP technologies. These are parabolic trough, parabolic dish, Linear Fresnel and power tower. These technologies differ mainly in the design, direction of concentration (line or point focus) receiver type and the tracking system. Key features of these technologies are listed in Table 2-1.

The line focus uses parabolic mirrors to concentrate the rays of the sun onto the absorber tube in the focal line where the heat transfer fluid is passing. It rotates on a single line tracking the solar energy throughout the day. Point focusing is a three dimensional sun tracking system. It focuses the concentrated heat on a relatively small surface near the focal point. Point focusing achieves very high concentration factor of about 800 compared to the line focus which achieves a concentration factor of 30 to 40 [17].

Point focus uses a large number of mirrors surrounding a central tower also called a power tower. The trapped sun rays are reflected to the central point on the tower where the heat transfer fluid absorbs the heat. The point focus can achieve temperatures up to 500^0C while the line focus goes up to 250^0C. Point focus uses more land than the line focus as it employs thousands of heliostats to reflect the incident sunlight onto the receiver [17]. The concentration ratio C is given by Equation 2-1 [17].

$$C = \frac{I_a L}{\pi d_0 L} = \frac{I_a}{\pi d_o} \text{ .... } \mathbf{2\text{-}1}$$

where

d_0 : Diameter of receiver pipe (m)

L : Collector length (m)

I_a : Parabola width (m)

Table 2-1 Features of CSTP Technologies [17]

One dimensional tracking. The DNI is focused on a linear absorber. This makes tracking the sun simpler	Two dimensional tracking. The DNI is focused on a single point receiver. It achieves high temperatures.			
Linear Fresnel	Power Tower	Fixed receivers are stationary devices that remain independent of the plants focusing device. This eases the transport of collected heat to the power block	Fixed	Focus Type
Parabolic Trough	Parabolic dish	Mobile receivers move together with the focusing device. In both line focus and point focus designs mobile receivers collect more energy.	Mobile	

The four CSTP technologies are briefly described below.

2.3.1.1 Parabolic Trough

Parabolic trough CSTP plant is the most proven of all the CSTP technologies because of the nine large commercial scale solar power plants, which has been operating in California Mojave desert since 1984. They represent a total of 354MW of installed capacity [22, 26].

It is a line focusing system which uses parabolic trough-shaped mirrors to concentrate the solar radiation. Parabolic troughs can be used in two ways. First, they can be used for electricity generation by concentrating the solar irradiation to attain temperatures between 300°C and 400°C. Secondly they can be used for desalination, heating a swimming pool, refrigeration and cooling where the temperatures are maintained between 100°C-250°C [24,27].

Parabolic trough collectors trap the sun along the East-West direction during the day. The collectors are aligned mostly in North-South direction [28].This ensures that there is continuous energy available on the linear receiver. Its solar collectors are modular. The collector arrays can be more than a 100 m long and the curved section can be 5m by 6m.

Steel coating on the pipes carrying the thermal heat collected is allowed to enable high absorption levels of the solar radiation and minimal emission of infrared radiation. These pipe conduits are insulated in an evacuated glass envelope. Parabolic trough needs some blanketing of its pipes because the high temperature generated in the pipes lead to the formation of mist and an explosive mixture with oxygen. Blanketing is done with such gases such as argon, nitrogen and other pressurized inert gases which do not support any combustion [25, 27].

Parabolic troughs are dynamic in a sense that they rotate around an axis called the tracking axis. Collector rotation along its axis requires a drive unit as shown in Figure 2-2 which enables the collectors to track the sun as it cruises in the sky. This is done with the help of the control mechanisms installed for that purpose.

Many collectors are connected in series to make one large collector and are driven by a single unit drive. The choice of drive unit depends on the size and dimensions of the collector. There are two types of commercially available collector control units so far:

- Control units based on the sensors: photo cells are used to detect the position of the sun.
- Control unit based on astronomical algorithm: Determines the sun vector using mathematical algorithms that finds the elevation angle of the sun and the azimuth every second and measures its angular position by some electronic devices.

The oil used in parabolic trough CSTP as heat transfer fluid for temperatures up to 393°C is VP-1, eutectic mixture of 73.5% diphenyl oxide and 26.5% diphenyl. The choice of oil as the heat transfer fluid for this system is dictated by the stability of the maximum bulk temperature oil can handle, because above these temperatures oil cracking occurs.

In parabolic trough CSTP technology storage is either direct or indirect. Direct Storage uses the HTF for steam generation and storage. In the indirect system the HTF is used to generate steam and as a storage media. The molten salt in storage is charged by taking the hot heat transfer fluids from the solar field and running it through the heat exchanger as will be described in section 2.3.1.1.1; Figure 2-3 [28, 31].

The cold molten salts flows from the cold tank to the heat exchanger where it is heated and stored in the hot tank for later use. When the molten salt is needed for generation the reverse happens. The HTF flows in the heat exchangers where it gets heated by the hot thermal energy stored in the tanks.

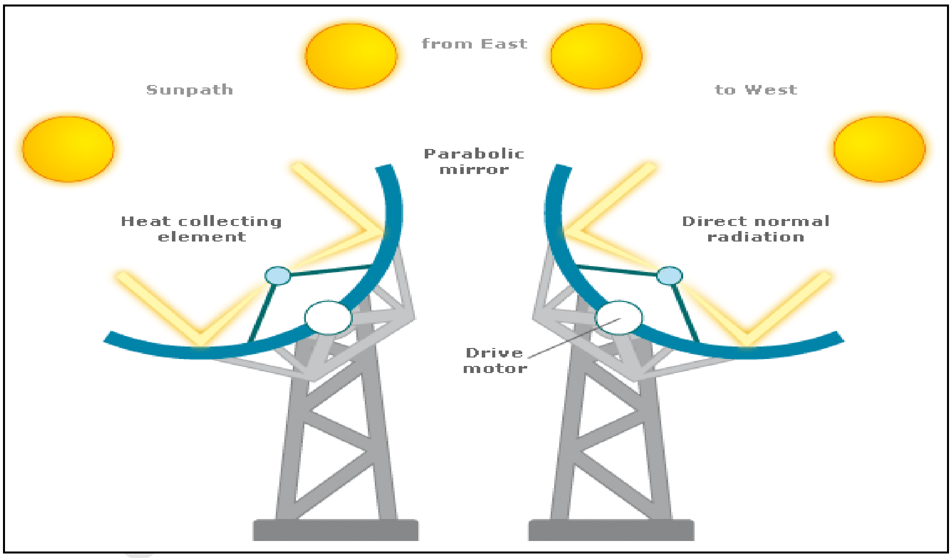

Figure 2-2 Parabolic Troughs [30]

2.3.1.1.1 Electricity Generation with Parabolic Troughs

Parabolic trough consists of four main blocks such as, the solar collector field, thermal block, steam Rankine cycle and a power block as shown in Figure 2-3.

The solar collector system consists of a parabolic trough collector field (mirrors) and the oil circuit for the HTF. The solar collector field collects the DNI and converts it to thermal energy using the circulating HTF.

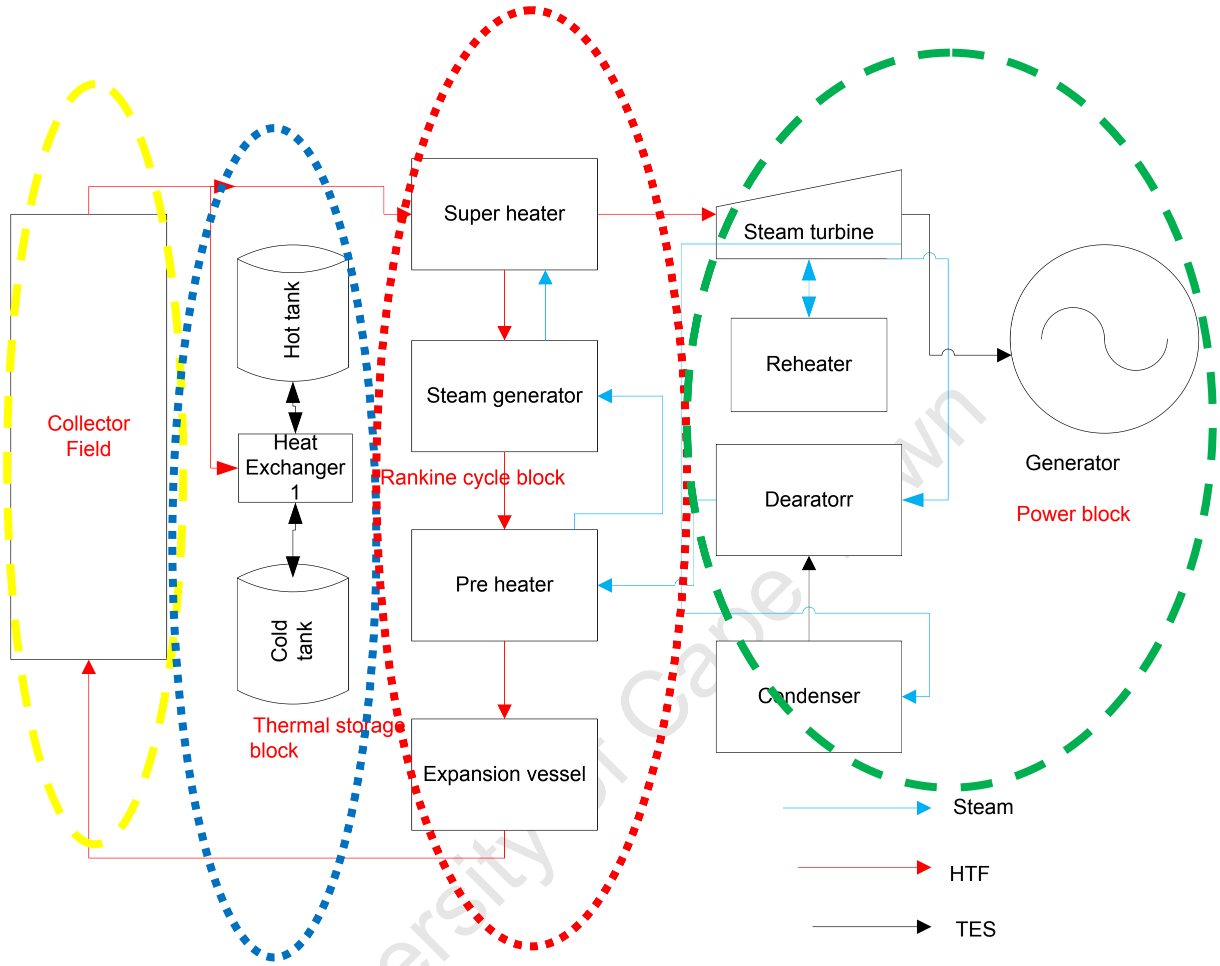

Figure 2-3 Simplified Scheme of Parabolic Trough CSTP Plant Showing main blocks [31]

The thermal block consists of two tanks (cold and hot) and the heat exchangers. The tanks contain the molten salts. One tank contains cold molten salt (290^0C for solar salt) while the other one stores the hot molten salt (390^0C for solar salt). The thermal storage block is responsible for generation of steam for running the turbine when the solar radiation is low or not available. The thermal storage block is the intermediate between the Rankine cycle and the solar field. The exchange of heat between the thermal energy stored in the hot tank and the HTF occurs in the heat exchanger 1.

In periods of high solar radiation the amount of thermal energy collected by the HTF generates more steam from water that exceeds the turbine maximum thermal input. The excess heat carried by the HTF is diverted to the heat exchanger 1 where it is used for raising the temperature of the circulating molten salt from the cold tank to the hot tank. This is called charging of storage. In periods of low solar radiation the reverse happens. The HTF circulates in the heat exchanger 1 where heat exchange between the hot molten salt and HTF occurs. This is called discharging. The discharged molten salt is stored in the cold tank. The heated HTF is used to generate steam from water which runs the turbine generator. The steam generator acts as an interface between the solar collector fields (SCF) and the power conversion unit (PCU) [32].

After discharging the heat to the steam Rankine cycle system the HTF returns to the collector system and the process continues. For continuous generation the parabolic trough is equipped with a two tank molten salt thermal storage system which provides energy back up during hours of no sun.

The steam generator (steam Rankine cycle) consists of three main stages:

1) Pre heater: This is the first stage where water is heated to temperatures close to evaporation.
2) Evaporator: This is the Second stage where preheated water is evaporated and converted to steam.
3) Super heater: This is the third stage where saturated steam produced in the evaporator is heated to temperatures required by the steam turbines. The heated steam develops pressure. The evaporated steam is used to drive the electric generator. The steam attains a temperature of 377^0C at 100 bars of pressure [33].

The power block converts the thermal energy collected in the fields to electrical energy. It is made up of equipments such as turbine, generator, and heat exchangers for transferring the thermal energy collected in the solar fields to the turbine or from the thermal storage to the turbine. The power block is also equipped with a condenser for steam dissipation.

On average parabolic trough CSTP plant generates 1MW for every 8-10 acres of land used. Hence a 100MW plant can use approximately 900 acres of land [34].

2.3.1.1.2 Advantages and Disadvantages of Parabolic Trough

The success of parabolic trough CSTP technology is witnessed by the nine large scale installations in the Mojave Desert in the United States since 1984. The total installed capacity is about 354MW and the gross production recorded is 8,305,477MWe since 1985 to 2011[35].

Advantages of parabolic trough

The parabolic trough plant has the following advantages [36]:

i. It is equipped by a thermal which can supply energy in hours when the direct solar radiation is not available. Thus the radiation collection does not have to be simultaneous with the thermal energy supply.
ii. The solar field can be excluded from the possible disturbances because the storage fluid provides a good thermal cushion and avoids feedback of the disturbances affecting the output of the solar field.
iii. This was done as follows: Parabolic trough concentrating solar thermal power was discovered in the early 1980's. They have an operational experience of over 20 years.
iv. It has a proven net efficiency of 14 % (solar radiation to net electric output).
v. It has been in the market for since 1980's and it has commercially been approved
vi. Hybrid concept proven: Parabolic trough can be hybridised with other renewable energy technologies such as biomass.

Disadvantages of parabolic trough CSTP are [36]:

i. The direct steam generation is still in the development stages.
ii. Low cost and efficient storage systems have not been achieved thus far.
iii. The high winds in the desert may break the mirrors
iv. HTF spillage from the pipes may cause pollution to the environment.
v. The heat transfer increases cost of construction and maintenance.
vi. Parabolic trough utilises oil as a HTF which has a limited thermal capability of 400^0C.
vii. There is significant water usage in parabolic trough that employs the wet cooling method. Cooling of the parabolic trough consumes 90% of the total water used.

2.3.1.2 Parabolic Dish

A parabolic dish CSTP plant concentrates solar irradiation to a single point using a point focus system. The solar irradiation is concentrated on the concentrator along two axes. A reflective glass or a metalized glass reflects the incident ray to a small region called the focus. A Stirling engine is directly mounted at the base of the parabolic dish. The Stirling engine converts the concentrated heat into mechanical energy by compressing a HTF (for this case it is a gas) and then expanding it through a turbine to produce work. There are no means of storage for this plant which makes it less popular as compared to the parabolic trough. It requires continuous changing or adjustment of its position to maintain focus [20, 28].

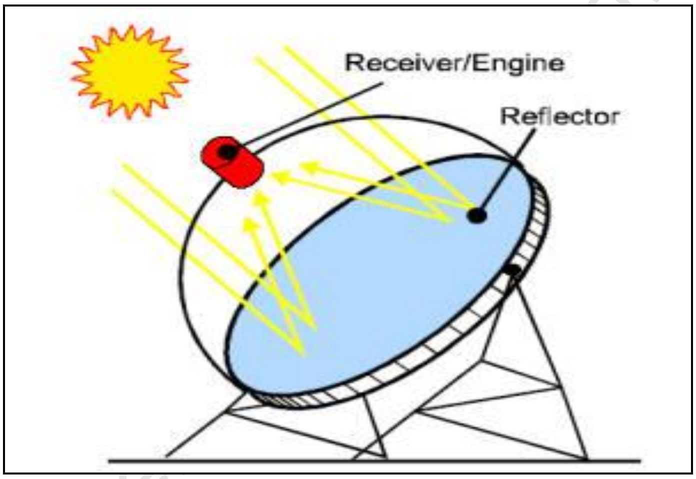

Figure 2-4 Parabolic Dish [28]

Parabolic dish achieves high solar to heat concentration ratio of about 800 and can attain temperatures of up to 1450°C. Its high efficiency enables it to convert 30% of the heat to electricity. This is enhanced by the high radiation concentration on the central receiver unit, serving as energy input to the power conversion system as shown in Figure 2-4.

Parabolic dishes can be installed in any landscape. Dish systems are air cooled and hence can be suitably located in deserts where there is dearth of water supply [19].

This system is mostly applicable to off-grid power generation such as islands and remote areas. Short/medium term key dish/Stirling systems are being looked into, for the option of hybrid dish operation, i.e. supplement combustion of natural gas integrated to the receiver system. This is done to enhance dispatch ability and add to the overall energy produced by the plant. It also ensures off- sun energy generation. This is achieved in two ways:

- ✓ The azimuth elevation tracking: In this case the dish is able to rotate in a plane parallel to the earth's surface and in another plane perpendicular to the elevation.
- ✓ The polar tracking: The dish rotates about the axis parallel to the earth's surface [19].

Technology shortages for the parabolic dish CSTP are reported to be as follows:

- The electricity output of a single dish is limited to small ratings less than 25kWe
- It has not been deployed in large scale.
- No commercial development has been done on it till date.
- Its cost and economic viability have not been assessed.
- The potential for innovations have not been done [19].

2.3.1.3 Power Tower

The power tower CSTP plant uses several mirrors called heliostats for tracking the solar radiation on a central receiver. The sun is tracked on two axes following the azimuth and elevation angles. A HTF which passes to the receiver is heated and used to generate steam [40]. The heliostats are about 120m^2 in area. They are usually curved and the mirrors reflect the sun rays to a central receiver. The receiver on the tower is designed to reduce the radiation and the convectional losses. The steam in the turbine expands and produces mechanical power and electricity. The cold tank molten salts are kept at a temperature of 45°C above their melting point (240°C) [41].

The technical feasibility of the power tower was proved between 1981-1986 by the operation of the six researches or proof of concept of solar power tower plants ranging from 1 to-5 MWe capacities. A single 100MW plant with 12 hours of storage requires 1000 acres of desert land to supply electricity to 50,000 homes [38, 39]. Generally they use 10 to-15 acres per MWe generated. The internal cross-section of the power tower is shown in Figure 2-5. It operates with the same principle as the parabolic trough CSTP and also applies the two tanks model for heat storage.

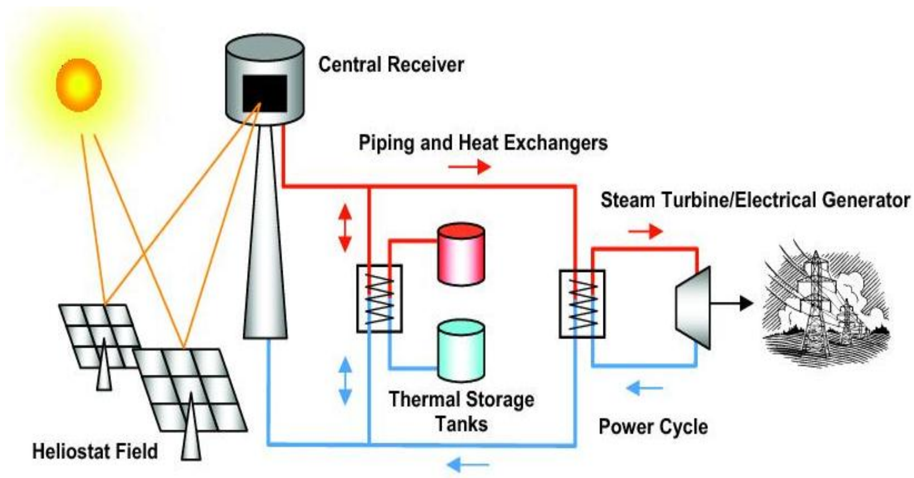

Figure 2-5 Scheme of CSTP Plant with Power Tower [40]

The cited **advantages** of this technology are:

- It has good conversion efficiency of over 40% compared to parabolic trough whose efficiency is about 30%. It attains high temperature of 1000^{0}C.
- Hybrid operation possible. Hybridization with fossil fuel and gas turbine is possible.
- Cooling of thermoelectric plant can be done through air cooling or wet cooling. Air cooling uses air for cooling the exhaust steam from the turbine while wet cooling uses water. In arid regions however water is not available. Power tower is better suited to use air for cooling than parabolic trough and Linear Fresnel.
- Unlike in parabolic trough where the ground has to be levelled for mirror setting, power tower can be located in any landscape [40].

The main disadvantage is that this technology has not been well commercialized.

2.3.1.4 Linear Fresnel Reflectors (Line focus, fixed receiver)

Linear Fresnel reflectors track the sun using one axis. The shape of a linear Fresnel resembles the parabolic trough. They consist of thin mirrors which focus the solar energy on a fixed absorber pipes carrying a heat transfer fluid as shown in Figure 2-6. The name Fresnel originated from a French scientist by the name Augustine Jean Fresnel. It uses flat mirrors which are easy to make [41].

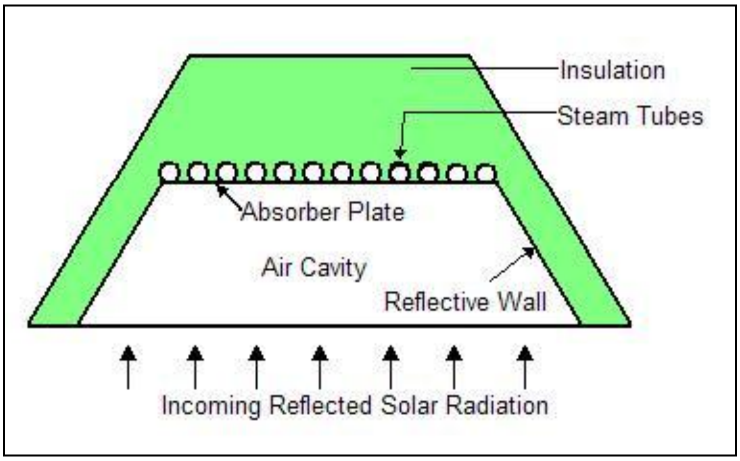

Figure 2-6 Linear Fresnel Showing Concentration of sun rays to heat HTF [42]

The flat mirrors used in Linear Fresnel have a capability of concentrating the solar irradiance 30 times its normal intensity. The concentrated solar energy is absorbed by the heat transfer fluid which undergoes a heat exchanging process to generate steam to power a steam turbine generator. The reflectors are located at the base and reflect the sun rays on a linear axis similar to the parabolic trough.

The absorbers carrying the HTF are located on the focal point of the mirrors. The tubes used for carrying the steam generated by the HTF are insulated with a glass cover to reduce emission losses.

Figure 2-7 shows a compact linear Fresnel reflector. It has two tracking apertures that are used for reflecting the rays of the sun. This technology is also applied in desalination of sea water as shown in Figure 2-8 and electricity generation. Linear Fresnel can also be used as a fuel saver in hybrid plants [41] such as fossil fuels hybrid with linear Fresnel.

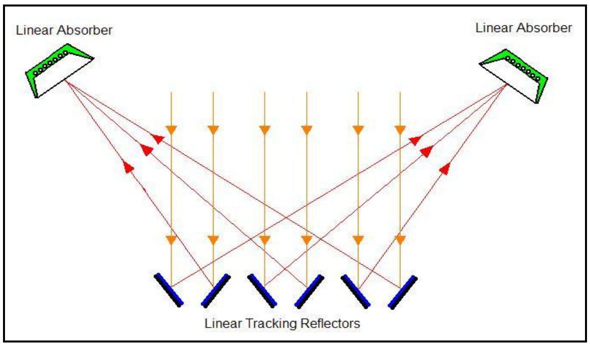

Figure 2-7 Compact linear Fresnel reflectors [42]

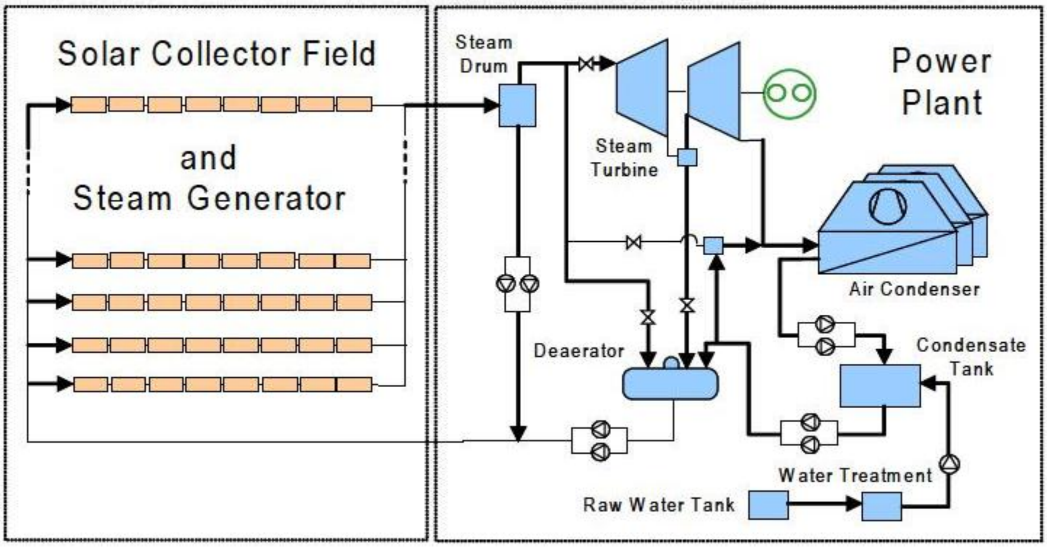

Figure 2-8 Linear Fresnel CSTP system applied to Desalination and Electricity Generation [41]

The main advantages of the linear Fresnel CSTP are:

- The collector system is made of simple cheap locally available materials e.g. scrap glass metals with a minimum number of parts.
- Onsite manufacturing is possible which facilitates local employment generation and development of associated industries.
- Dry cooling can be used in Linear Fresnel and hence reducing water consumption [41].

2.3.2 Exploitation of CSTP in the world

Solar energy is the key resource for CSTP plants but it is unevenly distributed in the world. Figure 2-9 shows areas of the world with the highest potential of CSTP deployment. In this figure Africa has a considerable stretch of land which receives high amount of solar radiation mainly because of the Sahara Desert in North Africa. Some parts of South Africa and Kenya have good DNI pattern.

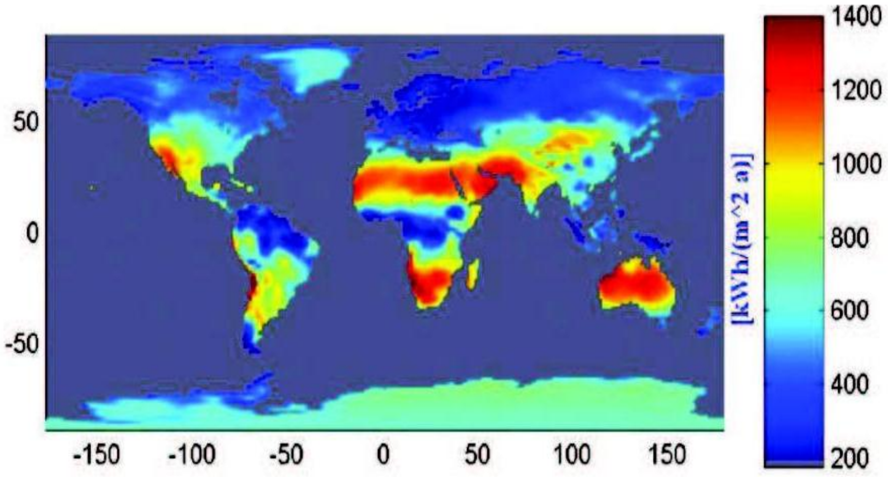

Figure 2-9 Major locations with High DNI pattern [17]

With the formation of the Kyoto Protocol in 1997 to help curb the rising climatic disruption and global temperatures, CSTP plants has since gained popularity in many countries. The protocol was committed to reduce the emissions of the greenhouse gases by 5.2% between the year 1990 and 2008 [43].

This has since then led many countries and regional bodies to move towards achieving this goal. The European Union commited to reduce their greenhouse gas emissions by 8% by the rapid deployment of renewable energy in their energy mix from 6% in 2005 to 12% by the year 2015[43]. This further boosts the adoption of renewable exploration such as wind, solar, tidal and geothermal.

The stakeholders of renewable energy in countries like Spain, Portugal ,Italy ,Greece, Israel,Algeria and Morocco have been supported by their governments through capital grants in the endeavour to promote energy generation through CSTP. This is a very important strategy considering how much they cost during construction [43]. This means all the stakeholders and enterpreneurs who wish to invest in CSTP will have the government support in their business start-up. The investors who would wish to export CSTP energy would be charged cheaper rates by the government for their efforts to promote renewable energy.
USA is the world leader with respect to installed capacity of CSTP plants. Currently the total amount of installed capacity from CSTP in the USA is about 1292MW which accounts for 59% of the total installed capacity from CSTP plants globally.

It is also estimated that by the year 2020 over 2 million homes will be powered by CSTP plants in USA. Major CSTP plants in USA are listed in Table 2-2.

Spain has the second largest installed capacity of CSTP at 746MW as of 2010-2011 and much more is under development. It has been projected that by 2013 Spain will have a total of 1789MW from the CSTP. This has been greatly influenced by the introduction of the feed-in tariff introduced in 2002. Other large scale CSTP projects have been announced in South Africa, Egypt, Mexico, Morocco and United Arab Emirates.

Desertec, a German CSTP project in the Sahara desert, is underway and its reported to announce over 100,000MW of installed capacity. Once completed it will be the biggest energy source accessed from the desert on earth[44].

Table 2-2 Exploitation of CSTP plants in USA [44]

Name and location	Year built	Capacity	Type
SEGS(Solar Electric Generating Systems)I-II,Daggett,CA	1985-1986	44MW	Trough natural gas;3 hours thermal storage system
SEGS III-IV,Krame junction CA	1987-1989	5@30MW	Trough natural gas hybrid
SEGS VIII-IX,Harper Lake,CA	1990-1991	2@80MW	Trough natural gas hybrid
APS Saguaro, Tucscon,AZ	2006	1 MW	Trough
Nevada Solar 1,Boulder city,NV	2007	64MW	Trough
Kimberlina Solar thermal energy project,Bakersfield,CA	2008	5MW	Linear Fresnel reflector
Keahole Solar Project, Kailua-Kona,HI	2008	0.5MW	Micro Trough
Sierra Sun tower,Lancaster,CA	2009	5MW	Power tower

2.4 Exploitation of CSTP Plants in Africa

CSTP is the only renewable energy that is likely to offer both immediate and long term solution to Africa's energy needs. A study done by Greenpeace reports that the generation and importing of electricity from North Africa through Desertec project is likely to become one of the cheapest sources of electricity in Europe. This includes the cost of transmission and distribution. This shows a lot of potential in CSTP in Africa [45].

South Africa has identified some provinces which are potential viable for CSTP deployment. This has been done using the geographical information systems. Many of these provinces have good landscape (7% slopes) and has least threatened vegetation. The energy research Centre 2007, reported that if parabolic trough plants CSTP technologies are deployed, a total of 510.3GW of electricity is possible in the Northern Cape,25.3GW in Free State,10.5GW in western Cape and 1.6GW in Eastern Cape which totals to about 547.6GW[46].

Installation of large scale parabolic trough CSTP plants in South Africa is expected to bring about a major reduction of the GHG emissions in the country. The move towards this has been spear headed by the introduction of Renewable Energy Feed in Tariff (REFIT) of ZAR 2.10 [46].

Algeria has announced to generate 6% of its total energy from renewable energy by the year 2015. This will constitute of 100MW from wind, 170 MW from CSTP plants and 5.1MW from the solar PV. Algeria, a desert a desert region, receives with an approximate DNI of 7.5kWh/m^2/day which is capable of providing 162TWh of energy per year. Algeria has been considered by the Desertec project for generation and transmission of power using parabolic trough CSTP systems to Europe because of its adequate DNI pattern. Adoption of parabolic trough CSTP in Algeria has been boosted by the introduction of REFIT. Algeria currently has one operational parabolic trough of 25MWe [47].

Egypt and Morocco have partnered with the World Bank to finance the CSTP to reduce the dependency on fossil fuels hence the GHG emissions. MENA (Middle East and North Africa) has proved that CSTP is possible in Africa by starting the Desertec project which is aimed at generating and transmitting power to the UK. This will also benefit Egypt and Morocco to help curb about 20,000 and 40000 tons of CO_2 emission respectively. It is also expected to create jobs in the local communities [48].

2.5 CSTP Off Sun Generation

CSTP systems are able to generate electricty during the day depending on the availability of sun hopefully coinciding with peak-demand hours. The incoporated HTF is able to ensure a stable electricity generation for upto 15-30 minutes to endure a passing cloud but at night or during an extended overcast condition, power generation requires one of the three options: fossil fuel or other renewable back up or thermal storage. Many CSTP systems today are supplemented by fuels such as natural gas to meet the base load power generation at all times.

The other alternative is to equip the CSTP plant with a thermal storage which also allows the plant to meet its base load electricity demands without the need for backup systems. CSTP plants with backup can be operated upto 70% of the year as opposed to 15-30% without storage[3].

2.6 Technical and Economic Comparison of CSTP Technologies

Power tower achieves higher temperatures than parabolic trough. The cost per kWe investment is higher mainly because the capital cost for the power tower is higher than the parabolic trough [34].In this comparison parabolic trough has the highest value of installed capacity, followed by power tower, Linear Fresnel and finally the parabolic dish. The reason for maximum deployment of parabolic trough CSTP is that it’s the most commercially proven technology all over the world.

Hybridization of parabolic trough with other renewable energy like biomass and integrated solar gas combined cycle (ISCC) has been proven [16] as shown in Figure 2-10. In ISCC the high temperature exhaust gas is passed through a heat recovery steam generator from which high pressure steam is used in a steam turbine. Additional thermal energy from the solar collector field is injected to the heat recovery steam generator (HRSG) of a conventional combined cycle plant. This boosts the steam production and also the electrical output. Such installations are now on operation and achieve up to 50% efficiency.

Technical and economic comparisons of the CSTP technologies are summarized in Table 2-3 and Table 2-4 respectively.

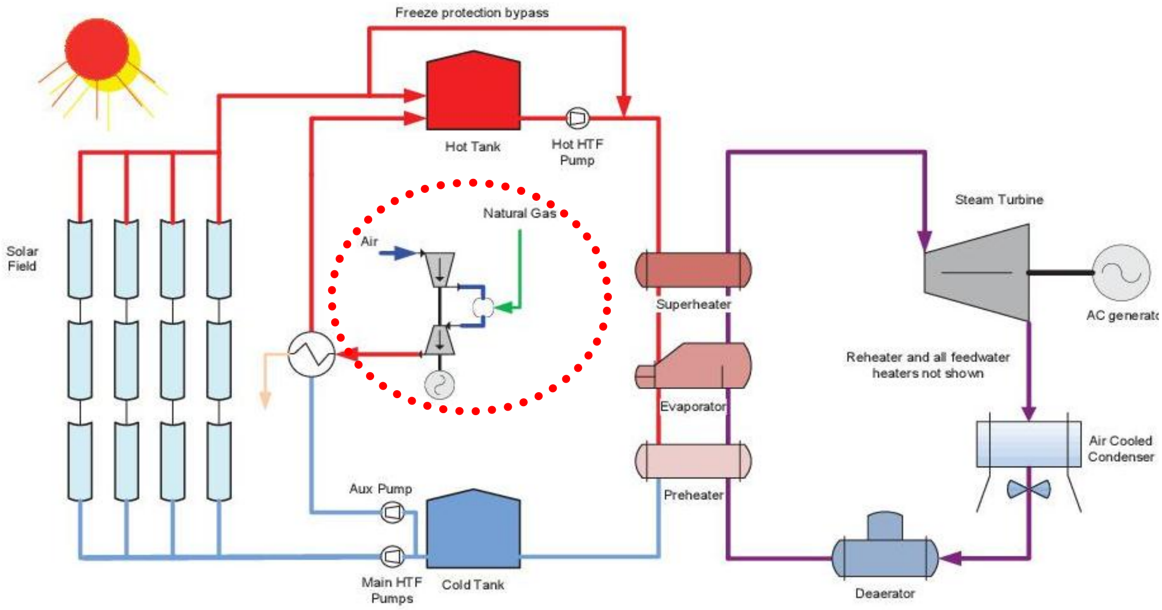

Figure 2-10 Configuration of Hybrid Parabolic Trough with Natural Gas [49]

Table 2-3 Technical Comparison of CSTP Technologies [119]

Technology	**Focus**	**HTF**	**Temperature**	**Hybrid**
Parabolic Trough	Line	Therminol oil, VP-1	400^0C	Possible
Power tower	Point	Molten salts	1000^0C	Possible
Parabolic dish	Point	N/A	750^0C	Still in R&D phase
Linear Fresnel	Line	Steam	270^0C	Possible

Table 2-4 Economic Comparison of CSTP Technologies [119]

Technology	**Cost ($/KWe)**	**World capacity, MWe**		
		Operational	**Under construction**	**Announced**
Parabolic Trough	4,156	570	1550+140 ISCC	5775.1
Power tower	4,500	34	22	1514
Parabolic dish	6,000	0.5	0	1600.08
Linear Fresnel	2,200	8.4	0	487

2.7 Heat Transfer Fluid (HTF)

Parabolic trough has proven excellent energy transfer using VP-1 therminol oil as HTF. This HTF is suitable because of the following reasons:

- High density (1899 @300^{0}C) and viscosity of 3.26 @300^{0}C. It hence does not evaporate easily and can stand high temperatures [50, 67].
- Low vapour pressure (33.1 bars): This is the pressure of vapour that forms above a liquid or solid when it is enclosed. The HTF can hence stand high temperatures achieved by the collectors.
- Moderate specific heat capacity of 1495@300^{0}C: Specific heat is the amount of heat that a given mass of a liquid/solid or gas must absorb to raise its temperature to another level. At 300^{0}C the liquid has fully attained its heat capacity and can generate steam from water.
- Thermal stability: It has thermal maximum temperatures of 393^{0}C. It is reliable even when operated at maximum temperatures.
- Low chemical reactivity: It does not corrode the metal pipes.
- Low cost ($0.49 per kg) [50, 51].

Table 2-5 shows a comparison of Therminol VP-1 with other HTFs. In this comparison it is seen that of all the salts, VP-1 have the lowest freezing point which is very important in the economic analysis of CSTP plants. HTF with high freezing point increases the operation and maintenance costs [50].

Table 2-5 Characteristics of HTF for CSTP [50]

Property	Solar salt	Hitec	Hitec XL	Therminol VP-1
Freezing point(^{0}C)	220	142	120	13
Upper temperature(^{0}C)	600	535	500	393
Density @ 300(^{0}C),kg/m^3	1899	1640	1992	815
Viscosity @ 300(^{0}C),cp	3.26	3.16	6.37	0.2
Heat capacity @ 300(^{0}C),J/kg-K	1495	1560	1447	2319

2.8 Thermal Energy Storage (TES)

CSTP plants depend on the energy of the sun which is intermittent in nature and hence energy storage becomes inevitable. TES is mainly incorporated between the solar field and the steam turbine block in CSTP plants .The use of thermal storage with CSTP plants increases their energy dispatch ability hence reduce the variability and uncertainty of the real net power output [52].This is because heat energy stored can be dispatched when the DNI is low or zero during hours of no-sun or extended cloud covers. Storage in CSTP provides this energy reserve without involving any rotating machines or combustion as done for fossil fuel generator back up. TES hence alleviates the costs of deployment of expensive standby generators as energy backup systems [52]. TES increases the capacity factor of the plant without necessarily using the fossil fuels which pollutes the environment.

The advantages of TES can hence be summarised as shown below

- It enables for energy dispatch-ability without the need of fossil fuels: In this case the CSTP plants are able to generate energy from storage in the hours of no sun.
- Thermal energy storage provides load shifting: The stored energy can be used to meet demand on request. CSTP plants equipped with thermal energy storage can hence supply energy during the peak periods of the day.
- Thermal energy storage increases the total energy output of a CSTP plant. This increases the revenue from the plant.

The principle options for the type of TES used in a parabolic plant depends on the daily and yearly variation of the DNI.

The main options of TES storage are:

2.8.1.1 Buffer Storage

The sun is an intermittent source of energy. Its ability to reach the earth is inhibited by some passing clouds. The buffer storage is applied in CSTP plants in such moments to help augment energy production when the irradiation goes low. In hours of low DNI efficiency of energy production decreases because the turbine operates at part load caused by intermittent nature of the sun. Turbine trips are possible especially if the cloud cover is regular after a short span of time. Buffer storage requires small storage capacities of about one hour maximum [53].

2.8.1.2 Delivery period displacement

It is mostly deployed for big CSTP. It is mostly used for energy shifting in between hours. This is done by storing excess energy collected during the daytime to be dispatched in periods when the demand for energy is high. Delivery period displacement does not increase the solar fraction or the field collection area. The size of this TES storage system ranges from 3-6 hours [53].

2.8.1.3 Delivery period Extension

The purpose of this type of TES is to extend the period of the power plant operation with solar energy. It has a size of 3-12 hours of full load. It occupies big areas of land and also increases the solar fraction than systems without storage [53].

2.8.1.4 Yearly averaging

They require much larger TES lands and solar fields. They are very expensive and still under research [53].

2.8.2 Advantages of TES over mechanical or chemical storage

i. The estimated cost of adding TES to a CSTP plant is low (\$72/kWh) compared to storage batteries which costs \$300/kWh [54, 55].

ii. High round trip efficiencies of 98% [56] mainly because TES does not have to go through any conversion process to be stored or discharged. This is the ratio of the useful energy recovered from the storage system to the amount of energy initially extracted from the heat source.

iii. Provides load shifting to hours of peak electricity demand [57].

2.8.3 Disadvantages of TES

i. TES can only be used to store thermal energy from the CSTP plant and has no capability of storing the electric energy from the rest of the plant.

2.8.4 TES Design

The size of TES is usually dependent on the rating of the CSTP plant. The amount of energy in TES is measured in MWh or specified by the number of hours of storage. The number of hours in this case means the number of hours the plant can be run at its full load capacity using the storage.

There are two types of TES designs:

2.8.4.1 Two –Tank Indirect System

It is made up of two storage tanks, hot tank and cold tank containing a storage fluid mainly a molten salt and some heat exchangers. A two tank storage system uses one tank for the cold molten salt and a hot tank for the hot molten salt. During storing of energy, the hot HTF from the field flows into the oil to salt heat exchanger. The cold molten salt flows from the cold tank through the heat exchanger where it is heated and stored in the hot tank. During discharging the hot molten salt from the hot tank flows to the heat exchangers. It heats the HTF circulating in the oil to salt heat exchanger. The heated HTF is used to evaporate steam which drives the turbine for electricity production. During discharging of TES the reverse happens. The molten salt is used to heat the HTF [58]. Figure 2-11 shows an indirect storage system.

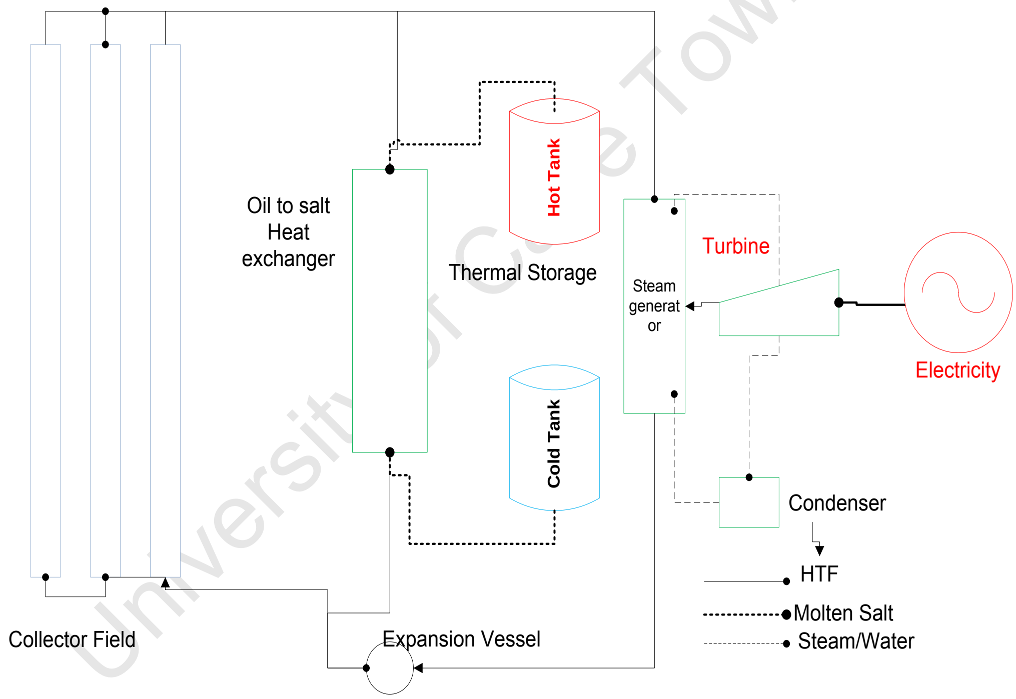

Figure 2-11 Two Tank Indirect System [57]

2.8.4.2 Indirect Single Tank thermo cline System

It is an indirect storage TES design where the fluid used for storage is different with the fluid used for used as a HTF. In this TES design, the hot and cold fluid is kept together. Since the hot and cold fluids are kept in the same tank separation of the two fluids in the main challenge. The hot and the cold fluid remain apart within the same tank because of their density differences. The storage fluid stratifies with temperature. The hot layers remain at the top while the cold layers remain at the bottom of the tank. These systems form the thermo cline systems. Thermal maintenance of the thermo-cline systems requires continuous charging and discharging to avoid mixing of the cold and hot HTF. Thus the single tank reduces the cost because only one tank is involved. This technology is mostly applicable for large scale CSTP which are charged and discharged for multiple hours at full power [56, 57]. A thermo-cline system is shown in Figure 2-12.

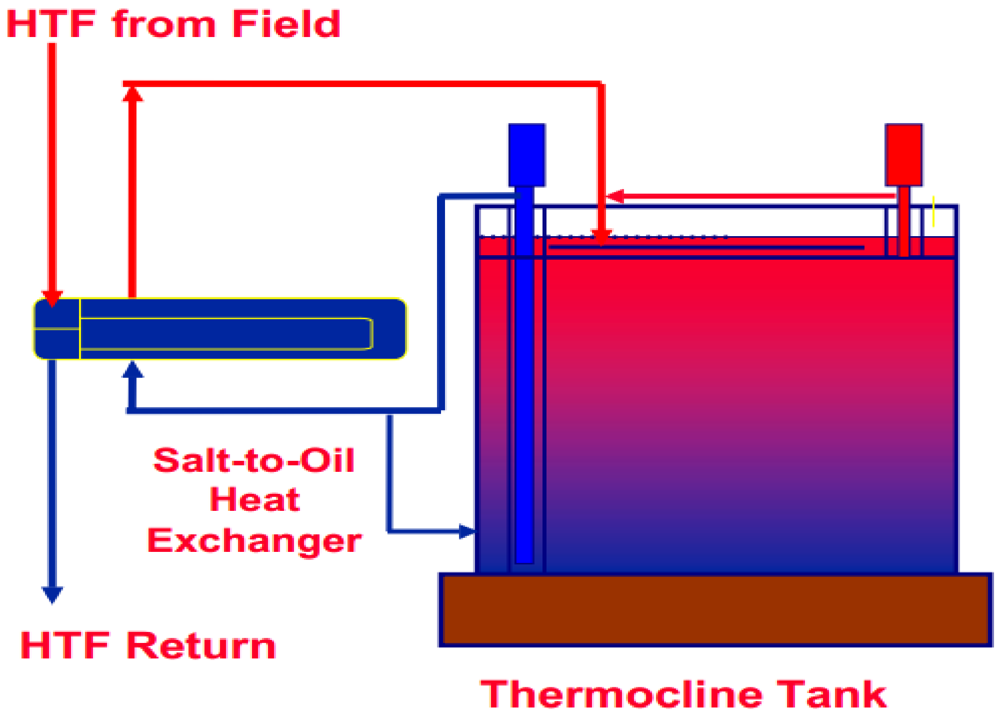

Figure 2-12 Indirect Single Tank Thermo-cline System [57]

2.9 Performance parameters Considered is CSTP Deployment

CSTP technologies are classified on the method of solar concentration using point focus or line focus. In both cases mirrors with high reflectivity are applied for solar concentration to heat water to produce steam for electricity production [59].

The technical parameters considered in CSTP deployment are capacity factor, annual solar efficiency, thermal cycle efficiency and concentration.

Solar efficiency is the ratio of the amount of energy produced to the amount of energy received from the sun as shown by Equation 2-2.

Capacity factor is the ratio of the actual output of a power plant over a period of time and its potential if it had operated at full nameplate capacity in the entire time. The output can also be in terms of hours the plant is in operation for the whole year as shown by Equation 2-3. These parameters depend on the CSTP technology applied [58]. Parabolic trough CSTP technology is most proven. Many of its parameters have been demonstrated and standardised. The point focusing CSTP plants have high concentration ratio but their capacity factor is still in R&D [58].

Performance of CSTP plants depends on some environmental parameters which include available DNI, land terrain and availability, water availability, methods of cooling, wind velocity etc. Table 2-6 summarizes the current performance of CSTP technologies.

The solar efficiency of a plant increases with the ability to concentrate the solar radiation. Point focusing plants have a higher concentration ratio than the line focusing CSTP plants. This is mainly because for the point focused technologies all the mirrors focus the DNI on one point unlike for the line focusing technologies where the DNI is focused on the tube carrying HTF by each of the mirrors arranged in rows [60].

Table 2-6 Performance Data for Various CSTP Technologies [59, 61]

Technology	Capacity (MW)	Concentration Ratio	Peak Solar Efficiency %	Annual solar efficiency%	Thermal Cycle efficiency %	Capacity factor %
Trough	10-200	70-80	21(d)	10-15(d) 17-18(p)	30-40(ST)	24(d) 25-70(p)
Fresnel	10-200	25-100	20(p)	9-11(p)	30-40(ST)	25-70(p)
Power Tower	10-150	300-1000	20(d) 35(p)	8-10(d) 15-25(p)	30-40(ST) 45-55(CC)	25-70(p)
Dish Stirling	0.01-0.4	1000-3000	29	16-18(d) 18-23(p)	30-40 (Stirling) 20-30(ST)	25(p)

(d)=demonstrated; (p) =projected; ST= Steam Turbine; CC=Combined Cycle

$$Solar\ Efficiency = \frac{Net\ power\ generation}{incident\ beam\ radiation} \quad \text{....... 2-2}$$

$$Capacity\ factor = \frac{Solar\ operating\ hours\ per\ day}{8760\ hours\ per\ year} \quad \text{....... 2-3}$$

2.10 Environmental factors to consider in the CSTP Plants Deployment

The main factors considered in the deployment of CSTP plants are as discussed below:

2.10.1 Solar Field area

The size of the Solar Field is determined by the rated power output capacity (MWe). The size of solar field is normally measured by its area. The concept of Solar Multiple (SM) is the most applicable method in the determination of the size of field to be deployed for CSTP plants.

In the design of the CSTP plants the solar field can be sized to meet or exceed the energy requirements of the steam turbine. SM is used for field normalization based on the size of the SF [62]. SM is the ratio of solar field thermal energy output to turbine gross thermal energy demand at design point conditions as shown by Equation 2-4.

Solar field with solar multiple of 1 provides energy sufficient for the power block at its rated capacity [62].

$$SM_{Design} = \frac{Q_{Thermal,\, Field}}{Q_{Thermal,\, Power\ Block}} \quad \textbf{....... 2-4}$$

Increasing the value of SM increases the amount of land area. For example an SM of 2 means that the solar field area is twice as much the area occupied by solar field area of SM=1.

Table 2-7 shows the relative land size for different energy technologies.

Table 2-7 Land Uses for Different Energy Technologies [63]

Technology	Land occupied (m^2 per MWh)
Coal (including pit coal mining)	3700
CSTP	3600
PV	3200
Wind (land with turbine and roads)	1300
Geothermal	400

Utility scale CSTP plant requires large stretch of land as shown in Table 2-8. This may have negative impacts on the existing land uses such as glazing, military uses, mineral production etc. The land occupation of different CSTP technologies shows that line focusing CSTP plants occupies smaller areas as compared to point focusing CSTP plants.

Table 2-8 Land Uses for Different Energy Technologies [59]

Technology	Capacity (MW)	Land use(m2/MWh/Year)
Parabolic trough	10-200	6-8
Linear Fresnel	10-200	4-6
Power Tower	10-150	8-12
Dish Stirling	0.01-0.4	8-12

2.10.2 Impacts on Soil, Water and Air Resources

Construction of CSTP plants requires big land. The land must be cleared and levelled. This increases the chances of soil erosion and change of the drainage pattern of that area. Parabolic trough and power towers are known to consume large volumes of water which is a rare resource in the deserts and semi-arid areas and hence this need might be difficult to address [64].

2.10.3 Ecological Factors

The massive clearing of land interferes with wild animals habitats and forces them to leave a particular area. This may pose danger to their lives. Cutting down of trees affects the rainfall pattern and drainage of an area.

2.10.4 Other factors

CSTP could potentially have some interference on the aircrafts operations if the reflected light beams become misdirected and fall on the plane pathways. The operation of CSTP involves high temperatures that may cause negative impacts to the environment or safety to the workers. CSTP uses molten salts, coolants, hydraulic fluids, lubricants etc which can be very harmful especially if spillage occurs [64].

2.11 CSTP Water Use and Cooling Technologies

Locating CSTP or any other thermal power plants in arid or semi-arid areas with water scarcity would be a problem owing to their huge water consumption. The quantity of water used for generating one unit of electricity shapes the local constraints of CSTP deployment [65].

CSTP generates electricity energy through steam cycles like coal and nuclear plants. The main difference between them is the fuel used turn water into steam. There are two major water processes in a steam turbine cycle. These are:

- Steam cycles
- Cooling process

In CSTP plants most of the water is consumed through cooling [65]. In CSTP water is used in a closed loop steam cycle. The steam is cooled using a condenser and is converted back to water to be recycled.

Two main cooling methods applied in CSTP plants are:

- Wet cooling
- Dry cooling

2.11.1 Wet Cooling

It uses water for cooling the condenser. The waste heat from the turbine flows to the condenser. There is a circuit of water in the condenser which removes the heat from the hot steam. The steam is condensed to some water and heat. The heat is dissipated to the atmosphere as shown in Figure 2-13. The water is pumped to steam generator and the cycle continues. All of the 11 large scale CSTP plants in USA use water cooling [65].

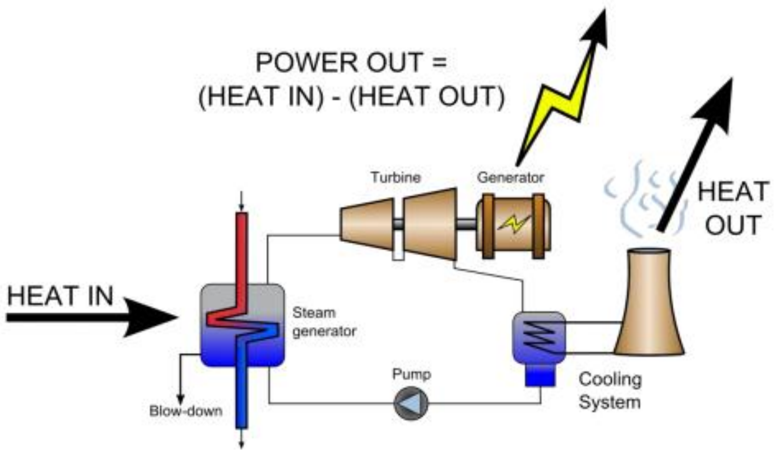

Figure 2-13 Wet cooling system [65]

Wet cooling is an economic and highly efficient cooling technique. In USA it is reported that on average all thermal energy generating plants consumes 470gal/MWh. Parabolic trough plants consumes about 800gal/MWh of water, 2% of which is used for mirror washing [66].

2.11.1.1 Advantages of wet cooling

i. Lower installed costs compared to a dry cooled plant(5% lower compared to a dry cooled plant)
ii. Higher solar to electric conversion efficiency (above 15%)

iii. Lower parasitic losses: The electric parasitic losses refer to the amount of energy consumed by the plant itself to generate electricity. Since water is denser than air, it is able to cool the exhaust steam from the turbine more efficiently than the air. Therefore the parasitic loads are lower.

2.11.1.2 Disadvantages of wet cooling:

i. Water treating chemicals and minerals in the water tend to concentrate after some time. Draining of this water needs to be done to reduce clogging of pipes and rusting. This water is also harmful to the environments as it contains high salts concentration.
ii. In the course of water treatment, minute particles of the treatment chemicals might drift in the ambient air and cause pollution.
iii. Land requirement for a wet cooled plant is much larger than of dry cooled plants. A wet cooled CSTP plant of 250MWe occupies 3500-4500 acre feet of land while for the same plant capacity; a dry cooled plant would occupy 300-500 acre feet of land [66].

2.11.2 Dry cooling

This is a method of cooling the CSTP plants by using dry air. It is reported that dry cooling eliminates 90% of the total water usage in CSTP. In this case the heat from the steam cycle is rejected out to the air. A dry cooled plant would need 80gal/MWh of water mainly for mirror washing. The air is blown to the steam pipes which have convective cooling fins for heat dissipation over their surfaces. Air has a lower capacity for heat carriage than water hence dry cooling is less efficient than water cooling. The massive pumps deployed to extract heat over the surface of the condensers consume part of the electricity generated [65, 66].

2.11.2.1 Advantages of dry cooling

- Saves water, reportedly by 90% hence most applicable in deserts where water is the main challenge [66].

2.11.2.2 Disadvantages of dry cooling

- Higher capital costs: The air cooling condensers add 5% to the capital cost which further increases the cost of electricity by 10%. Inferior performance at high temperatures, for example at 90°F the output energy is reduced to 95% while for wet cooling the output remains at 100% .This is mainly because there must be a temperature difference between the ambient air and the exhaust steam for cooling to occur, which is hard to maintain especially on hot summer days in the dry cooled plant [70].

- Fan noise: Dry cooled plants noise originates from the massive fans blowing air in the condenser. This noise is undesirable to the surroundings.

2.12 Challenges of CSTP Deployment

The main challenges of CSTP deployment in many parts of the world compared to other energy technologies such as wind, hydro etc. are as discussed below:

- **High electricity costs**

The competitive prices of other non-renewable sources of energy are the biggest barrier in CSTP deployment. The cost of electricity production from CSTP plants was reported to be twice expensive compared to the costs of electricity from fossil fuels in 2007. The electricity cost of CSTP plants in 2007 in USA was 15$$cents/kWh as compared to the price of natural gas and nuclear which were 4$ cents/kWh and 7$cents/kWh [43].

- **Technology**

The parabolic dish, Linear Fresnel and power tower are in their early design stages. The overall reduction cost of electricity from CSTP can be done through extensive research aimed at technology improvement of CSTP plants. This would help reduce the construction and maintenance costs involved.

- **Investment in grid infrastructure**

The development of the grid infrastructure will enhance CSTP integration to the grid with other sources of energy. This further enhances the trade of electricity produced from CSTP with other countries. CSTP are mainly located in arid areas which are far from the grid and the transmission lines. Therefore the costs of transmission and distribution of electricity from these arid areas is high. For example, transmitting and distributing electricity from CSTPs in North Africa to Europe is expected to cost about 47 billion GBP by the year 2020 and 395 billion by the year 2050[43].

- ***Large area requirement***

The principle of operation of the CSTP plant is similar to the traditional steam turbines in fossil-fuelled thermal power plants. Typical CSTPs on average occupies 5-10 acres of land per MW of installed capacity. Table 2-9 shows the electricity yield of CSTP plants and the area occupied. It can be observed that though the area occupied by the CSTP plants is large, the respective electric yield is low. A zero storage CSTP plant would occupy 5-6 acres of land per MW which increases to about 8 acres for a CSTP plant with 6 hours of storage. Deployments of CSTP plants have also raised concerns on their impacts on fauna and flora.

Table 2-9 Details of Major CSTP Plants in the World [32]

Name	**Location**	**Energy in MWe**	**Mirror area(m^2)**	**Heat Transfer Fluid**	**Year**
Eurelios	Adrano, Sicily	1	6200	Water steam	1981
SSPS/CRS	Almeria,Spain	0.5	3700	Sodium	1981
Sunshine	Nio, Japan	1	12900	Water steam	1981
Solar one	Barstow, USA	10	71500	Water steam	1982
Themis	Targasonne, France	2.5	11800	Molten salts	1983
CESA	Almeria Spain	1.2	11900	Water steam	1985
SPPS	Shchelkino, Ukraine	5	40000	Water steam	1985
Solar two	Barstow, USA	10	71500	Molten salts	1996

- ***Availability of water***

Water availability is of great importance in CSTP deployment in a locality. A CSTP plant requires water continuously for steam generation. It is reported that a 280MW CSTP plant requires approximately 2.3-2.6 million litres of water per year [43].

2.13 Economic Impacts of CSTP Deployment

Economic impacts of CSTP plants can be viewed in three main ways: These are

- **Direct impacts from construction facilities**

These include temporary engineering, procurement and construction. After construction and commencement, jobs such as permanent operations, maintenance, Engineering and administration are created.

- **Indirect impacts from stimulating secondary economic activity within the region**

These include manufacturing, hospitality and services, infrastructure, ancillary services and commerce.

- **Induced effects arising from changes in income and consumption**

CSTP deployment provides employment to the local people. In Spain it is reported that during the construction of the 100MWe Nevada CSTP plant in 2009, over 2500 jobs were created. Each year in Spain a total of 817 jobs are created through CSTP construction, maintenance and running. This creates an employment index of 2.9 each year. The deployment also attracts the private investments which include plants transmission facilities, ancillary services and infrastructures. This improves the economic status of that particular region and leads to higher income and wealth in the region when new services and products for their private consumption are demanded. Deployment of CSTP plants relieves the country from paying the carbon emission taxes and hence the money that would have been paid can be used for expanding the energy industry especially in the developing countries [68, 69, 70].

2.14 Chapter Overview

This chapter discussed Electricity generation from CSTP technologies putting more emphasis on the parabolic trough CSTP plant. As per the existing research literature it is the most proven technologically, economically in terms cost and land usage compared to its close competitor the power tower. Parabolic dish and Linear Fresnel are not well commercialized.

Water usage which is the main challenge for CSTP plants in the arid areas has also been discussed. In this regard the evaluation of the most economic cooling option (dry or wet) has been investigated. The type of cooling applied at a given plant depends on the availability of water bodies near the plant. It is of concern that for water cooled parabolic trough a total of 800gal/MWh of water is used which is the main reason that is lagging CSTP deployment especially in the African countries that are still in the developing stage. Air cooling eliminates 90% of the water used but increases the LCOE by 2-10% [68].

The technical performance, economic and all parameters used in the CSTP analysis have been discussed. This thesis will use the information gathered from research to investigate and analyze the technical and economic analysis of parabolic troughs CSTP plants deployment plant in Lodwar, Kenya and the most suitable type of cooling for this arid area.

3 Model Development and Simulation

3.1 Introduction

This chapter presents the methodology followed in this research work for modelling and simulation of the proposed parabolic trough CSTP plant in Lodwar, Kenya. It starts off by describing the simulation software used, parabolic trough CSTP working principle and the factors considered in choosing the site in Kenya. The blocks making up the parabolic trough are described and their parameters tabulated for input into the simulation software. The wet and dry cooling are later introduced which forms the main case studies of this research work.

The effect of a CSTP plant on power system stability due to interconnection with other power systems (local power grid) is also described and modelled.

The data used in this research work has recently been used by the National Renewable Energy Laboratories, Sandia National Laboratories and other researchers to help evaluate the economics and performance of parabolic trough CSTP plants.

The following section describes the simulation software used for modelling of the solar parabolic trough CSTP plant.

3.2 System Advisor Model

The simulation software used for this research work is the System Advisor Model (SAM). System Advisor Model was developed by National Renewable Energy Laboratories (NREL) and Sandia Laboratories in Golden Colorado, USA.

SAM is an open source performance and financial software which is designed mostly for facilitating decision making for investors who are actively involved in the renewable energy industry. SAM (https://www.nrel.gov/analysis/sam/) provides yearly models to assist the stakeholders in assessing the performance and cost of PV and Concentrating Solar Thermal Power (CSTP) electricity generation systems. The software has modules that are able to predict 8760 hours performance of different CSTP and PV systems based on design parameters and the climate files that include solar irradiance and the weather data for the selected location. SAM estimates the cost of energy for standalone renewable energy plants based on the design input parameters [71, 72]. The SAM model is as shown in Figure 3.1 below.

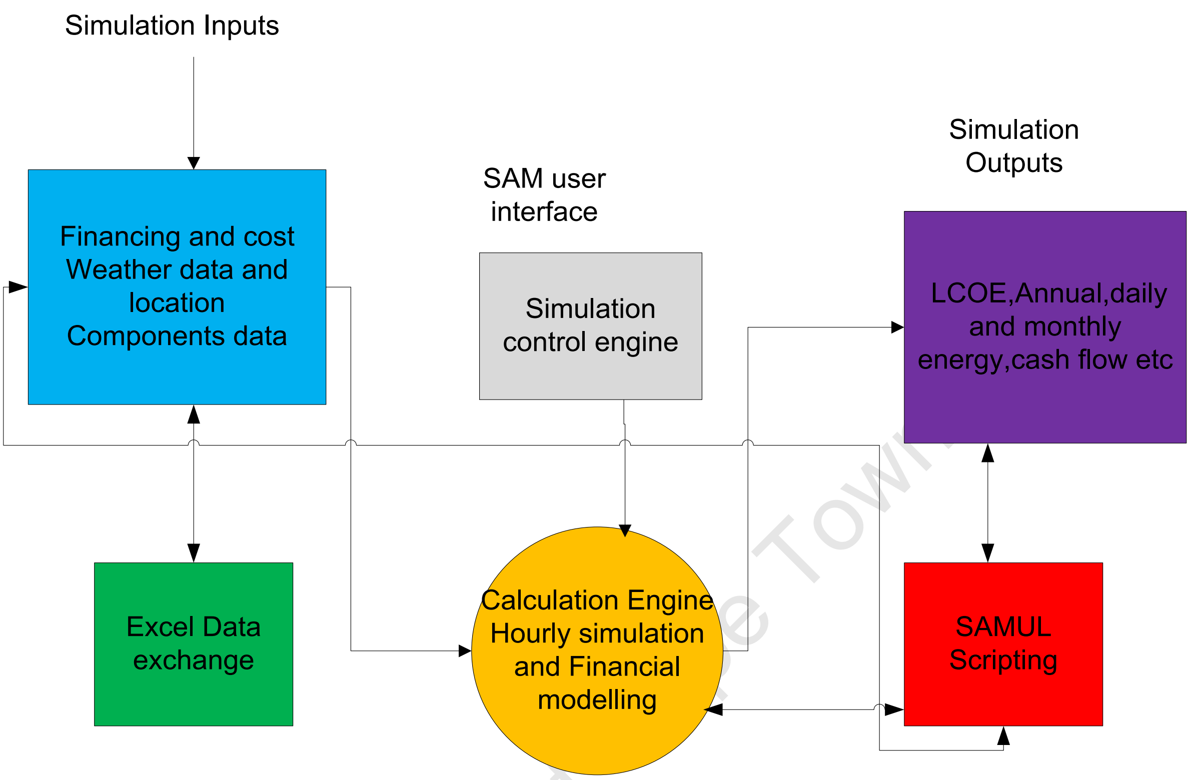

Figure 3-1 Flow chart Illustrating SAM Model Structure [72]

SAM also accounts for incentives which are necessary for the overall performance of the plant. SAM's spreadsheet allows for the exchange of data with Microsoft Excel. The new version of SAM, 2011-06-30, includes the performance analysis of the following technologies:

i. Solar PV systems (flat plate and concentrating)
ii. Solar parabolic trough concentrating systems with and without molten salt storage
iii. Power tower concentrating solar power systems (molten salt and direct heat)
iv. Solar water heating for heating residential houses or for commercial application
v. Electricity production from Geothermal resources
vi. Power generation from biogas [49, 72].

The levelized cost of electricity (LCOE) which is the main metric tool used in the determination of financial viability of different renewable energy technologies based on input parameters is described in the following section.

3.3 Levelized Cost of Electricity (LCOE)

LCOE is a metric tool used for weighing options and technology comparison of the energy generation technologies. LCOE($/kWh) accounts for the capital costs involved in the plant development and all the other costs involved in running the plant over its entire lifetime. It is especially very useful in comparison between the grid connected plants and standalone application.

LCOE is defined as the total costs of a system over its entire lifetime divided by its annual energy output multiplied by its useful lifetime (availability) as given by Equation 3-1[72].

$$LCOE = \frac{\text{a x Annualized capital costs} + \text{Annual O \& M costs}}{\text{Annual Energy generated x plant availability}} \quad \textbf{....... 3-1}$$

Where

a = the annuity factor defined as the present value of a project that earns one dollar for a given number of years. Annuity factor is described by Equation 3.2 below.

$$a = \frac{K_d(1+K_d)^n}{(1+K_d)-1} + K_{insurance} \quad \textbf{....... 3-2}$$

Where

K_d =Interest rate charged on borrowed loans

$K_{insurance}$=Annual plant insurance

n =Depreciation period in years: This simulation assumed a 30 years plant life time, shown in O &M: operation and maintenance costs.

The calculation of LCOE involves all the costs incurred in the daily running of the plant including operation and maintenance costs, fuel cost and capital cost. LCOE is the minimum price at which energy must be sold to break even. It is also used to assess the economic viability of a plant. The LCOE model is shown in Figure 3-2. In this simulation LCOE is used as the main metric to compare the dry and wet cooled plants.

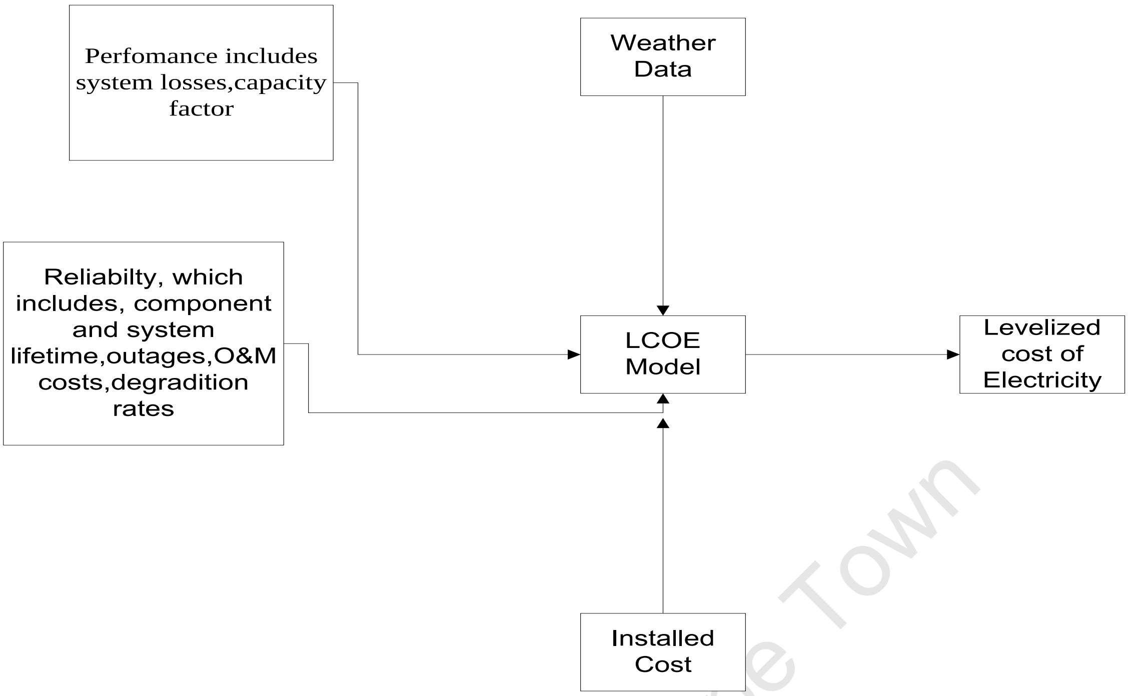

Figure 3-2 LCOE Model [72]

The following section shows the assumptions made during simulation and modelling of the parabolic trough.

3.4 Scope, Assumptions and limitation in Simulation and Modelling

The design parameters for the parabolic trough CSTP plant modelled in this research work are as follows:

- 55MWe gross (or 50MWe net) steam Rankine cycle power plant
- LS-3+ solar field collectors using VP-1 therminol oil as the heat transfer fluid (HTF) at 393^0C.
- The fluid used in the thermal energy storage is the solar salt.
- A design solar field thermal output to power block input ratio (Solar multiple) of 2.0.
- Thermal energy storage of maximum 12 hours
- The investment costs of dry cooling are 5% more than wet cooling
- The load profile used for comparison of hourly energy production is the same for dry and wet cooling

The following section describes the benefits of parabolic trough installation.

3.5 Benefits of parabolic trough

It was mentioned in the literature that parabolic trough is currently the most proven CSTP technology and therefore its operating parameters have been standardised. The benefits of parabolic trough plant are listed below.

i. ***Daytime Peaking***

Parabolic troughs are reported to have a very good daytime peaking capability. They are able to generate peak electricity during sunny hours when air conditioning and fans are at their peak usage [22].

ii. ***Environmental impacts***

Compared to other thermal plants like nuclear or coal parabolic troughs have no emissions of greenhouse gases [22].

iii. ***Economic impacts***

Construction of the plants in a particular site helps to create local jobs, businesses and industries. They also provide permanent employment in the running of the plant after commissioning. Some materials used for construction of the plant can be locally manufactured and this boosts the economy [22].

iv. ***Hybridization with other Renewable Energy technologies***

Parabolic trough can be hybridized with other renewable energy technologies like biomass in order to ensure round the clock generation [16].

3.6 Principle of Operation of Parabolic Trough CSTP

This section present the description and principle of operation of parabolic trough CSTP plant including modelling and simulation aspects.

Figure 3-3 shows the four main blocks making up the parabolic trough CSTP plant. These main blocks are the solar field, steam Rankine cycle block, thermal storage block and the power block. The solar field is made up of the collectors and the receivers. The collectors track the DNI from the sun in a single line axis in an East to West direction. The collected DNI is linearly focused on the receivers which pass on top of the collectors carrying the HTF for this case, Therminol VP-1 oil.

In the steam Rankine cycle heat exchange between the HTF and water occurs.

The heated HTF circulates through the heat exchangers in the thermal block where it is used to evaporate water into steam. The steam is superheated and pumped to the power block where it rotates a steam turbine coupled to a generator to produce electricity. The steam temperatures and pressure are maintained at 377^0C at 100 bars of pressure [22].

The waste steam is condensed back to water and pumped again to the heat exchangers for steam generation. Cooling of the condensers can be done either by water or air. The HTF is pumped back to the collector field for heat collection and the cycle continues.

Energy is stored during hours of high radiation when a lot of thermal energy is generated than the power block demands. Parabolic trough allows for energy storage in the hot tank to be dispatched when there is no sun or during hours of low irradiance [67].

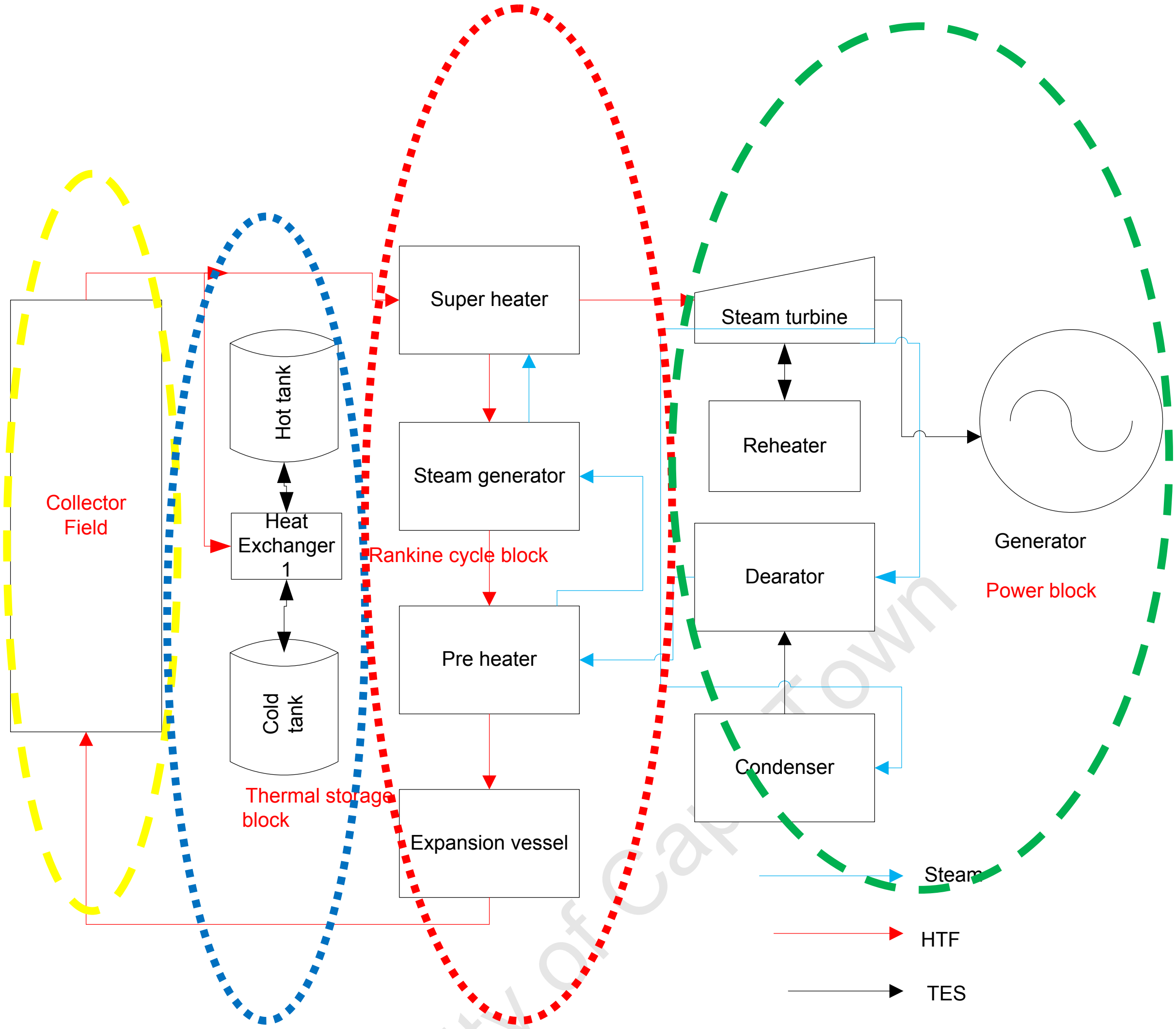

Figure 3-3 Solar/steam Rankine Parabolic Trough System Schematic [22]

The following section describes the parameters for assessing parabolic trough CSTP locations in Kenya.

3.7 Parameters used to Assess Parabolic Trough CSTP Locations in Kenya

Kenya's geographical location is highly favourable for the large-scale deployment of concentrating solar thermal power technologies. Especially some parts of the Eastern province of Kenya such as Lodwar, has over 4500 hours of sunshine per year with an average solar radiation of 1836kWh/m^2 [34,72].The areas marked black on the map of Kenya in Figure 3-4 shows the hottest parts which lie on the Eastern part of the country.

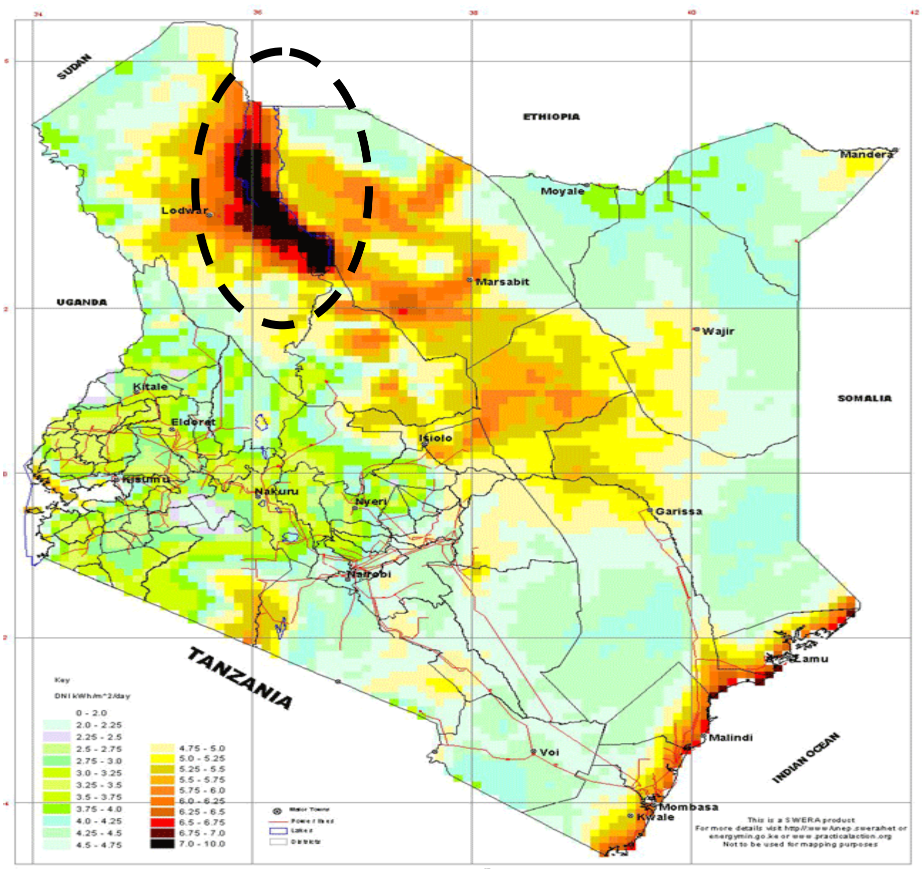

Figure 3-4 Map of Kenya showing good DNI Sites [76]

The main parameters considered in selecting the parabolic trough CSTP sites in Kenya for the modelling and simulation are discussed below.

3.7.1 Solar Resource in Kenya

A suitable location for CSTP plants requires a good solar irradiation of over 1800-2000kWh/m^2 (5-6.5kWh/m^2/day) annually [74]. The energy output from CSTP plants located in areas with good DNI pattern is high which ensures cost effectiveness of the plant operation. This is because revenues from CSTP are obtained from selling the energy generated. In such locations the atmospheric moisture should be low and almost no cloud cover throughout the year [77].

The reasons considered in locating the parabolic CSTP plant in Lodwar, Kenya and the corresponding maximum irradiation at design are described in the following section.

3.7.2 Geographical and Environmental Location of the Simulated Plant

Amongst many of the deciding factors that affect the productivity and efficiency of a parabolic trough plant, the most important one is the geographical location of the plant.
Fluri et al [102] described the environmental factors to be considered during parabolic trough CSTP deployment in a given geographical location. His findings were compared with the environmental factors of Lodwar as shown in Table 3-1 below.

Table 3-1 Summary of Environmental factors considered when Locating a CSTP Plant [102]

Consideration factors	Criterion	Lodwar
Direct Normal radiation (DNI)	Annual range of DNI >5kWh/m^2/day	Lodwar receives DNI greater than 5kWh/m^2/day
Vegetation	The parabolic plant must lie in an area where the vegetation in that area is defined as "least threatened"	Most parts of Lodwar have little or no vegetation.
Slope	The slope percentage will preferably be less than 1%	Lodwar being a Desert is a flat area.
Land usage	Recommended to use marginal or fallow land	Lodwar is a marginal land
Water usage	Reduce water usage in water scarce water areas by implementing other cooling methods in order to save water	Wet and dry cooling types are investigated

Majority of the Kenyan land is agricultural based. There are also forests which are the main sources of woody biomass for lighting and cooking. Thus only the waste lands which are unsuitable for agriculture and human settlement, can be considered as sites for the parabolic trough construction.

It is estimated that Kenya has $1.26902X10^{11}m^2$ of such wasteland in the Eastern part of Kenya. Assuming 1% of the whole waste land is deployed for parabolic trough CSTP plants, $1.25632X10^{11}m^2$ of the land still remains, which indicate that land availability will not become a problem in the future. Assuming 1MWe per day is generated by a mirror area of $20{,}324m^2$, 1% of the total arid area in Lodwar Kenya can generate about 6,243 MWe per day [34, 75, 77].

SAM is used to analyse the different DNI, wind and temperature of the main hot locations in Kenya. Lodwar is chosen because it has the highest DNI of $1836kWh/m^2$/year and an average wind speed of 4.0 m/s as shown on Table 3-2 below. It also has a high average annual dry bulb temperature of **29.7^{0}C.**

Table 3-2 Possible CSTP Locations in Kenya [72]

City in Kenya	Elevation in meters (m)	Latitude in degrees	DNI (kWh/m^2)/year	Average Dry bulb temp(^{0}C)	Average Wind speed (m/s)
Kisumu	1146	-0.1	1570.8	22.9	2.9
Lamu	6	-2.27	1505.6	27.1	4.1
Lodwar	**515**	**3.12**	**1836.9**	**29.7**	**4.0**
Malindi	23	-3.23	1419	26.4	4.6
Marsabit	1345	2.3	1721	20.1	8.9
Meru	1554	0.08	1525.2	18.3	2.5
Mombasa	55	-4.03	1363.1	25.9	3.5
Narok	1890	-1.13	1389.3	17.3	3.5
Nyeri	1759	-0.5	1238.8	17.4	2.2
Voi	579	-3.4	1309	24.4	3.3
Mandela	231	3.93	1474.6	29.5	4.5

The monthly DNI, wind speed and dry bulb temperatures of Lodwar are shown in Table 3-3.

Table 3-3 DNI and Dry bulb Temperature of Lodwar [72]

Month	DNI(Wh/m^2)	Dry bulb Temperature(^0C)	Wind speed(m/s)
January	222.46	29.539	3.6909
February	216.95	30.304	3.5854
March	180.55	30.610	4.8980
April	168.62	30.081	4.5800
May	178.35	30.058	3.2690
June	209.92	29.352	3.8753
July	219.44	28.527	4.6449
August	237.06	29.102	3.4749
September	267.42	29.889	4.2060
October	212.45	29.836	4.0008
November	190.68	29.941	4.3360
December	213.08	29.637	3.0950

The average daily and monthly DNI of Lodwar are as shown in Figure 3-5 and Figure 3-6. As shown in these two figures the highest DNI occurs in the month of September and lowest in April. Figure 3-7 and Figure 3-8 shows the average monthly dry bulb temperature and wind speed respectively. The amount of DNI that reaches the ground depends on the amount of interference in the sky. These interferences include wind speed, thickness of ozone layer and humidity. For example, the month of March receives the highest dry bulb temperature, yet it has a lower DNI than September. The reason is that in March, Lodwar records the highest wind speed which interferes with the DNI. The wind conditions should also be considered so as to determine the structural design of the plant.

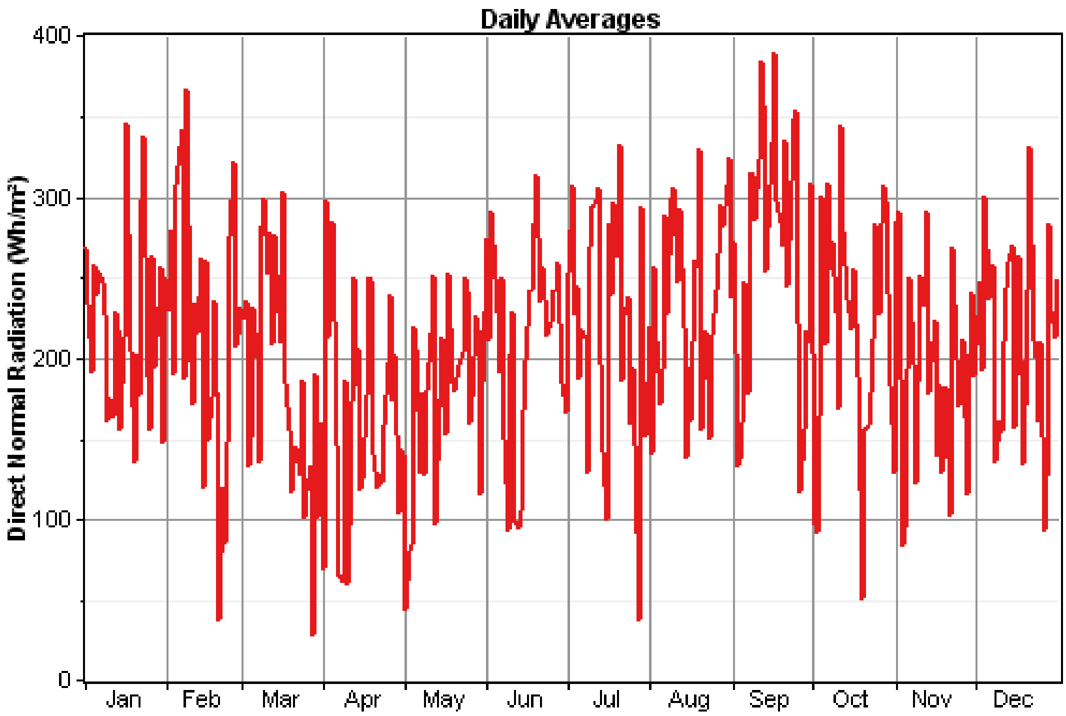

Figure 3-5 DNI (Wh/m^2) per day of Lodwar[72]

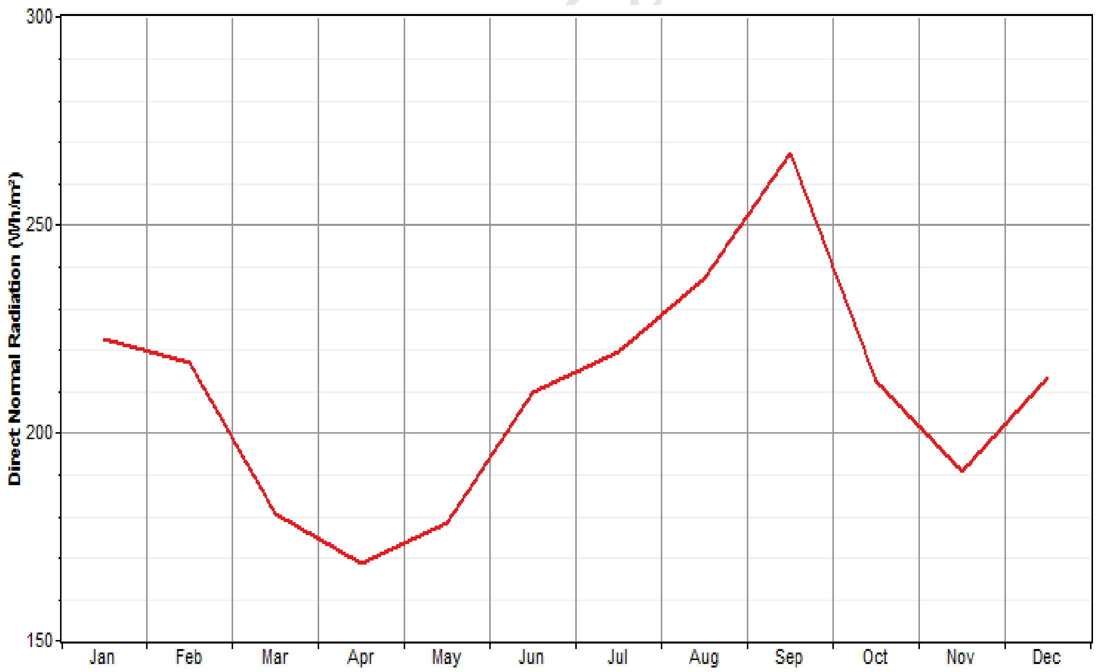

Figure 3-6 Monthly DNI ((Wh/m^2)) of Lodwar[72]

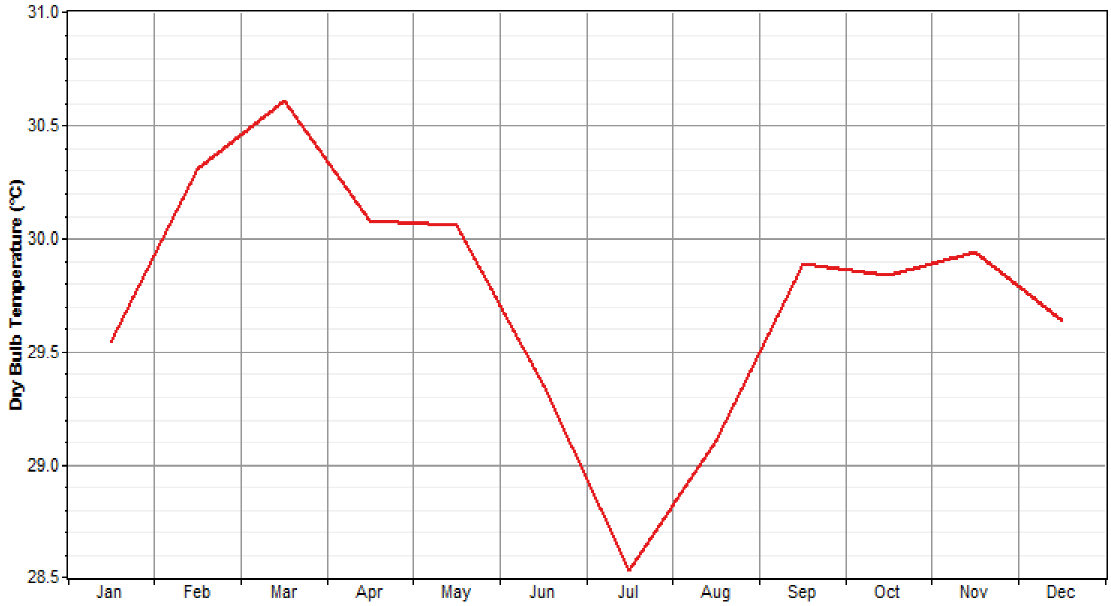

Figure 3-7 Dry Bulb Temperature (^{0}C) of Lodwar[72]

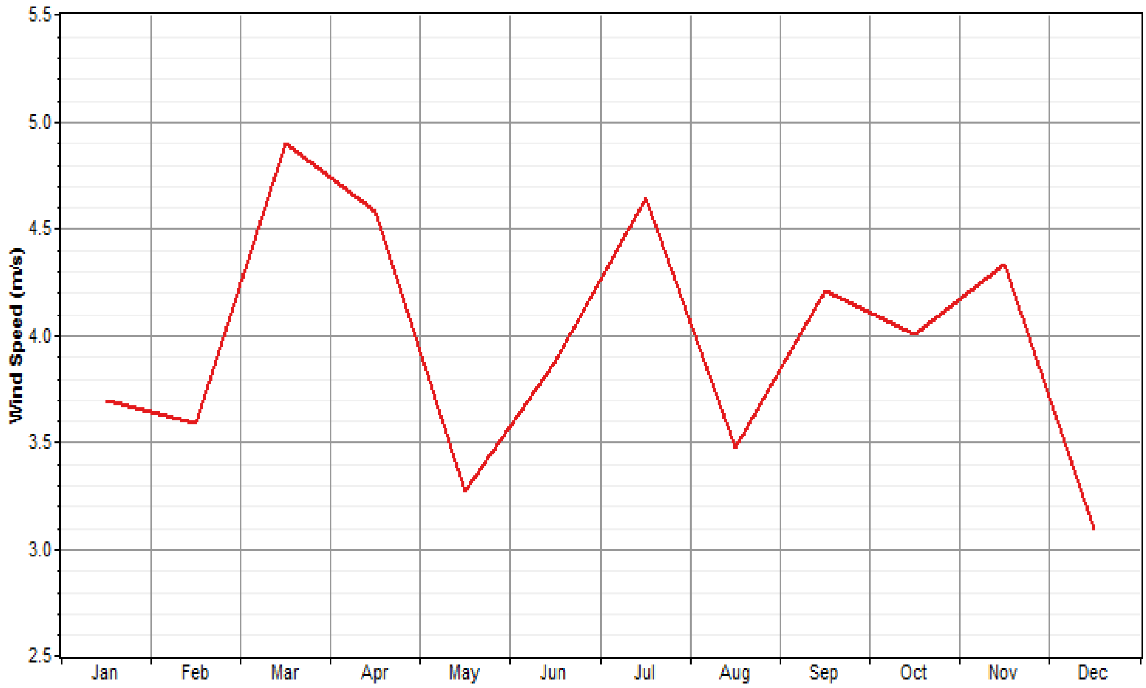

Figure 3-8 Wind Speed (m/s) in Lodwar[72]

In order to determine the daily maximum solar irradiation of a Lodwar design irradiance is determined. The following section describes the maximum irradiation at design.

3.7.3 Design Irradiance

The irradiation at design is the highest amount of energy that can be collected per square metre per day in a certain area. In order to determine design irradiance for the parabolic trough CSTP plant the maximum cosine adjusted DNI is determined. In this case the site of the plant is chosen to be Lodwar, Kenya. The collector tilt is set at zero degree which is horizontal to the equator. The collector azimuth is set at zero degree which assumes midday hour at the time of testing. The solar multiple and the number of storage hours are set at 2 and 12 respectively (*Storage and solar multiple are discussed in sections 3.12 and 4.8 respectively*). The simulation is run. The maximum irradiance at design for Lodwar is obtained to be 958.838W/m^2/day and is used in this research. It occurs in the month of September. This is shown in Figure 3-9.

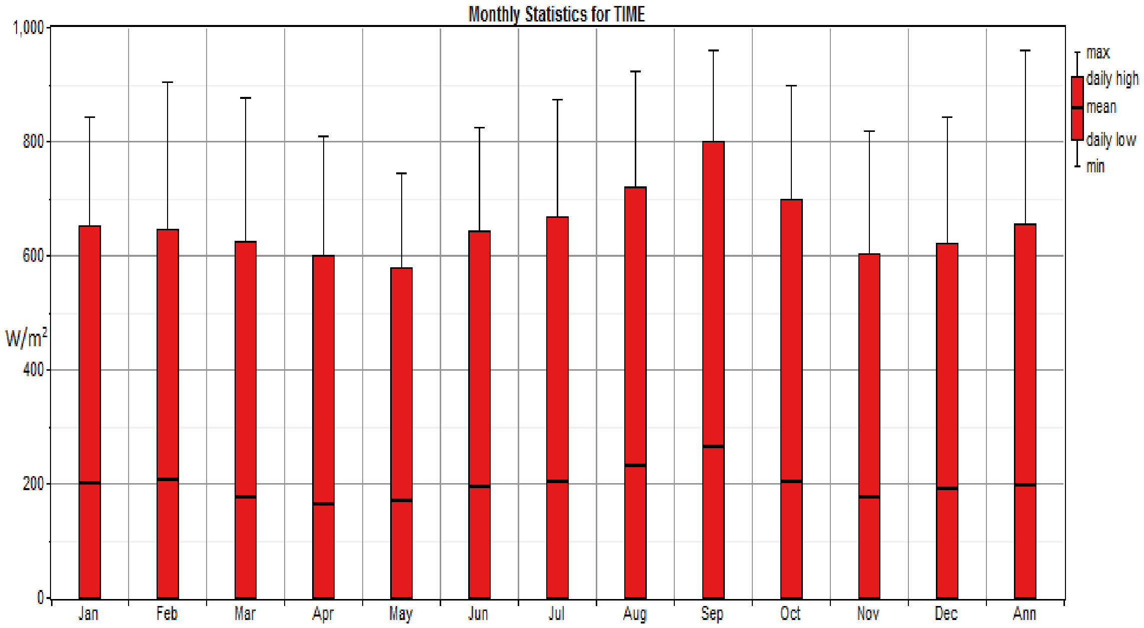

Figure 3-9 Collector Adjusted DNI for Design Irradiance

The following section describes the design of a solar parabolic trough plant.

3.8 Parabolic Trough Plant Design

This section looks at the different parts of a parabolic trough CSTP plant and its design parameters considered in the simulation. The main parts include the solar field, thermal energy storage, steam Rankine cycle, and the power block. These blocks are interconnected as shown in Figure 3-10 below.

The solar field is made up of the collectors and receivers. The thermal block consists of two tanks and a heat exchanger. The power block is made up of the steam Rankine cycle and turbine. The steam Rankine cycle is the main component of the power cycle. It is made up of heat exchangers and the turbine.

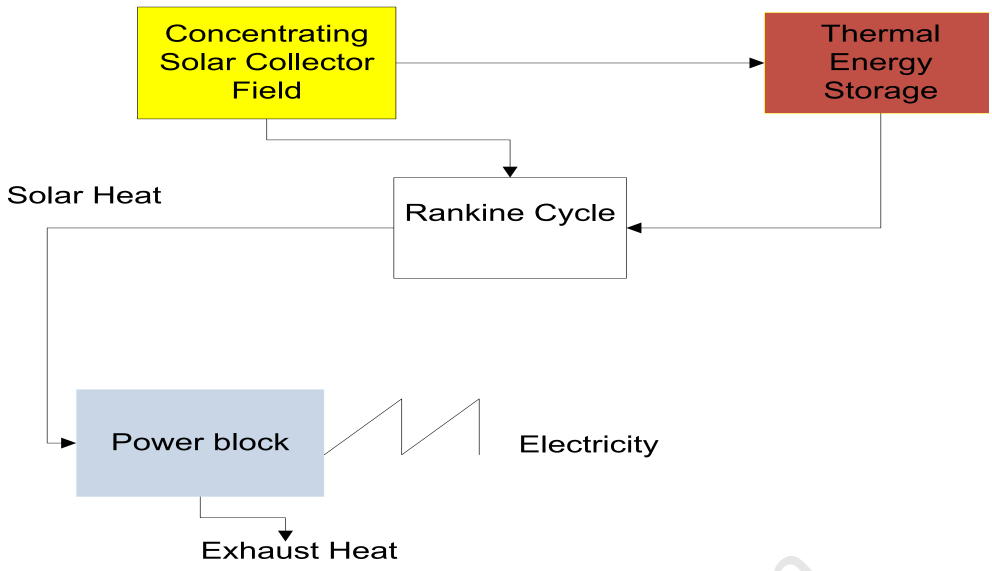

Figure 3-10 Interconnection of the main Parabolic Trough CSTP Blocks [22]

The energy conversion from solar irradiation to electricity through the different blocks is described on Figure 3-11.

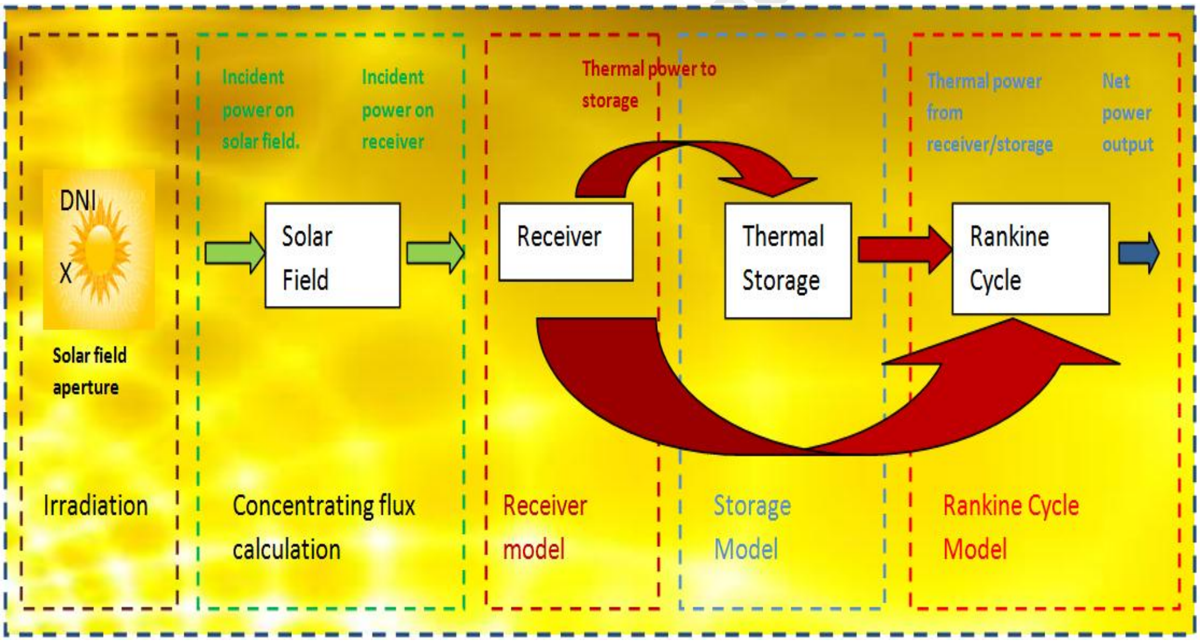

Figure 3-11 Parabolic Trough CSTP Simulation Model [22]

Each block making up the parabolic trough CSTP is discussed in the following sections.

3.9 Solar Collector Field

The solar collector field is the heat collecting element of the parabolic trough plant. The solar field consists of solar collectors which are parabolic in shape. They are used for focussing the heat energy from the sun on the receiver tubes carrying HTF. The collectors used in this simulation are LS-3 parabolic trough collectors. They have a solar concentration of 71. Solar concentration is the ratio of the width of the parabolic trough to the diameter of the receiver pipe as described in section 2.3.1, Equation 2-1. They have been used in the most successful parabolic trough plants building called the SEGS VI in the Mojave Desert, California generating 30MWe [22, 74]. Each collector has an aperture area of 545m^2 and aperture width of 5.75m. The actual field aperture area to operate a 50MWe parabolic trough at a solar multiple (size of field) of 2 in Lodwar is found to be 427,280m^2.

Solar Multiple (SM) is the ratio of solar field maximum thermal energy output to power block gross thermal energy input at turbine design point conditions. In this case SM defines the solar field size as a multiple of the capacity size of the turbine energy output as discussed in section 4.8, Equation 4-30.

The number of collectors used was 784. The collectors are arranged in rows. The distance between rows of collectors is 15m. The rows form collector 98 loops. Each loop has 16 collectors as shown in Figure 3-12. The collectors are supplied with equal amount of HTF.As shown in this figure the blue colour represents the cold HTF pumped to the collectors from the steam Rankine cycle/power cycle, while the red colour represents the hot HTF flowing from the collectors to the steam Rankine/power cycle. All the collectors are supplied with the equal amount of HTF.

The collectors in the same loop are supplied with HTF on the same header.

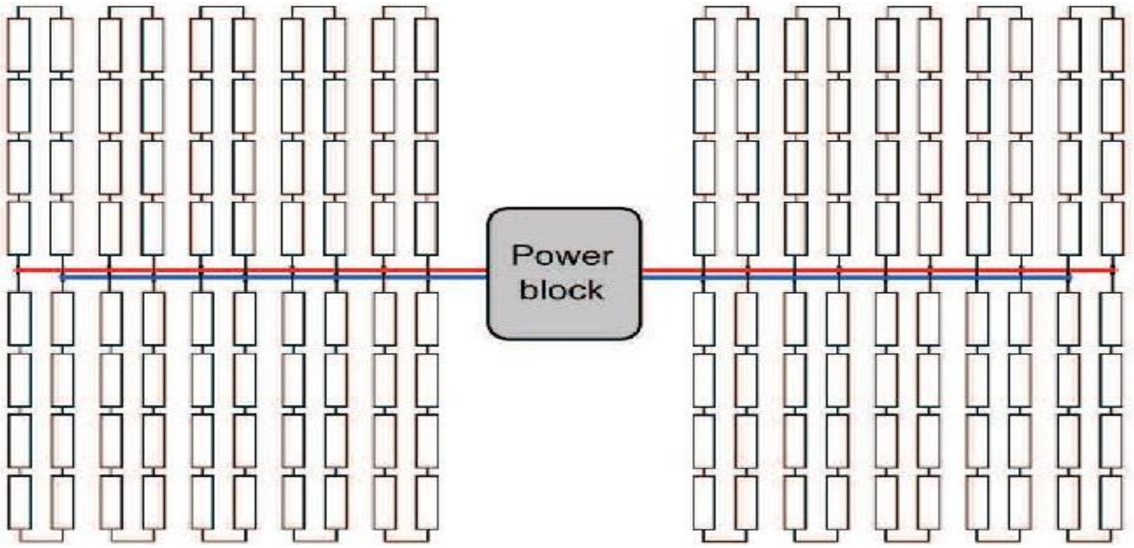

Figure 3-12 Alignment of the Solar Field [72]

The net amount of energy absorbed by the collector field depends on the temperature of the solar field, direct normal irradiance (DNI), velocity of the HTF in the current hour, ambient temperature and the wind speed. This is shown in Figure 3-13 below:

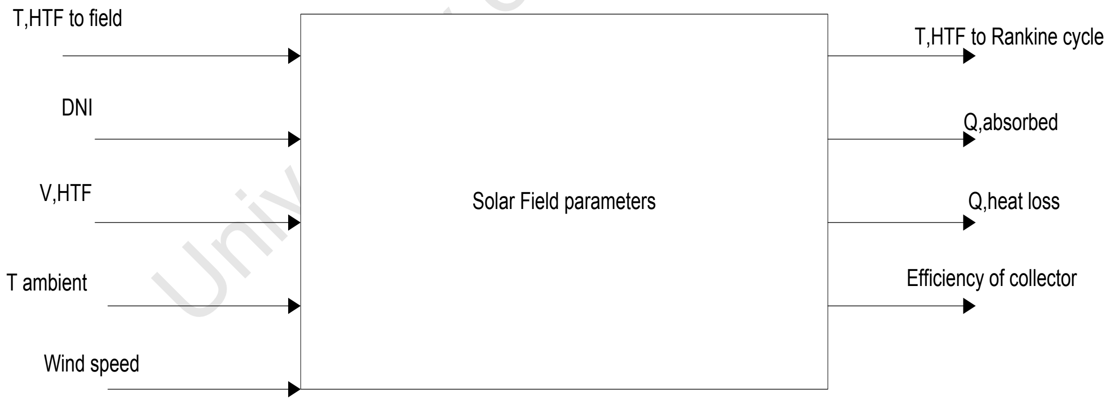

Figure 3-13 Flow diagram of the collector [72]

where

T, HTF to field: This is the temperature of the HTF fluid entering the solar field (^{0}C).

DNI: This is the amount solar radiation received per unit area on a collector (Wh/m^{2}).

V, HTF: This is the velocity of the mass flow rate of HTF in the collectors (kg/s). The velocity depends on the temperature at that particular time. When the temperatures are higher the speed increases.

T, ambient: The temperature of the ambient air (^{0}C). The amount of energy yield on a collector depends on the ambient temperature. At low ambient temperature the amount of energy absorbed is low.

T, from solar field: It is the temperature of the HTF that flows from the collector field to the steam Rankine cycle (^{0}C). This depends on the solar irradiation at a particular time. At high solar radiation the temperature of the HTF is high and vice versa.

Q, absorbed: This is the net amount of energy absorbed by the HTF (MWh). It depends on the solar availability, direct normal radiation, collector cleanliness, collector field and HCE properties and the shading factor.

Q, heat loss: They include energy lost through piping, energy used in pumping the HTF, and energy lost through emissions in the tubes carrying HTF (MWh).

Efficiency of the collector field: This is the conversion ratio of the incident energy on the collector to the total amount of energy absorbed.

Wind speed: This is velocity of the wind. High velocity of wind lowers the amount of DNI collected on the surface of a collector.

The main parameters used in the design of the parabolic trough solar field are described in the following section. The parameters were obtained from past literature.

3.9.1 Definition of the input values of solar parabolic collector field

The inputs made on the SAM user interface under the solar field include:

Aperture width of collectors (m): this is the width of a collector

Aperture length of collectors (m): This is the length of a collector

Mirror cleanliness factor: This factor accounts for dust on the mirrors that reduce the ability of a mirror to reflect all the solar irradiance falling on it. Mirrors are cleaned after one month but due to dust accumulation especially in the deserts where wind speeds are high it is accounted for.

Table 3-4 shows the different parameters used in the design of the solar field.

Table 3-4 Parabolic Trough reference plant configuration used as base case for all Simulation [16, 26, 38, 78, 79, 80, 81]

Collector Type	LS-3+
Longitude	35.6
DNI	5.03kWh/m^2/day
Collector direction	Axis in North South direction
Solar multiple	2
Glazing	Double glass
Number of loops	97
Reflective aperture area	545m^2
Aperture length of solar collectors	100m
Aperture width(m)	5.75m
Agent fluid (HTF)	VP-1Therminol oil
Effective product transmittance	0.84
Emissivity of absorber E_b	0.92
Tubes distance	0.15m
Incident solar energy per unit area	952w/m^2
Heat Efficiency	0.85
Collector tilt	0^0
Absorptance	0.99
Intercept efficiency k_0	0.762
Thermal conductivity of absorber plate κ_a	384w/m K
Thermal conductivity of receiver	0.05 w/m^0K
Mirror Cleanliness factor	0.95
Emissivity of cover E_c	0.88
Average focus path length	2.11m
Piping distance between assemblies	1m
Envelope transmittance	0.97
Hot piping thermal inertia	0.2kWht/K-MWh
Cold piping thermal inertia	0.2kWht/K-MWh
Solar Field Availability	0.99
Field loop piping	4.5Wht/K-m
Absorber material	Copper
Number of modules per assembly	16
Row Spacing of collectors	15m
Mode of tracking	E-W horizontal

Solar field availability: This factor accounts for maintenance of the plant. This factor is used for calculation of the total energy absorbed.

Reflective aperture area (m^2): This is the area occupied by one loop of collectors.

Effective product transmittance: This is the heat loss coefficient between the HTF and the ambient air

Envelope transmittance: This is the heat loss coefficient between the absorber tubes and the ambient air

Thermal conductivity of absorber plate: This is the energy loss through conduction in the absorber tube.

Thermal conductivity of receiver: Amount of energy lost through conduction of the receiver tubes.

Average focus path length: This is the distance between the absorber tubes and the collectors.

Solar multiple: Solar multiple is the ratio of the of the energy supply by the solar field to the turbine design heat input as described in section 4.8, Equation 4-30. This allows the solar field to oversize or undersize itself depending on the amount of DNI at a particular hour of the day to match the load on the turbine. In this simulation SM of 2 has been used for sizing the field.

Design solar irradiance: This is the amount of solar radiation falling on a unit area per day. It is described in section 3.7.3.

Cold piping thermal inertia: This is energy loss through inertia as thermal energy is carried in the pipe.

Emissivity of absorber: A factor that accounts for emission losses in the absorber tube

Figure 3-14 shows the SAM user-interface window for the solar field inputs.

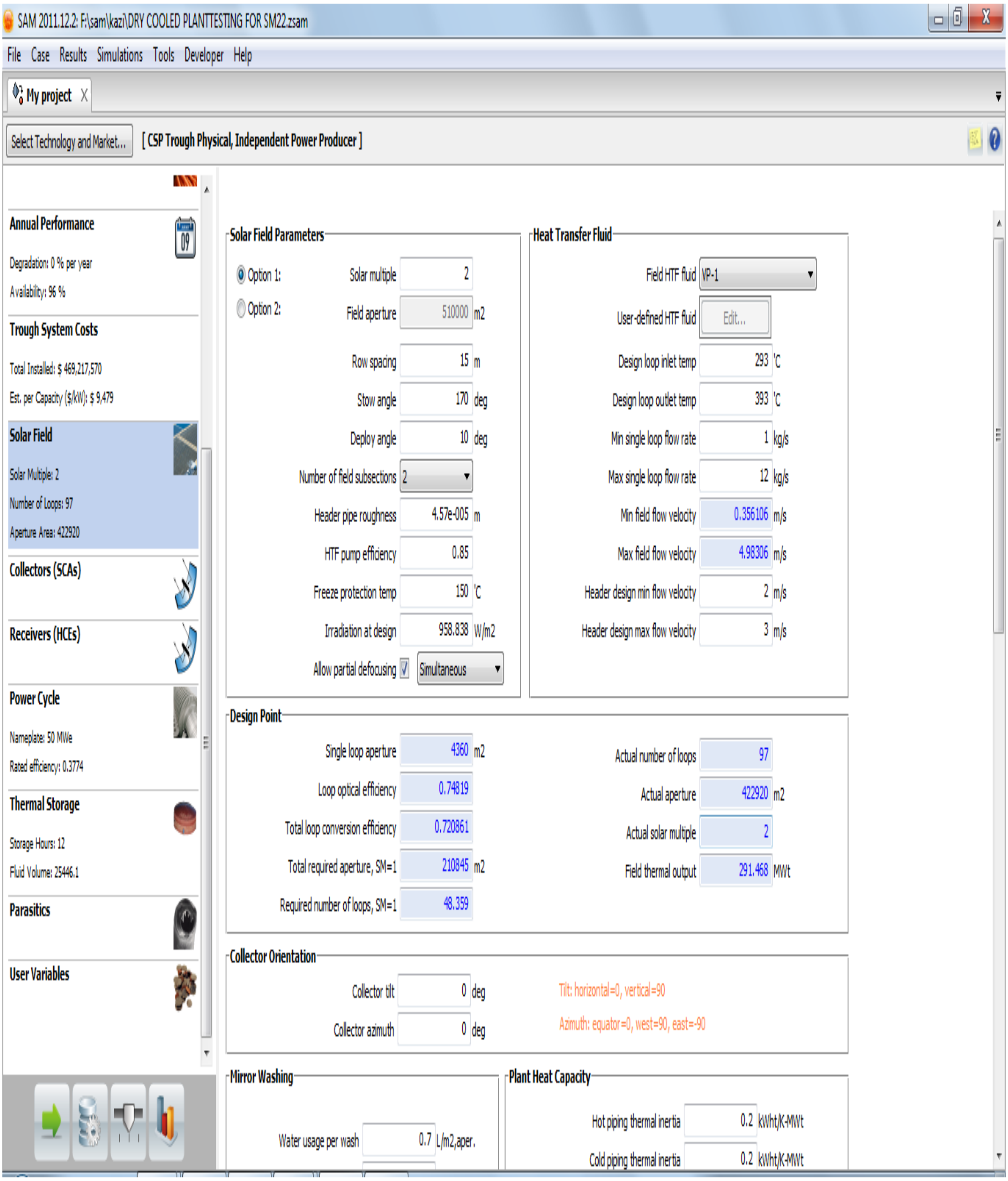

Figure 3-14 SAM Collector user-interface window

The collectors reflect the heat energy of the sun on the receivers as described in the following section.

3.9.2 Receiver Model (Heat collecting Element, HCE)

Heat Collecting Element (HCE) is the main component of a parabolic trough CSTP. It acts like the heart of the trough as it determines how much heat is absorbed and also largely determines the efficiency of the parabolic trough. The bellows shown in Figure 3-15 are fitted to ensure creation of maximum vacuum in the air-tight enclosure and also to accommodate for thermal expansion difference between the metal pipe and the glass envelope.

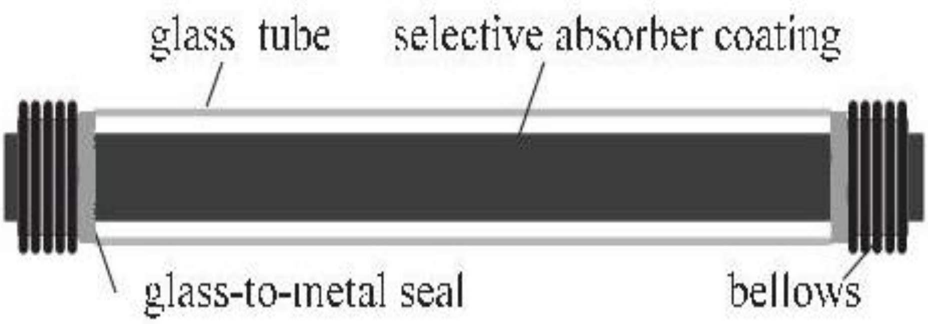

Figure 3-15 Schematic of a Typical Solar Receiver [83]

The receiver is modelled as a one dimensional energy flow. In Figure 3-16, a quarter of the cross-section of the receiver has been shown and the respective heat balance. In this case the concentrated irradiative flux from the collector passes through the transparent glass envelope and reaches the absorber tube at point R2.

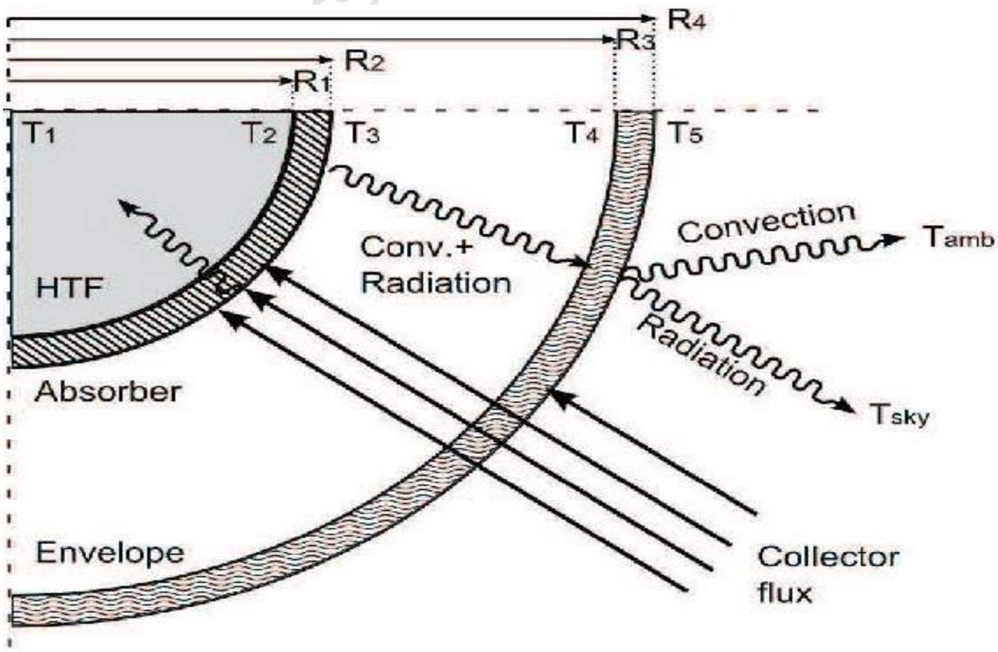

Figure 3-16 Cross Section of a Receiver [72]

During generation the heated HTF is used to drive the thermal energy through the absorber wall from R1 to R2 and finally into the cooler HTF. Thermal losses occur through radiation exchange with the glass envelope and convection.

SAM calculates the receiver temperature by Equation 3-3 below [72].

$$T_r = \frac{A_a I_a(t)\eta_{opt} + m_w c_w T_i(t) F_r + A_r U_l T_a(t)}{m_w c_w F_r + A_r U_l} \quad \text{....... 3-3}$$

where

A_a : Area of the solar field (m^2)

I_a : DNI (kWh/ m^2)

η_{opt} : Optical efficiency of the mirrors

m_w : Mass flow rate of HTF (kg/s)

c_w : Specific heat capacity of the HTF (kJ/kg ^0K)

T_i (t): Temperature of the HTF entering collector (^0K)

A_r : Area of the receiver section (m^2)

U_l : Overall heat coefficient.

F_r : Heat removal factor

The parabolic trough CSTP receiver model used in this thesis is Solel UVAC. It was used together with the LS-3 parabolic trough collectors in the Mojave Desert California in a parabolic trough plant that had a capacity of 30MWe [22, 74]. It has the parameters listed in Table 3-5. It was chosen for this simulation because it has a better precision and two axis of tracking [83].

The following section defines the receiver input parameters made on the SAM user interface.

3.9.2.1 Definition of the input values of solar parabolic Receivers shown in Table 3-5

Optical efficiency: It accounts for the amount of solar radiation absorbed by the receivers.

Mass flow rate (kg/hr): This is the amount of HTF that gets to each collector per hour.

Receiver emission: Accounts for the energy loss through emission in the receiver to the ambient air.

HTF Inlet temperature (^{0}C): Temperature of HTF entering the collector field.

HTF Inlet temperature (^{0}C): Temperature of HTF leaving the collector field to the steam Rankine cycle.

Cover reflective index: An index on the receiver that accounts on the amount of solar radiation reflected away.

Cover emittance: Accounts for energy loss through emission on the cover of the absorbers.

Specular absorptance: Accounts for the amount of energy absorbed by the receivers.

Receiver thermal losses (W/m^2-K): Amount of energy lost by the receiver.

Superheated cover thickness: This is the thickness of the absorber tubes carrying the hot HTF from the field to the steam Rankine cycle.

Table 3-5 Receiver Reference parabolic configurations used as base case for all Simulations [26, 79, 84, 85, 86]

Receiver type	Solel UVAC 3
Receiver outer diameter $d_o(m)$	0.066m
Glass envelope outer Diameter	0.07
Receiver inner diameter $d_i(m)$	0.077m
Absorber length $l(m)$	12.5m
Thermal conductivity	47.6W/m.^{0}C
Mass flow rate	38kg/hr
Receiver emittance μ_r	0.91
Superheated cover thickness	0.0025m
HTF Inlet/outlet temperature	293^{0}C/393^{0}C
Bellows shadowing	0.971
Cover emittance μ_c	0.88
Cover thickness $L(m)$	0.0025m
Cover extinction coefficient $K(m^{-1})$	32
Specular absorptance	0.93
Optical efficiency	0.94
Cover reflective index η_2	1.526
Receiver thermal losses	0.852W/m^2-K
Heating fluid(water,VP1 oil)	1

3.9.3 Incident solar Energy

The amount of solar energy that falls on the surface of the collector is called the incident energy. SAM estimates the amount of incident energy falling on the collector surface using Equation 3-4 below [78].

$$Q_{\text{incident}} = DNI * \cos\theta * I_{am} * Row\,shadow * End\,loss * \eta_{field} * \eta_{HCE} * SFA_{avai} .... 3\text{-}4$$

where

DNI =This is the amount of solar irradiation received on each square metre of the collectors

$\cos\theta$ = Angle of incidence: This is the angle between the solar radiation on a surface and the plane normal to the surface. The angle of incidence varies as the sun cruises the sky.

I_{am} = Incidence angle modifier: The incidence angle modifier accounts for the losses occurring on the surface of the mirrors and also the absorption losses.

$Row\,shadow$ = This is a performance factor that accounts for the shading of collectors in the morning and evening.

The position of the sun introduces losses to the collectors due to shading. In the morning all the collectors are tilted to the East as the sun is rising. As was shown in Table 3-4 the distance between the rows of the collectors is 15m. The collectors in the east most rows receive higher radiation than the collectors in the west most end. The collectors in the east most ends introduces a shade in the nearest row which lowers its full ability of solar heat collection.

$End\,loss.$ = This is a performance factor that accounts for the losses in the receivers.

η_{field} =Efficiency of the collector field. It accounts for dust on the mirrors and cracks on mirrors.

η_{HCE} =It is the efficiency of the receivers that accounts for the imperfections that may happen to the receivers. The amount of solar energy absorbed by the receivers varies. This variation is brought about the inaccuracies of the collector field mirrors, and the receiver tube materials.

SFA_{avai} =This is the fraction of the collectors that are in operational at any given time.

At high solar irradiation some collectors are defocused to minimize losses as discussed in the following section.

3.9.4 Collector Defocusing

There exists a possibility where the solar field might deliver more than the rated thermal energy to the power cycle. At high temperatures more steam is produced which results in an increase in the steam mass flow rate in the turbine which exceeds the turbine gross thermal energy design input. The excess thermal energy is lost as waste. To avoid thermal energy wastage some collectors are defocused to maintain the turbine gross design input.

Defocusing of the collectors is done by tilting the collectors. This lowers the concentration of the DNI on the absorber tubes thus consequently lowering the temperature of the HTF. In this parabolic CSTP model, sequential defocusing is implemented. The collectors are defocused sequentially starting with the collectors with the highest temperature. Sequential defocusing is a type of defocusing scheme where the collectors with the highest temperature are tilted off first followed with the collectors with less temperature.

Sequential defocusing has more control over the magnitude of the defocusing. This reduces the probability of over defocusing and losing thermal energy which in turn increases the LCOE and reduces the efficiency of the plant.

This usually implies a steady increase in the solar field HTF temperature results in the increase in the mass flow rate of the HTF. Equation 3-5 describes the defocusing scheme [78].

$$M_{HTF,\max} = V_{HTF,\max} X \rho_{HTF} X \left(\frac{D_{\min}}{2}\right)^2 \quad \textbf{....... 3-5}$$

where

$M_{HTF,\max}$ = Maximum flow rate of HTF (kg/s)

$V_{HTF,Max}$ = Maximum HTF velocity (m/s)

$\rho_{HTF.}$ = HTF density kg/m^3)

$D_{\min}$ = Minimum diameter of receiver tube (m)

The power cycle steam Rankine cycle is described in the following section.

3.10 Power cycle and the Steam Rankine cycle

Power cycle is the movement of thermal energy from collection to electricity production. The power cycle used in parabolic trough is a steam Rankine cycle. In a steam Rankine cycle water is used for steam generation. Figure 3-17 shows the steam Rankine cycle processes.

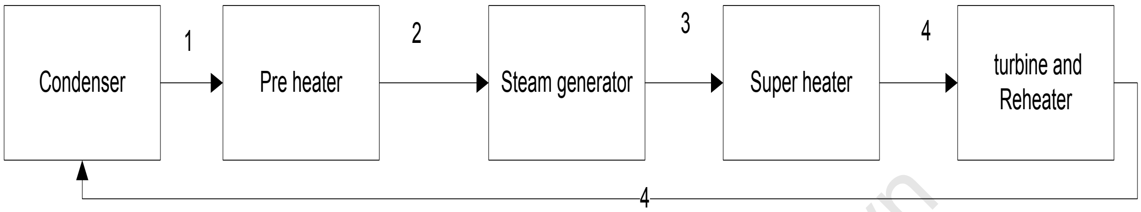

Figure 3-17 Steam Rankine Vapour Processes [114]

The steam Rankine cycle converts heat into work. The steam Rankine consists of four main processes: 1-2, 2-3, 3-4, 4-1.

Process 1-2 (Pre-heater): The temperature of the HTF from the field is at 317.17^{0}C. The hot HTF flowing from the field enters the pre-heater where it comes into contact with water flowing in pipes from the dearator. The dearator water temperature is raised to about 234.8^{0}C at a pressure of 103.56 bars.

Process 2-3 (steam generator): The hot HTF is heated to a temperature of 377.22^{0}C in the evaporator. The HTF comes into contact with an economizer, evaporators, super heater and a re-heater. The economizer is used as a pre heater for raising the temperature of the incoming high pressure feed in water to saturation.

Process 3 -4: The temperature of the HTF is raised to 393^{0}C in the super heater which evaporates water at a temperature of 377^{0}C. The steam is converted into mechanical energy by the turbine. The turbine is coupled to a generator which produces electricity as a result.

Process 4-1: The exhaust steam from the high pressure turbine is recycled at the re-heater to a temperature of 377^{0}C at a pressure of 100 bars to run the low pressure turbine.

Figure 3-18 and Figure 3-19 the pressure volume and heat entropy diagrams for an ideal Rankine cycle diagram respectively.

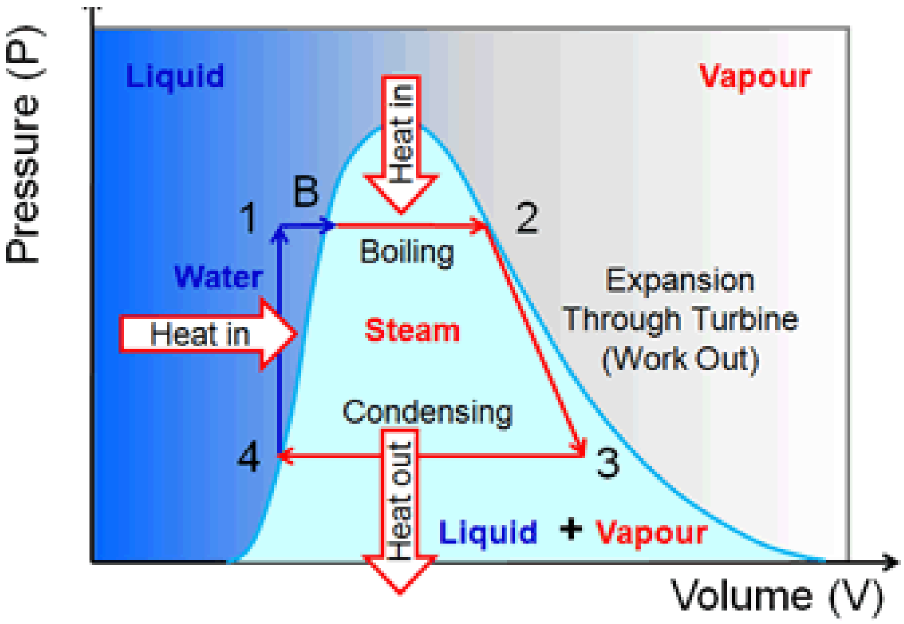

Figure 3-18 Ideal Rankine Pressure volume Diagram [116]

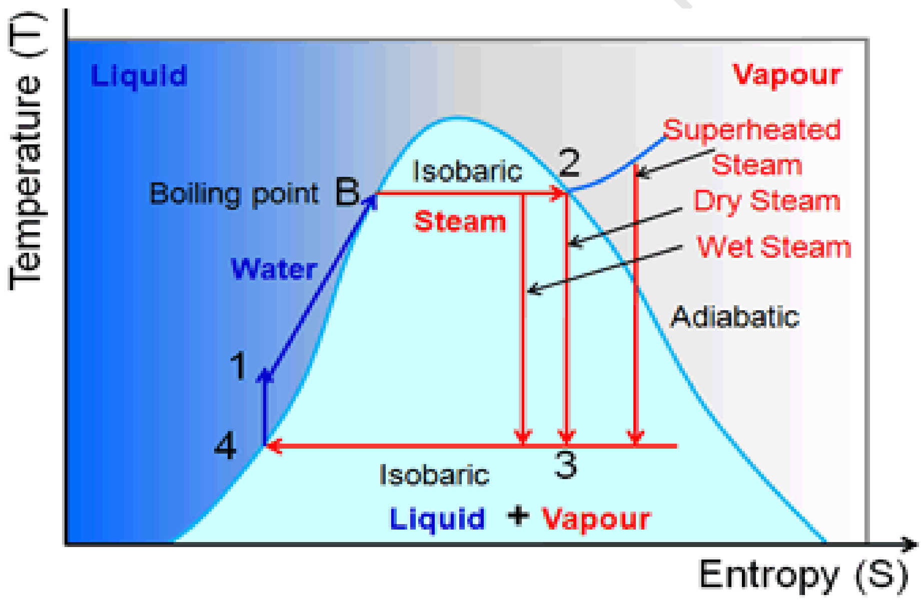

Figure 3-19 Ideal Rankine Heat Entropy Diagram [116]

Figure 3-20 depicts the heat exchange between water and HTF in the Rankine cycle.

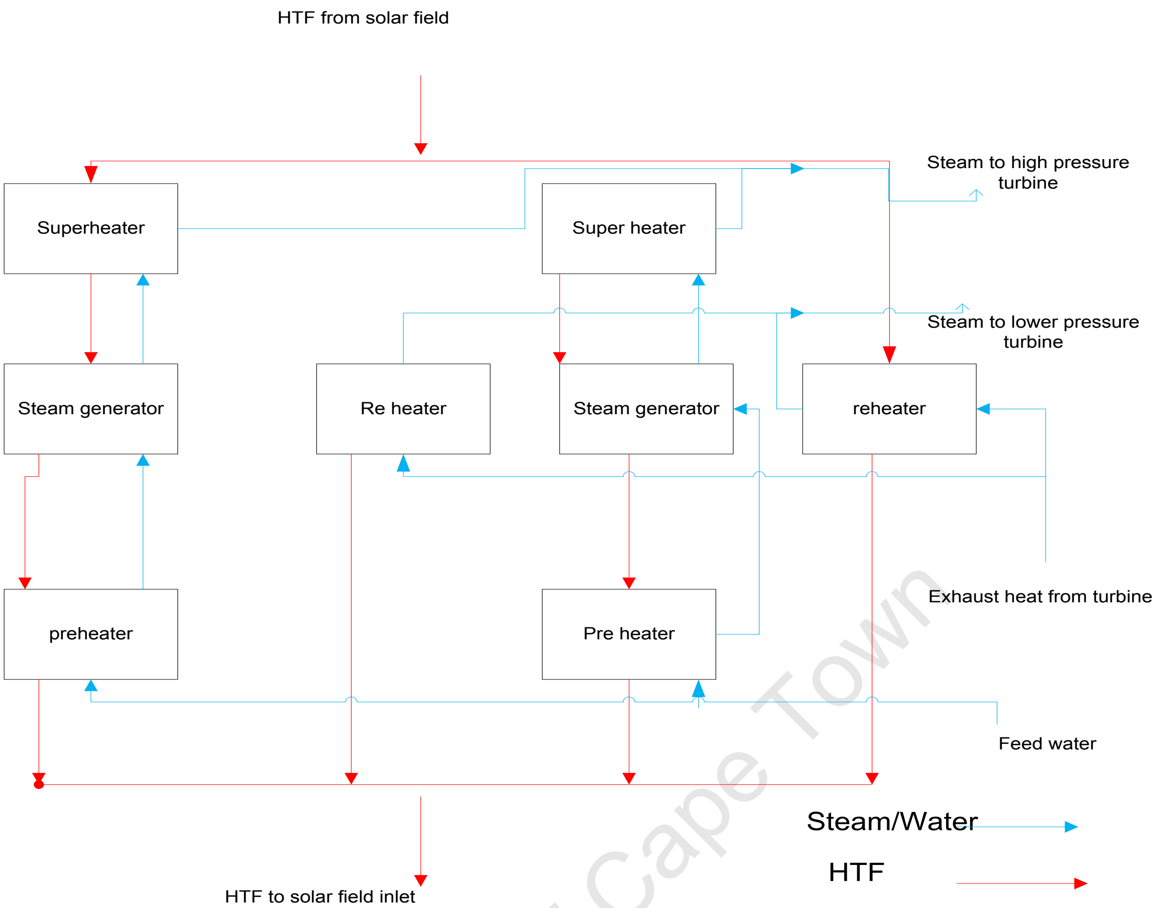

Figure 3-20 Flow diagram of HTF and water Interaction in the Steam Rankine cycle [114]

The temperature of the heat exchangers in the Rankine cycle is shown in Table 3-6 and Table 3-7.

Table 3-6 Steam Temperature in the Steam Rankine cycle [114]

Heat exchanger	Temp of Steam Input(^{0}C)	Pressure of steam (Bars)	Temp of steam output(^{0}C)	Pressure of steam out (bars)
Pre heater	234	103	----	103
Steam generation	------	103	----	103
super heater	------	103	377	100
Re heater	------	18	377	17.09

Table 3-7 HTF Input and output Temperature in the Steam Rankine cycle. [114]

Heat exchanger	Temp of HTF Input (^{0}C)	Temp of HTF out ^{0}C
Pre heater	317	293
Steam generation	377	317
Super heater	393	293
Re heater	393	293

The assumptions made in the modelling of the Rankine cycle are:

- It is assumed that all the steam generated by the turbine is used for energy generation by the turbine. In this case the potential and the kinetic energy developed as the heat exchange occurs are assumed negligible.
- The losses that occur as the HTF and water moves from one heat exchanger to the other are assumed to be negligible [114].

The difference between the steam heat input and the steam heat rejected is the work done by the turbine as shown in Equation 3-6.

$$Work\ done\ by\ turbine = Heat\ input - Heat\ rejected \quad \textbf{....... 3-6}$$

The efficiency of the Steam Rankine cycle is described using Equation 3-7 below.

$$efficiency\ \ of\ Rankine\ cycle = \frac{work\ done\ by\ turbine}{heat\ input} \quad \textbf{....... 3-7}$$

Where the work done by turbine is the output mechanical energy from the turbine (MWh) and the heat input is the total heat energy input to the turbine by the hot steam (MWh). The higher the work done the higher is the steam Rankine cycle efficiency. Lower steam Rankine cycle efficiency is caused by inefficient cooling methods which lead to energy wastage in the solar field collection. For example if the cooling method applied is unable to reject a certain quantity of heat in a specified period of time the HTF saturates and absorbs no more heat energy. This leads to energy dumping which reduces steam Rankine cycle efficiency.

The performance of the steam Rankine cycle is based on steam balances as shown in Equation 3-8 [16]:

$$\eta_R = \frac{E_p}{\mathrm{m}_s x (h_g - h_e)} \quad \textbf{....... 3-8}$$

where

E_p =Electrical power produced (MWh)

m_s =Steam flow in the heat recovery boiler (kg/s): When the mass flow rate of steam to the turbine is higher the effective heat input is higher and hence the efficiency of the steam Rankine cycle. The net electrical energy increases.

h_g = Enthalpy of the steam at the heat recovery boiler outlet (kJ/kg)

h_e = Enthalpy of the steam at the heat recovery boiler inlet (kJ/kg).

η_R = steam Rankine cycle performance/efficiency of the Rankine cycle (%).

The design parameters of the steam Rankine cycle entered on the SAM window interface are shown in Table 3-8.

Table 3-8 Steam Rankine Cycle inputs [82, 87, 88, 90]

Item	Value	Unit
HTF inlet Temperature	393	^{0}C
HTF outlet Temperature	293	^{0}C
Steam temperature at turbine inlet	377	^{0}C
Boiler pressure	100	Bar
Turbine net power	50	MWe
Pump isentropic efficiency	0.695	%
Turbine gross power	55	MWe
Condenser pressure	0.085	Bar
Steam extraction pressure, high pressure	23.9	Bar
Steam extraction pressure, Low pressure	2.9	Bar

The following section discusses the energy balance in the power cycle.

3.10.1 Energy balance in the power cycle

As was previously described in section 3.9.2, the amount of energy absorbed by the receivers is subject to losses. These losses include emission losses between the absorber tube and the ambient air, shadowing losses and reflection losses.

Figure 3-21 describes an over view of the energy transfer process within the solar parabolic trough plant. In this figure I_{coll} is the DNI (W/m^2) received by the collector, A_{coll} (m^2) is the area of the collector; $I_{coll}.A_{coll}$ is the incident thermal energy received from the sun by the collectors (MWh)

Q_{abs}(MWh) is the absorbed solar thermal energy; Q_{loss}(MWh) is the absorber heat loss energy; Q_u (MWh) is the useful thermal energy input to the steam Rankine cycle ; P_{el} (MW) is the output electric power produced .

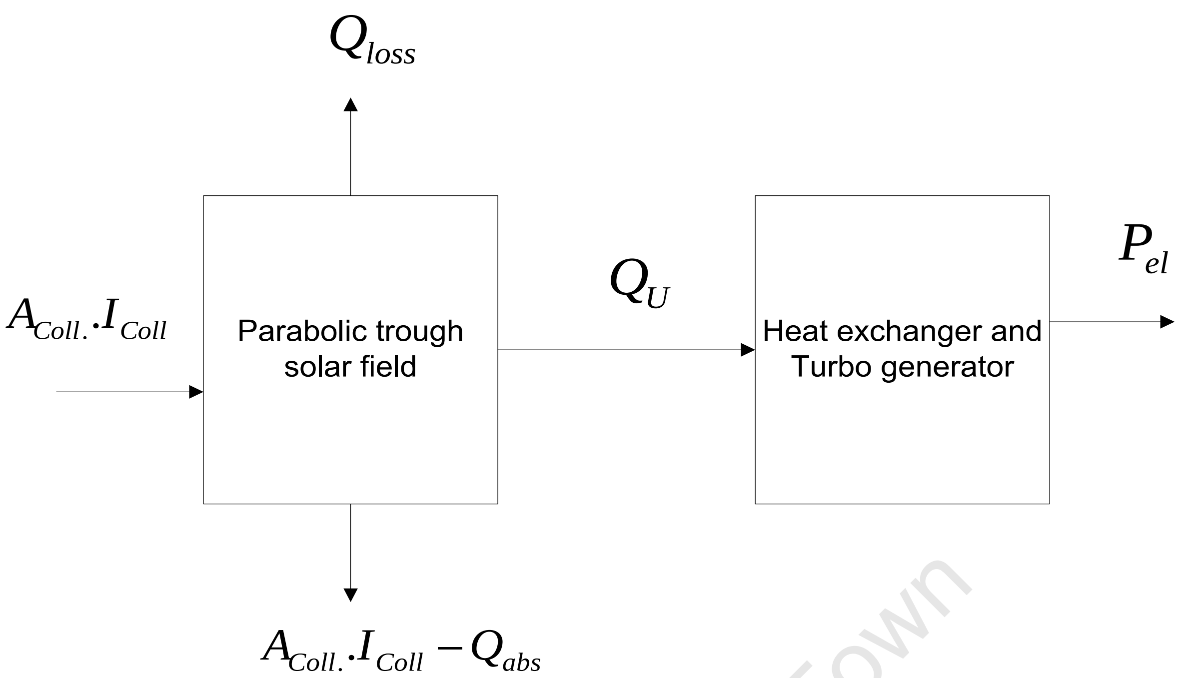

Figure 3-21 Simplified Process Diagram Describing Energy Balance Relationship within the CSTP System [91]

In the first block the rows of collectors are aligned for DNI collection. The collectors reflect the concentrated solar energy on the HTF contained in the line absorber tubes where this concentrated energy is captured and absorbed. Energy losses in the parabolic trough occur through pumping of HTF to and from the solar field, pumping of thermal energy storage (TES) from the cold tank to the hot tank and energy lost during cooling.

Thermal losses at the collector must be taken into account which includes solar thermal energy not captured by the absorber after reflection $(I_{coll}.A_{coll} - Q_{abs})$ and other heat loss that occur through natural convection with the surrounding air around the absorber tube containing solar field HTF. These energy losses are illustrated in Figure 3-22.

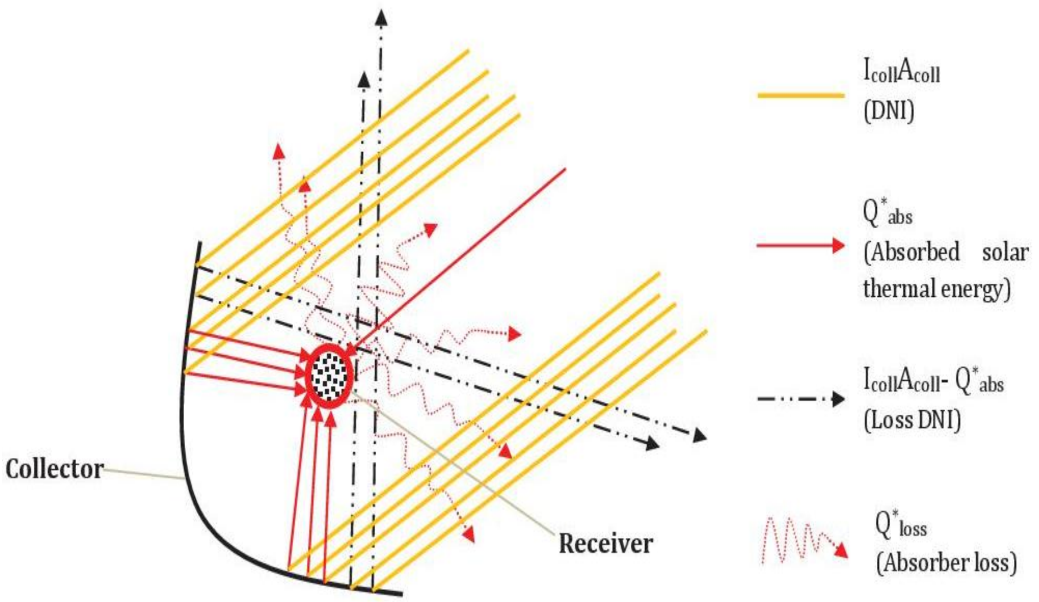

Figure 3-22 Sketch illustrating how losses occur during Solar Thermal Heat Collection [91]

Due to the losses incurred the resultant useful thermal energy which goes to the heat exchanger is given by Equation 3-9.

$$Q_u = Q^*_{abs} - Q_{loss} \text{....... } \mathbf{3\text{-}9}$$

where

Q_u = Field output thermal energy: This is the amount of energy that leaves the absorbers to the power block (MWh)

Q^*_{abs}= This is the amount of energy absorbed by the receivers (MWh)

Q_{loss} = Energy losses in the absorber (MWh)

The following section describes the power block model.

3.11 Power block

The power block converts the thermal energy collected in the fields to electrical energy. It is made up of equipments such as turbine, synchronous generator, and heat exchangers for transferring the thermal energy collected in the solar fields to the turbine or from the thermal storage to the turbine and the condensers for cooling the exhaust steam from the turbine. SAM uses the rated output energy to calculate the plant's capacity factor as will be shown in the results summary.

In order to determine the electrical energy produced the total mechanical power is determined by Equation 3-10 below [91].

$$W = m_v = W_w + W_{ev} + W_v \textbf{ .... 3-10}$$

where W is the total mechanical power (kWh) , W_w is the mechanical unit per unit mass in the water pre heating region (J/kg), W_{ev} is the mechanical mass per unit mass in the evaporation region (J/kg) and W_v is the mechanical work per unit mass in the steam superheating region (J/kg).

The total electric power ($p_{el,\max}$) produced is as shown in Equation 3-11 below [91].

$$P_{el,\max} = \eta_{el} W \textbf{ .... 3-11}$$

where η_{el} and W are electric conversion efficiency and total mechanical energy respectively. The following section discusses the power block parameters.

3.11.1 Power block parameters

The design of the power block is described by the following parameters:

- Rated cycle conversion efficiency: This is the thermal to electrical conversion efficiency of the power cycle at design conditions.
- Design inlet temperature: Temperature of HTF from the collectors to the heat exchangers at design.
- Design outlet temperature: Temperature of the HTF leaving the heat exchangers to the collectors.
- Steam cycle blow down fraction: This is the mass flow rate of steam that is extracted during cooling and replenished.

For the plant control, the following is also considered:

- Fraction of thermal power needed for standby: This is the amount of heat power in TES to keep the plant in a standby mode.
- Power block start up time: This is the amount of time the power block takes to start producing energy.
- Fraction of thermal power needed for start-up: This is the amount of power needed to start up the turbine at its designed thermal input. It is a function of the design turbine thermal input $Q_{pb,\,design}$ and the turbine start up energy fraction $F_{startup}$. This is shown in Equation 3-12.

$$Q_{startuprequired} = F_{startup}\ x\ Q_{pbdesign} \textbf{ .... 3-12}$$

- Minimum required Temperature: Temperature at which the flow of HTF starts through the heat exchangers [98].

These parameters are as shown in Table 3-9 below:

Table 3-9 Design Assumptions of Power Block [67, 98]

Design Inlet Temperature of HTF	393^0C
Design Outlet Temperature of HTF	293^0C
Boiler operating pressure	100 bar
Efficiency of PCU at load	0.5
Minimum required Start up Temperature	300^0C
Steam Cycle blow down Fraction	0.02
Standby period	2hrs
Fraction of thermal power for standby	0.2
Power block start up time	0.5hrs
Gross cycle efficiency	37.4%
Design Gross power output	55MWe
Net Power output	50MWe

In the following section electric parasitic losses that occur in parabolic trough CSTP plant are discussed.

3.11.2 Electric Parasitic Losses

The electric parasitic losses refer to the amount of energy consumed by the plant itself to generate electricity. These losses include

Drive motor losses: This is the energy drawn by the motors for tilting the collectors to track the sun.

Electronic circuits: The electronic circuits and computers used for system control also draw energy from the plant.

Pump motors: The pumps used for pumping HTF, water and thermal energy stored draw energy from the plant.

TES pumps: Electrical energy loss consumed by the pumps used for pumping the HTF.

Required power for HTF pumping through the power cycle: Amount of energy required to drive the HTF in the power cycle.

Fraction of rated gross power consumed at all times: This is the amount of energy that is lost at all times, 24 hours per day for all the 8760 hours in a year.

In order to achieve more accurate and realistic results parasitic losses must be taken into account in the simulation as shown in Table 3-10.

Table 3-10 Parasitic Losses considered During Modelling [67, 98]

Piping thermal loss coefficient	0.45	W/m^2-K
Tracking power	125	W/SCA
Required power for HTF pumping through the power cycle	0.15	kJ/kg
Fraction of rated gross power consumed at all times	0.0055	-
Hot piping thermal inertia	0.2	Wht/K-MWh
Cold piping thermal inertia	0.2	KWht/K-MWh
TES pumps	0.02273	MWe/MW capacity
Field loop piping thermal inertia	4.5	Wht/K-m

The net power output of the solar parabolic plant is equal to the difference between gross energy output and the total parasitic losses as shown in Equations 3-13 and 3-14 [82].

$$\text{Net power output} = \text{Gross power output} - \text{total parasitic losses} \quad \text{....... 3-13}$$

$$\%\ \text{Parasitic loss} = \frac{\text{Gross power} - \text{net power}}{\text{Gross power}} x100 \quad \text{....... 3-14}$$

The following section describes the thermal energy storage considered in this simulation.

3.12 Model of the Thermal Energy Storage

The thermal Energy storage (TES) is used for storing thermal energy. The stored heat in the storage can be used for running the turbine in hours of low or no solar radiation. TES is very useful especially in meeting the energy demand in places where the peak energy demand occurs at night. The other advantage of TES is that it can be used for generation shifting in hours when energy demand, i.e. the owner of the plant may choose to generate energy in the morning and dispatch it at night or peak tariff hours of the day. Thermal energy storage is divided into two i.e. direct storage and indirect storage. In direct storage the fluid used as the HTF is the same fluid used for storing energy. In the indirect storage the storage fluid is different from the HTF fluid. Indirect storage uses two tanks unlike the direct storage. The two tanks are the hot tank and the cold tank. The cold tank stores the hot thermal storage fluid while the cold tank stores the cold thermal storage fluid.

The following section describes the modelled storage using SAM.

3.12.1 Modelled Thermal Energy Storage

The modelled storage type is a two tank indirect storage. An indirect storage is a storage system where the HTF used is different from the storage fluid. Indirect storage consists of a cold tank, hot tank and a heat exchanger. The hot molten solar salt is stored in the hot tank and the cold molten solar salt is stored in the cold tank as shown in the circled portion of Figure 3-23. The heat exchange between the HTF and TES stored happens in the heat exchanger 1. In this thesis TES is used to extend the period of the power plant operation. The size of TES used is 12 hours with a maximum thermal capacity of 1748.81MWh. It is assumed that this thermal capacity is able to run the power block at its design capacity of 55MWe for 12 hours. It is further assumed that the TES system has the same power capacity during charging and discharging (*Charging and discharging of TES is discussed on section 3.14*). The losses during charging and discharging of TES are assumed to be 0.031% per hour.

The type of TES fluid used is a solar salt. It is made up of 60% sodium nitrate and 40% potassium nitrate. Solar salt was preferred for storage because of the following reasons:

- Cheaper than VP-1 oil ($0.49 per kg and $5.8 per kg storage cost)
- Higher operation temperatures which increases the storage capacity and reduces the cost of storage. The cold storage tank operates at a nominal temperature of 293 ^{0}C. The hot storage tank operates at a nominal temperature of 385^{0}C.
- High density (1899 @300^{0}C), viscosity of 3.26 @300^{0}C and a heat capacity of 1495@300^{0}C [67, 94].

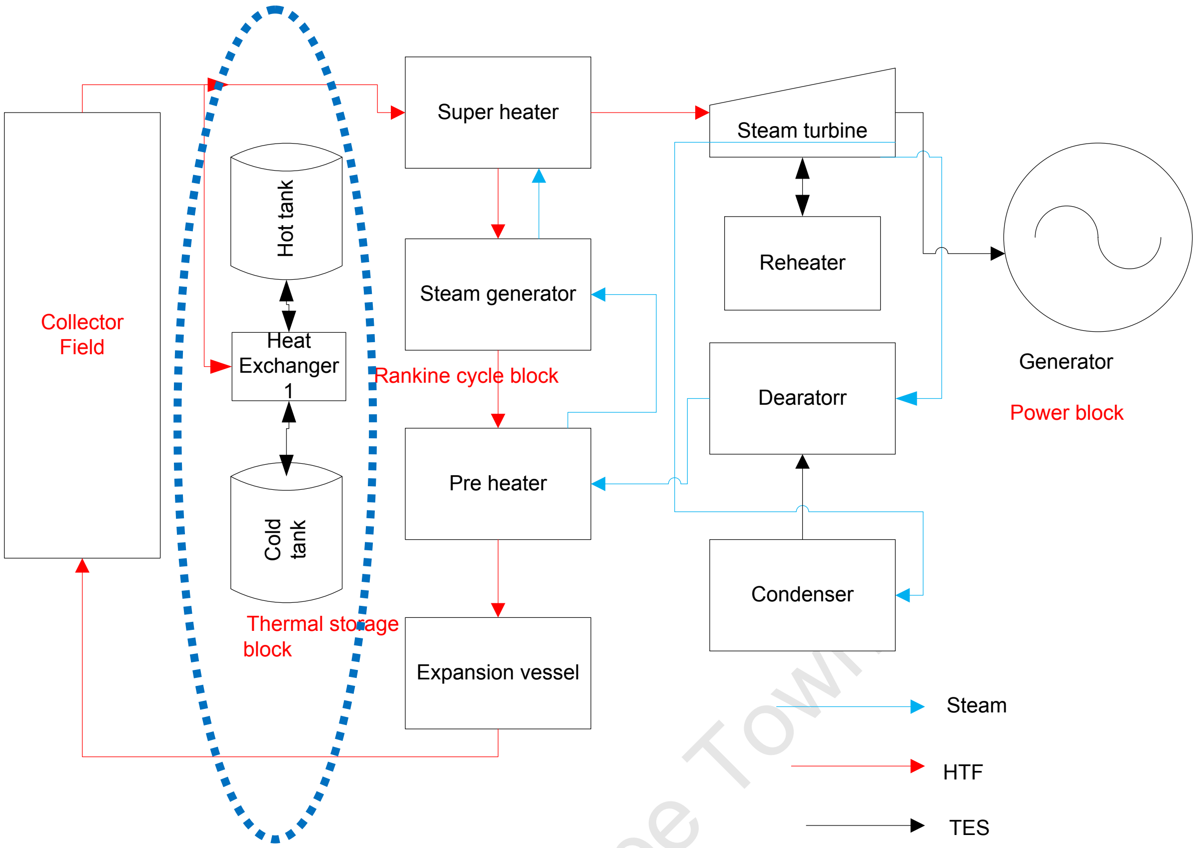

Figure 3-23 Schematic flow Diagram of Parabolic Trough Showing TES Block [38]

The amount of energy stored in the hot tank depends on the temperature of the solar field, HTF mass flow rate and the ambient air temperature as shown in Figure 3-24

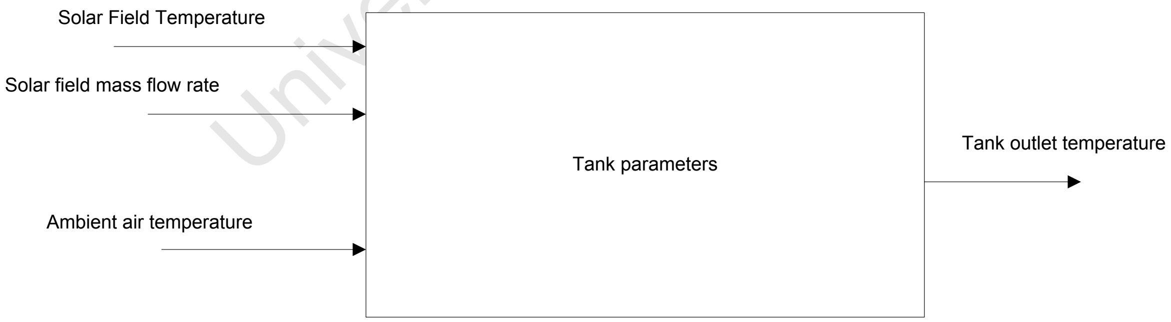

Figure 3-24 Heat balance in Thermal energy storage

The heat balance in the storage tank is governed by Equation 3-15 [114].

$$\mathrm{M}_{,\mathrm{tank}} = C_{\tan k}\frac{dT}{dt} = m_{htf}c_{htf}(T_{in} - T_{out} + UA_{loss}(T_{env} - T_{in})) \text{ .... } \mathbf{3\text{-}15}$$

where

$\mathrm{M}_{,\mathrm{tank}}$ =the mass of the molten salt in the hot tank (kg)

$C_{\tan k}$ =specific heat capacity of the molten salt (kJ/kg. ^{0}C)

$\frac{dT}{dt}$ =temperature change of the stored molten salt with time (^{0}C)

m_{htf} =mass flow rate of the HTF in the heat exchanger 1(kg/hr)

c_{htf} =specific heat capacity of HTF (kJ/kg ^{0}C)

T_{in} =Temperature of the molten salt tank entering the tank (^{0}C)

T_{out} =temperature of the HTF after exchanging heat with molten salt (^{0}C)

UA_{loss} =loss coefficient of the tank (kW/^{0}C)

T_{env} =temperature of the ambient air surrounding the tank (^{0}C)

The inputs parameters of TES into SAM user interface are shown in Table 3-11.

Table 3-11 A 12-hour TES System Design for 50-MWe [79]

Parameter	Cold tank	Hot tank
Number of tanks	1	1
Height , m	28	28
Tanks heat efficiency	-	0.98
Diameter, m	34.2	34.7
Floor, wall, and roof area,m^2	6671.8	6844.2
Inventory Temperature ^{0}C	250	385
Thermal losses (%)	-	0.031
Mean Insulation Temperature ^{0}C	159	207
Specific heat capacity of TES kJ/kg-K	1.50182	1.50182
Density of TES (kg/m^3)	1872.49	1872.49
Tank heat Loss , kWt	210	246

Heat transfer fluid is used for absorbing the collected solar energy in the collector field as described in the following section.

3.13 Heat Transfer Fluid

In CSTP plants a heat transfer fluid (HTF) is a liquid media used for absorbing the collected thermal energy in the collectors. The collected heat is used for steam generation from water. The generated steam is used for electricity production. In this simulation VP-1 oil is used as HTF. It is a eutectic mixture made up of 73.5% dyphinel oxide (DPO) and 26.5% biphynel. The choice of VP-1 HTF in this research work was suitable because of the following reasons:

- It has a high density (1060kg/m^3 @ 25^{0}C) [92]. A high density ensures a greater energy capacity per given volume.
- Low vapour pressure (33.1 bars): This ensures that the HTF does not evaporate at high temperatures.
- Moderate specific heat (206kJ/kg)
- Thermal stability: It has thermal maximum temperatures of 393^{0}C. It is reliable and trouble free even when operated at maximum temperatures
- At standard pressure its temperature is 298^{0}C.
- Low chemical reactivity: VP-1 does not react with the pipes materials.
- Low cost ($3.96 per kg) [92, 93].

The following section describes the interaction of HTF and the thermal energy stored.

3.14 HTF and its interaction with TES

The heat transfer fluid circulates through the absorber tube where it is heated by the concentrated heat reflected by the collectors on the absorber tubes. The amount of heat that goes to storage depends on the heat capacity of the HTF and the temperature gradient between the charged and discharged states.

In the charging state the HTF absorbs the heat reflected on the absorbers. The absorbed heat is passed through to the steam Rankine cycle where it heats some water to form steam.
The excess heat in the HTF is used to raise the temperature of the circulating molten salt in the heat exchanger 1 as was shown in section 3.12.1, Figure 3-23 above. The heated molten salt goes to the hot tank.

In the discharging mode the hot TES from the hot tank flows through heat exchanger 1. Its thermal energy is absorbed by the HTF. The discharged molten salt flows to the cold tank. The charged HTF is used to evaporate water for steam generation.

The fully discharged state of a storage system is defined by the minimum temperature acceptable at the turbine (250^{0}C). Equations 3-16 and 3-17 define charging and discharging of TES [82].

$$Q_{TES,Charging} = Q_{Stored} + Q_{loss} \text{ .... } \mathbf{3\text{-}16}$$

$$Q_{TES,Discharging} = Q_{Stored} - Q_{loss} \text{....... } \mathbf{3\text{-}17}$$

where

$Q_{TES,Charging}$ = Indicates the charging of TES

Q_{Stored} = the available heat energy in the tank HTF before charging commences

Q_{loss} = the energy lost during charging (MWh)

The quantity of heat released or stored per hour by the heat transfer fluid during charging or discharging is the time integral over the heat flow into or out of the storage system in that period of time as shown in Equation 3-18 [82].

$$Q_{storage,Charging} = \int_0^t Q_{Stored} = \int_0^t (Q_{Stored} - Q_{loss}) \text{....... } \mathbf{3\text{-}18}$$

The charging of the storage tank (Equation 3-16) only happens when the temperature of the HTF is higher than the stored energy in the tank. Charging of TES is not a continuous process because of the intermittent nature of the sun.

In the discharging mode, solar salt flows from the hot tank to the cold tank. The amount of energy dispatched depends on the load demand on the system. Equation 3-19 describes the discharging of the solar salt [82].

$$Q_{storage,Discharging} = \int_0^t Q_{Stored} = \int_0^t (Q_{stored} + Q_{loss}) \text{....... } \mathbf{3\text{-}19}$$

Discharging occurs during hours of low radiation or at night.

The storage efficiency is dependent on the ratio between discharging and charging. During hours of high ambient temperature, charging of TES occurs at a higher rate. However the molten solar salt has a limited heat capacity beyond which no more heat is absorbed and the excess heat goes to waste. This reduces the efficiency. Equation 3-20 shows the efficiency of energy storage in the tanks [82].

$$\eta_{storage} = \frac{Q_{discharging}}{Q_{charging}} = \frac{1 - \frac{Q_{loss}}{Q_{Stored}}}{1 + \frac{Q_{loss}}{Q_{Stored}}} \text{....... } \mathbf{3\text{-}20}$$

where

$\eta_{storage}$ = the efficiency of energy storage in the tanks.

3.15 TES Dispatch

In periods of low solar radiation,night time or high energy demand periods an appeal for a continous and stable source of energy rises. As was previously discussed in section 3.12.1, TES system consists of the cold tank,hot tank and heat exchangers.

SAM has a dispatch mechanism whereby the hot TES comes into contact with the HTF in the heat exchanger 1 and energy transfer occurs. The heated HTF flows into the steam Rankine cycle where its used to evaporate water for electricity production.TES dispatch is divided into two:

3.15.1 TES dispatch with solar

This is the fraction of TES dispatched from storage when the solar irradiation is available.

3.15.2 TES dispatch without solar

This is the fraction of TES dispatched from storage when the solar irradiation is low or unavailable.

3.15.3 Choice of dispatch

The choice of the dispatch schedule will depend on the external controlling factors such as wind speed, peak production limits, fluid levels in the tank, and also the minimum fluid level temperature in the tank. Dispatch of energy depends on the following:

- Power block operating mode in the previous hour (charging or discharging).
- Quantity of energy in storage in the current hour
- Energy available from the solar field in the current hour
- Time of day and storage dispatch fraction value assigned to the time of day

TES dispatch can be uniform or non-uniform. In uniform TES dispatch all the energy stored is equally distributed in all hours of the day regardless of whether the sun is there or not. In non-uniform TES dispatch, energy is dispatched from storage depending on the load in the current hour. In this simulation both uniform and non-uniform dispatch are considered.

3.16 Scenarios of TES Dispatch

This section describes how and when TES dispatch occurs in the parabolic trough.

TES Dispatch occurs in three ways:

3.16.1 When energy to TES does not exceed maximum charge rate

The amount of energy that goes to TES in this case does not exceed the charge rate of the hot TES tank as shown in Figure 3-25. At this stage the amount of energy at the power block

(Q_{PB}) (kWh) is equal to the load at the power block ($Q_{PB,Load}$) (kWh) as described in Equation 3-21.

$$Q_{PB} = Q_{PB\,load} \text{ .... } \mathbf{3\text{-}21}$$

The amount of energy that goes to TES ($Q_{to\,TES}$) (kWh) is the difference between the amount of energy generated from the solar field (Q_{SF}) (kWh) and the amount taken by power block (Q_{toPB}) (kWh) as shown in Equation 3-22[97].

$$Q_{To\,TES} = Q_{SF} - Q_{to\,PB} \text{ .... } \mathbf{3\text{-}22}$$

The energy to start the power block ($Q_{startup}$) (kWh) is equal to the energy dispatched from TES for start-up ($Q_{fromTES}$) (kWh) as shown in Equation 3-25.

$$Q_{fromTES} = Q_{Startup} \text{ .... } \mathbf{3\text{-}23}$$

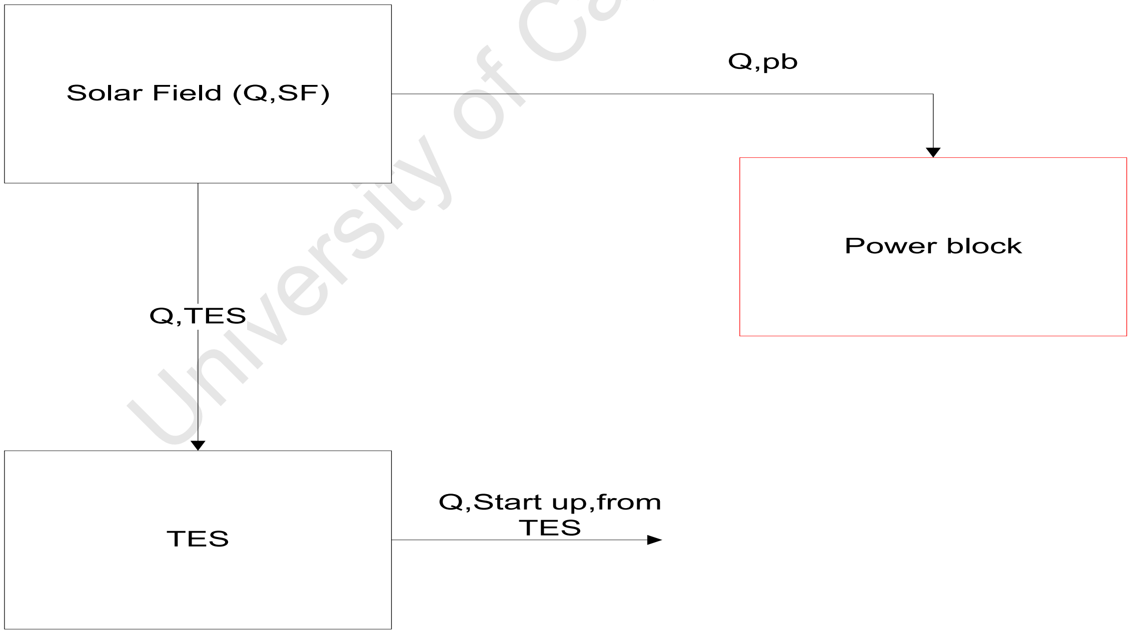

Figure 3-25 TES Movement when Maximum Charge rate is not exceeded [97]

3.16.2 Energy to TES exceeds maximum charge rate

The movement of TES when it exceeds the maximum charge rate of the hot tank is shown in Figure 3-26. In this case the energy to start the power block is equal to the demand of electricity as shown in Equation 3-24[97].

$$Q_{PB} = Q_{PB\ Load} \text{....... } \mathbf{3\text{-}24}$$

Also the energy in TES has reached its maximum charge rate ($Q_{TES,MAX}$) as shown in Equation 3-25[97].

$$Q_{TES} = Q_{TES\ MAX} \text{ .... } \mathbf{3\text{-}25}$$

The difference between the energy going to TES from the solar field and the maximum energy TES can hold is dumped (Q_{dump}) as shown in Equation 3-26 [97].

$$Q_{dump} = Q_{TES} - Q_{TES\ MAX} \text{ .... } \mathbf{3\text{-}26}$$

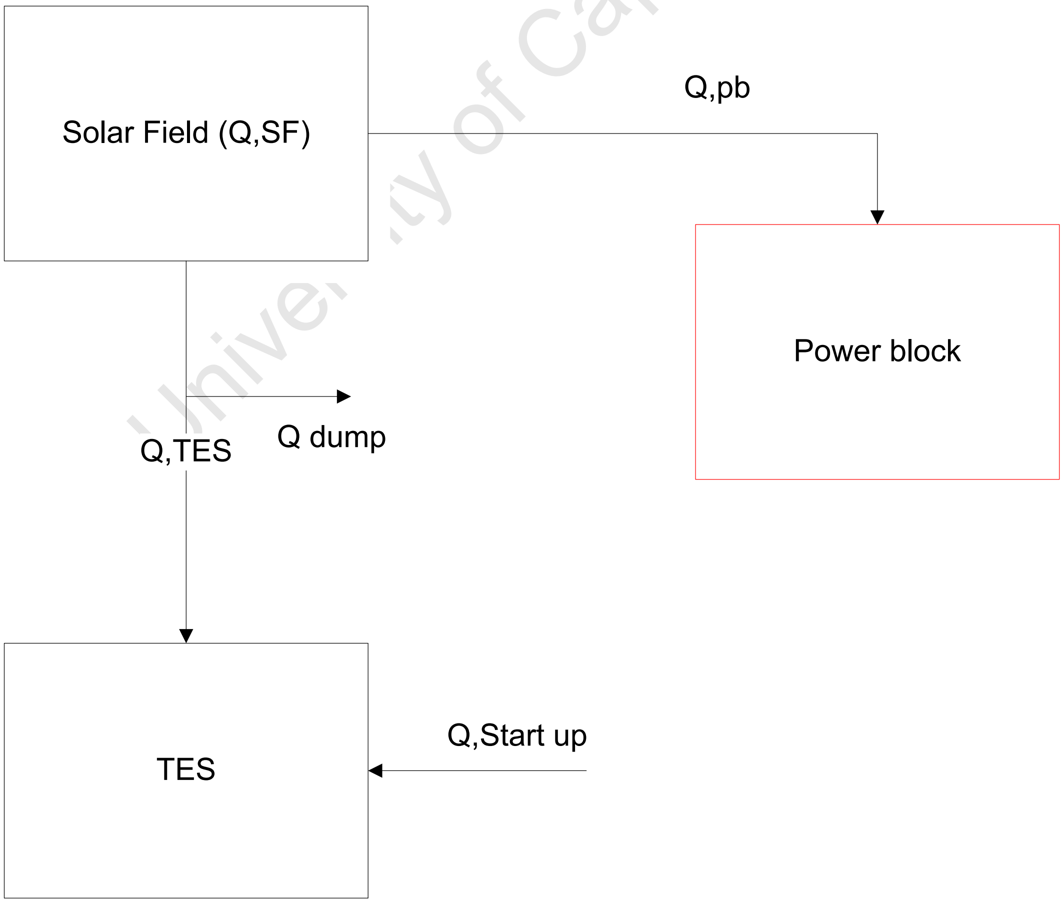

Figure 3-26 TES Movement when Maximum Charge rate is Exceeded [97]

3.16.3 When the Solar Field energy is less or equal to load requirement

When the solar field energy is equal or insufficient to supply the power block with enough energy, TES movement follows Figure 3-27. In hours of low DNI the solar field is not able to collect enough energy to run the power block or cannot meet the demand load at power block. In this case no energy goes to TES as shown in Equation 3-27 [97].

$$Q_{to,TES} = 0 \quad \textbf{....... 3-27}$$

Therefore energy is dispatched from TES to the power block as shown in Equation 3-28.

$$Q_{fromTES} = Q_{startup} + \left(1 - \frac{Q_{sf}}{Q_{to\ PB}}\right).Q_{fromTES\ MAX} \quad \textbf{....... 3-28}$$

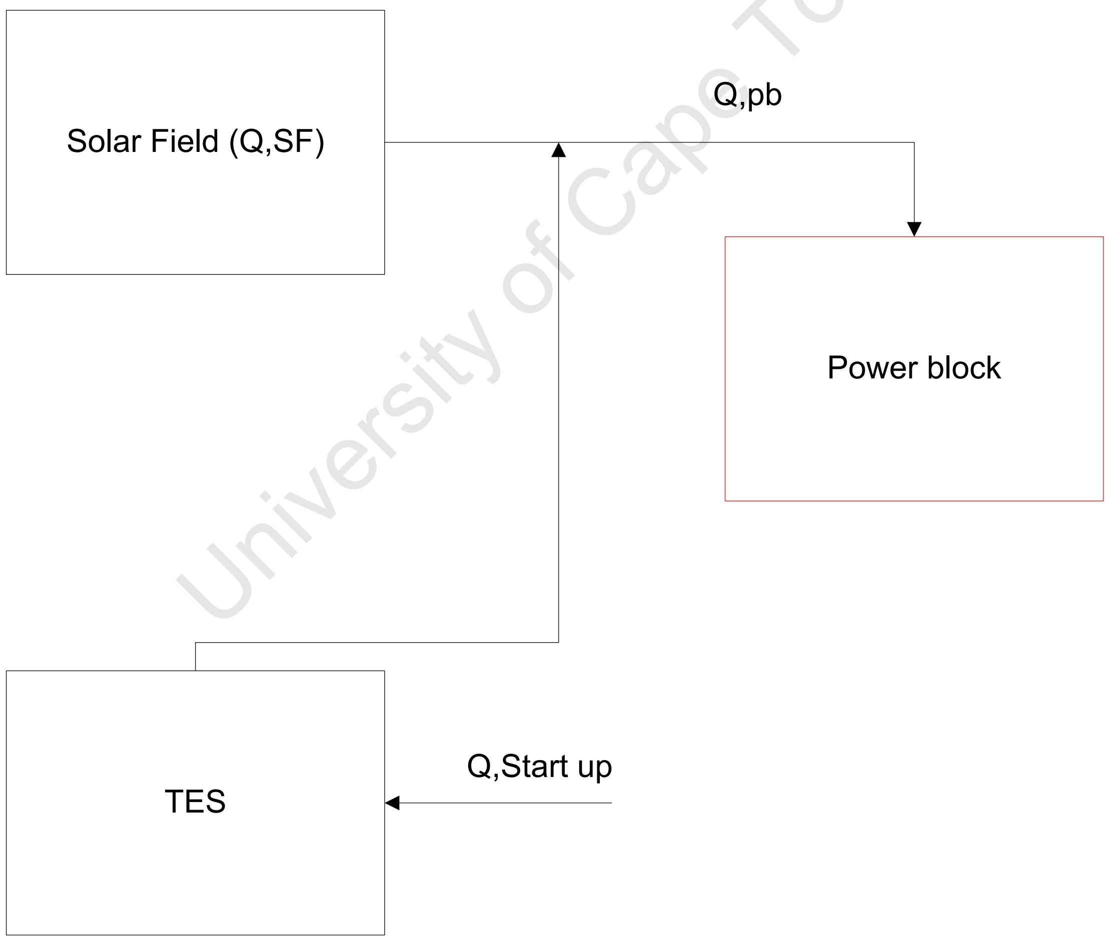

Figure 3-27 TES movement when Solar Field Energy is equal or less than Power Block load [97]

The following section discusses cooling of the parabolic trough plants.

3.17 Cooling

Parabolic trough plants are mostly located in arid regions of the world and needs cooling of the power block for proper functioning. Depending on the type of cooling type deployed the coolant is injected into the condenser where the hot exhaust steam from the turbine is flowing as shown on Figure 3-28.

The steam leaving the turbine (exhaust steam) gets into the condenser which is located beneath the turbine. The condenser can be air cooled (dry) or water cooled. Wet cooling systems use water for heat removal through evaporation and thus the temperature of the cold reservoir is driven by the wet bulb temperature. Dry cooling transfers heat directly from the steam working fluid to air using fans which blow off air on the surface of the condenser.

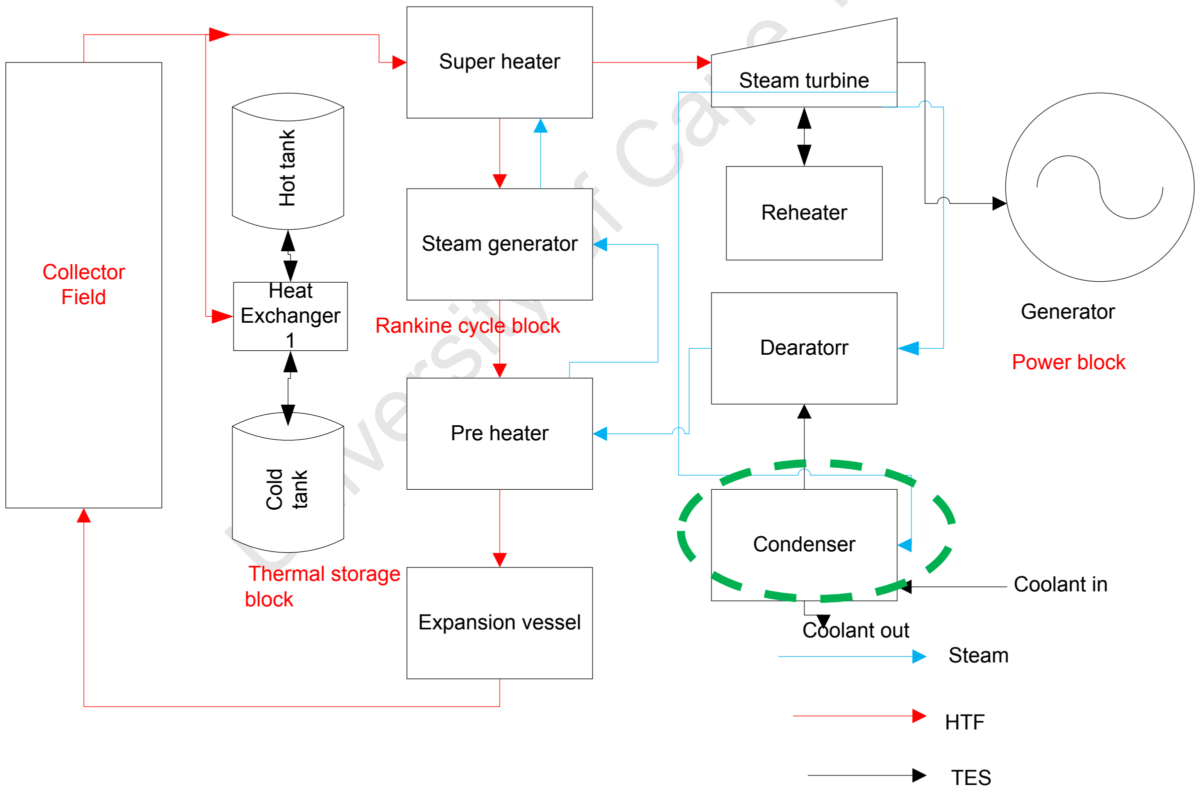

Figure 3-28 Schematic Diagram of Parabolic CSTP plant Showing Movement of the Coolant in the Condenser [38]

The following section describes the two cooling technologies their design parameters and cost.

3.17.1 Wet Cooling condenser Model

Wet cooling condensers operate by circulating water through the condenser. It has a shell on one side and a tube on the other side.

Condenser water is circulated on the tube side and the steam condenses on the shell side. This removes heat from the steam flowing from the turbine as rejected heat. In the process of cooling the coolant water rises in temperature. The rise in temperature is a function of heat rejection load and the mass flow rate of the circulating water as shown in Equation 3-29 [100].

$$\Delta T_{cw,des} = \frac{q_{rej}}{m_{cw} c_{p,cw}} \quad \textbf{....... 3-29}$$

where

$\Delta T_{cw,des}$ =cooling water rise (^{0}C)

q_{rej} = amount of heat energy rejected (MWh)

m_{cw} = mass flow rate of water in the condenser (kg/s)

$c_{p,cw}$ = the heat capacity of water (kJ/kg ^0K)

The advantages of wet cooling are

- Low installed cost: Compared to dry cooling wet cooling is cheaper to install. This is because the condensers used in dry cooling are massive and capital intensive as will be described in section 4.2.3.
- Low parasitic loads: The amount of energy drawn by motors, pumps and other electrical appliances in the CSTP plant are called parasitic loads. Water is denser than air. Hence the efficiency of heat rejection in wet cooled condensers is 5% higher than dry cooled condensers [100].

The main disadvantage of wet cooled condensers is that they use so much water for electricity production. It is reported that a parabolic CSTP plant uses 3.5m^3 for every MWh generated. Dry cooling consumes 0.3m^3 per MWh generated [101].

Figure 3-29 describes the wet cooling of a parabolic trough.

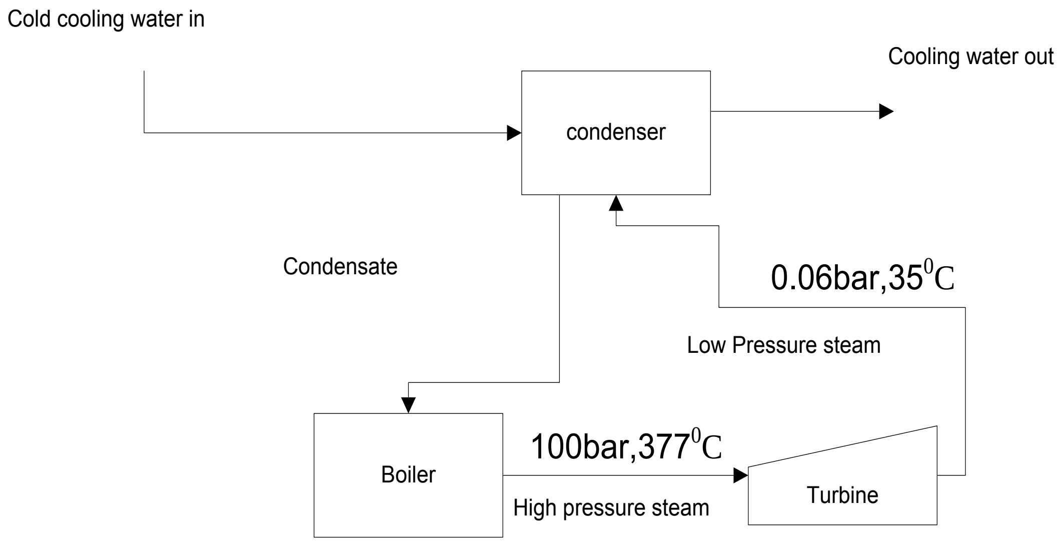

Figure 3-29 Wet cooling descriptive diagram [100]

The flow diagram of a wet cooled condenser is shown in Figure 3-30 below.

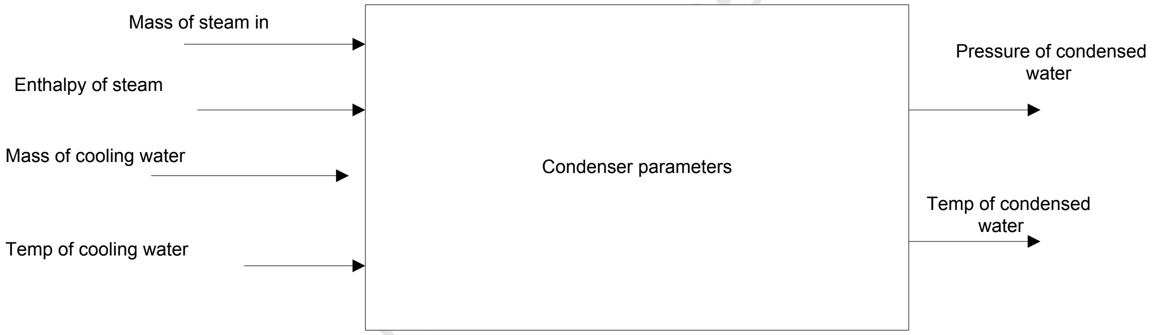

Figure 3-30 Flow Diagram of a wet cooled plant

The efficiency of the condenser is defined by equations 3-30 [114].

$$\eta_{condenser} = \frac{Q_{condenser}}{Q_{condenser\max}} \text{ .... } \mathbf{3\text{-}30}$$

where

$$Q_{condenser} = m_{steam}(h_{stea\min} - h_{steamout}) \text{ .... } \mathbf{3\text{-}31}$$

$$Q_{condener,\max} = m_{condenser}C_{condenser}(T_{steam} - T_{condenser}) \text{ .... } \mathbf{3\text{-}32}$$

where

$Q_{condenser}$=This is the amount of thermal energy carried by the exhaust steam

$Q_{condense\max}$ =Maximum amount of energy that can be handled by a condenser (kg/s)

m_{steam} =Mass of exhaust steam on the condenser (kg/s)

$C_{condenser}$= Specific heat of the condenser water (kJ/kg^{0}C)

Most wet cooled condensers are designed with an approach temperature ranging from 3^{0}C - 10^{0}C [112]. In this simulation $T_{approach}$ (the temperature difference between the cooling water and the wet bulb temperature) was varied between 3^{0}C and 10^{0}C. ΔT_{hot} (Temperature difference at hot side of the condenser) was maintained constant throughout at 3^{0}C. The wet bulb temperature was set at 22.86^{0}C which is the average wet temperature of Lodwar. The net power outputs and LCOE were recorded, LCOE, water consumption were recorded. The input parameters of the wet cooled condenser were as shown in Table 3-12.

Table 3-12 Design Parameters of Wet Cooling Model (Wagner et al, June 2011) [82]

Variable	Value	Units
Cooling water rise at condenser $\Delta T_{cw,des}$	10	^{0}C
Cooling water approach temp, $\Delta T\ approach$	10	^{0}C
Net Power output at design w_{des}	50	MWe
Power cycle efficiency η_{des}	37.1	%
Atmospheric Pressure p_{amb}	100	Bars
Temperature difference at hot side condenser, ΔT_{hot}	3	^{0}C
Drift loss fraction, f_{drift}	0.001	^{0}C
Cooling tower blow down fraction $f_{ct,bd}$	0.003	^{0}C
Circulating water pressure drop ΔP_{cv}	0.37	^{0}C
Fan isentropic efficiency $\eta_{fan,s}$	80	%
Fan pressure ration $r_{p,fan}$	1.0025	-
Efficiency of pump η_{pump}	75	%

The following section discusses dry cooling.

3.17.2 Dry Cooling Model

A dry cooled condenser forces the ambient air over some fin like structures as shown in Figure 3-31 below. The incoming exhaust steam from the turbine is condensed by the circulating air in the condenser. Dry cooling has the disadvantage such as:

- High fan power costs: Dry cooling dissipates the heat on the condenser surface using air. During the day when the ambient temperatures are high the ability of the fans to reject the exhaust steam is low. This is because the fans must draw so much air to cool a small volume of steam. This therefore calls for big strong condensers which are capital intensive and expensive to maintain.
- Higher noise levels: The blowing of air by the condensers produces a lot of noise. This causes noise pollution.

The main advantage of dry cooling is that it saves water by 90% compared to wet cooling.

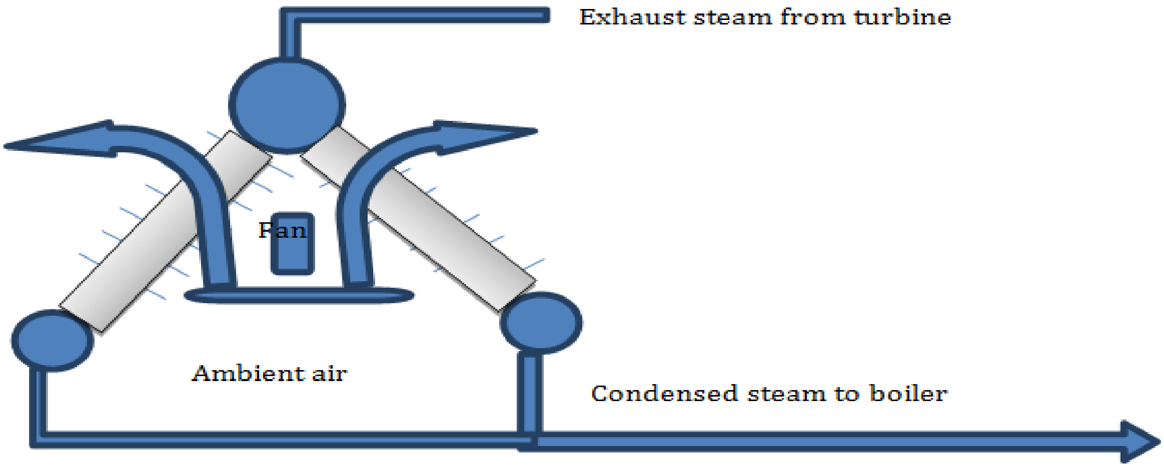

Figure 3-31 Dry Cooling Condenser [100]

SAM calculates the design steam mass flow rate by Equation 3-33 shown below [100].

$$m_{air} = \frac{q_{rej,des}}{c_{p,air}(T_{ITD,des} - \Delta T_{out})} \quad \text{....... 3-33}$$

where

$q_{rej,des}$ =the amount of heat energy rejected by the turbine at design (MWh)

$c_{p,air}$ =heat capacity of air (kJ/kg ^{o}C)

m_{air} =the mass flow rate of air getting in the condenser (kg/s).

$T_{ITD,des}$ =Initial temperature difference between the condenser and the outside air (^{0}C)

ΔT_{out} = Temperature difference at the hot side of the condenser (^{0}C)

The average dry bulb temperature of Lodwar is 29.734^{0}C. The initial temperature difference (ITD) was set at 16^{0}C which is recommended for dry cooled condensers [112]. ITD is the temperature difference between the dry bulb temperature and the set condenser temperature. The set condenser temperature is obtained by adding the average dry bulb temperature to the ITD. In this simulation the dry cooled condenser was set at 45.734^{0}C.

The ITD was varied between 5-25^{0}C to obtain the optimum condenser temperature for condenser design. The net power output, the cost of energy per kilowatt and capacity factor is recorded.

Dry cooled parabolic trough is modelled using the parameters shown in Table 3-13.

Table 3-13 Design Parameters of Dry Cooling Model [82]

Parameter	value	Unit
Initial Temp difference $T_{ITD,des}$ (Steam to ambient)	16	^{0}C
Condenser air pressure ratio $r_{p,cond}$	1.0025	^{0}C
Power output design w_{des}	50	MWe
Power cycle efficiency at design η_{des}	37.1	%
Fan isentropic efficiency $\eta_{fan,s}$	0.8	-
Specific heat of air $C_{p,air}$	1005	J/kgK
Fan mechanical efficiency η_{fan}	0.94	-
Power output at design w_{des}	50	MWe
Temperature difference at the hot side of the condenser ΔT_{out}	3	^{0}C

The following section discusses the governing scheme of a turbine.

3.18 Governing Scheme

The load on the turbine generating unit of a solar parabolic trough changes depending on the amount of DNI available and the consumer requirements. There occurs a mismatch between load and generation which leads to frequency variation. Variation of load varies the generation to maintain a constant frequency. The governing system considered in this thesis was as shown in Figure 3-32. Figure 3-33 shows the governing system.

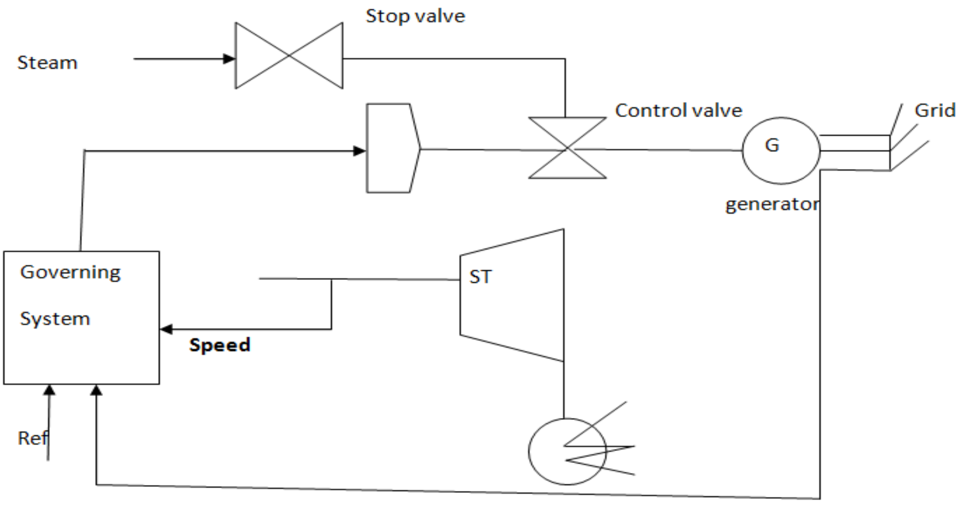

Figure 3-32 Steam Turbine Governing Scheme [108]

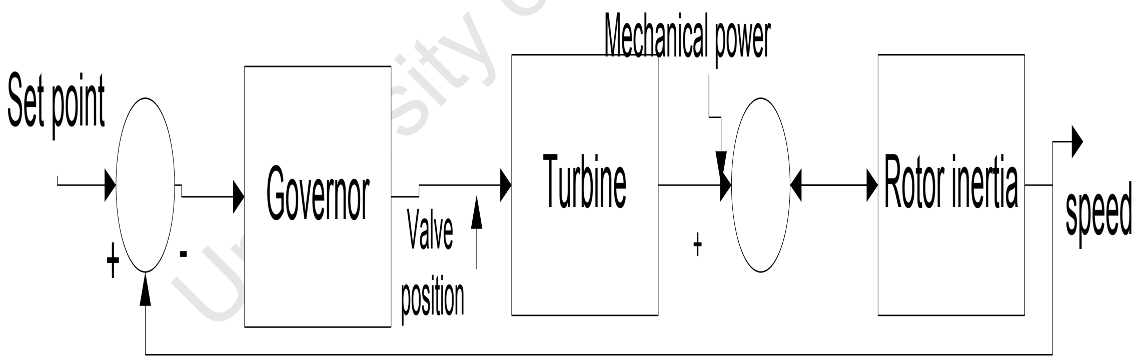

Figure 3-33 Governing System Functional Block Diagram [108]

The following section discusses the effects on power system stability due to interconnection with other power systems (local power grid).

3.19 Power System Stability

The stability of a power system is defined as the ability of electrical power system to regain a state of equilibrium after a disturbance. On the other hand power system instability is inability of a system to regain equilibrium after a disturbance. Power system stability applies to the all the interconnected generators, transmission and distribution systems. Power systems are nonlinear systems whose operation is highly dependent on loads and the generator outputs. When a power system is subjected to a disturbance (fault) its ability to resume back to the normal operation depends on the initial conditions and the severity of the fault. The types of disturbances occurring in power systems can be small or large. The small disturbances come in the form of load variations that occur continuously in the power system. Large disturbances include short circuits on the transmission lines, bus bars or even loss of generation [111].

3.19.1 Categorization of stability

Rotor angle stability is the ability of an interconnected power system to remain in synchronism even after a fault. This is dependent on the system ability to restore back its stable state between the electromagnetic torque and the mechanical torque of each of the generator unit. Rotor angle instability occurs when there are angular swings of some generators in the power system. This leads to loss of synchronism [111].

Voltage stability: This is the ability of a power system to maintain its voltage level in all the buses after a fault occurs. The ability to maintain the voltage level depends on the load demand and supply. Voltage instability occurs due to falling or rising of voltage levels on different bus bars. The results of voltage instability are loss of load in a given area and tripping of the lines [111].

Frequency Stability: This refers to the ability of a power system to maintain its frequency following a severe imbalance between the load and the generation. The restoration of frequency depends on the system's capability to maintain equilibrium between generation and load. Frequency instability leads to the inadequate machines operations, poor system coordination or even reduced generation capability [111].

The stability analysis of the modelled synchronous generator used was done using DIg SILENT power factory 14.0.

3.19.2 DIg SILENT

The synchronous generator was modelled in DIgSILENT. DIgSILENT stands for Digital Simulation and Electrical Network. It is a computer aided engineering tool used for the analysis of industrial, utility, and economic electrical power systems. The software is mainly used for power system design and optimisation. The software was designed in Stuttgart Germany [109]. In this thesis DIgSILENT power factory 14.0 was used for testing the stability of the synchronous generator.

3.19.3 Operating modes of the micro grid

The micro grid was operated in two modes i.e. standalone and grid connected. Standalone type of operation is when the micro grid is disconnected from the main electrical network. In this scenario the micro grid is expected to meet the load demands. The grid connected mode consists of the micro grid/generator linked physically to the main network. In this case the generator can take power from the grid or give power to the grid. The generator used for this simulation is synchronous motor.

3.19.4 System Modelling

The network consists of a micro grid and an external grid. The micro grid is made up of a synchronous, four loads, and two bus bars. The two bus bars are connected via transformer TFM1 which steps down voltage from 11kV to 230V. A two winding (11/230kv) transformer TFM2 is connected between bus bar 1 and bus bar 3. Some loads are connected to the system. A standalone system without the grid is shown in Figure 3-34.

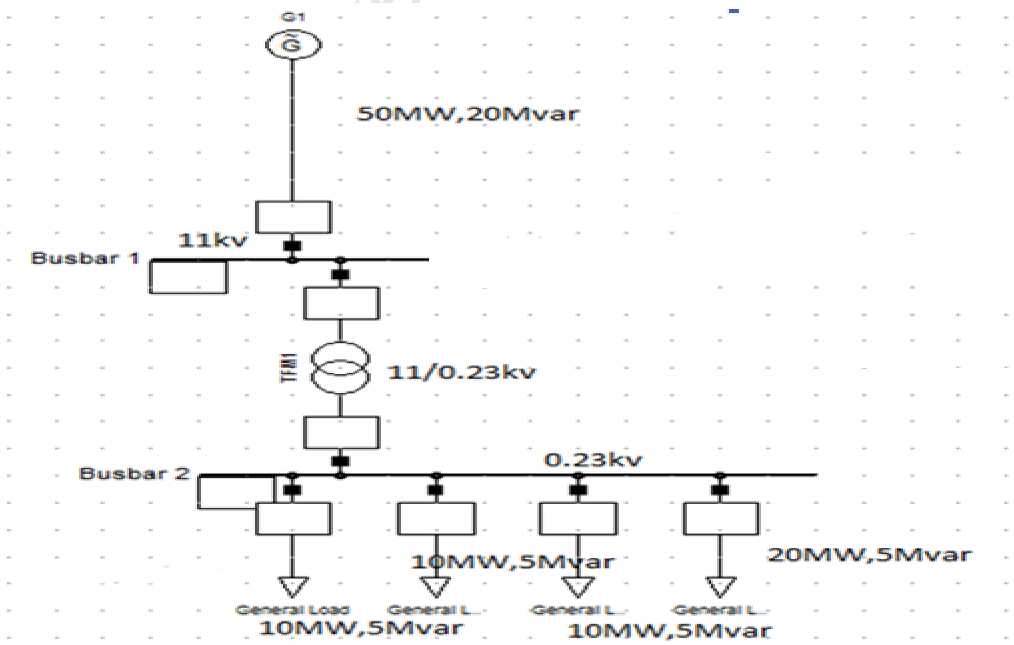

Figure 3-34 Standalone system

Figure 3-35 shows a grid connected standalone system. It consists of a 230MW grid and transformer, TFM2 connected to bus bar 1. The micro-grid is connected to a 230MW grid because it is a strong grid. A strong grid consists of a flat voltage profile, with a stable rated frequency, independent of all the loads connected to the system and the currents drawn. A grid is therefore said to be strong, when the maximum power capacity of the grid is several times higher than the size of the loads [118].

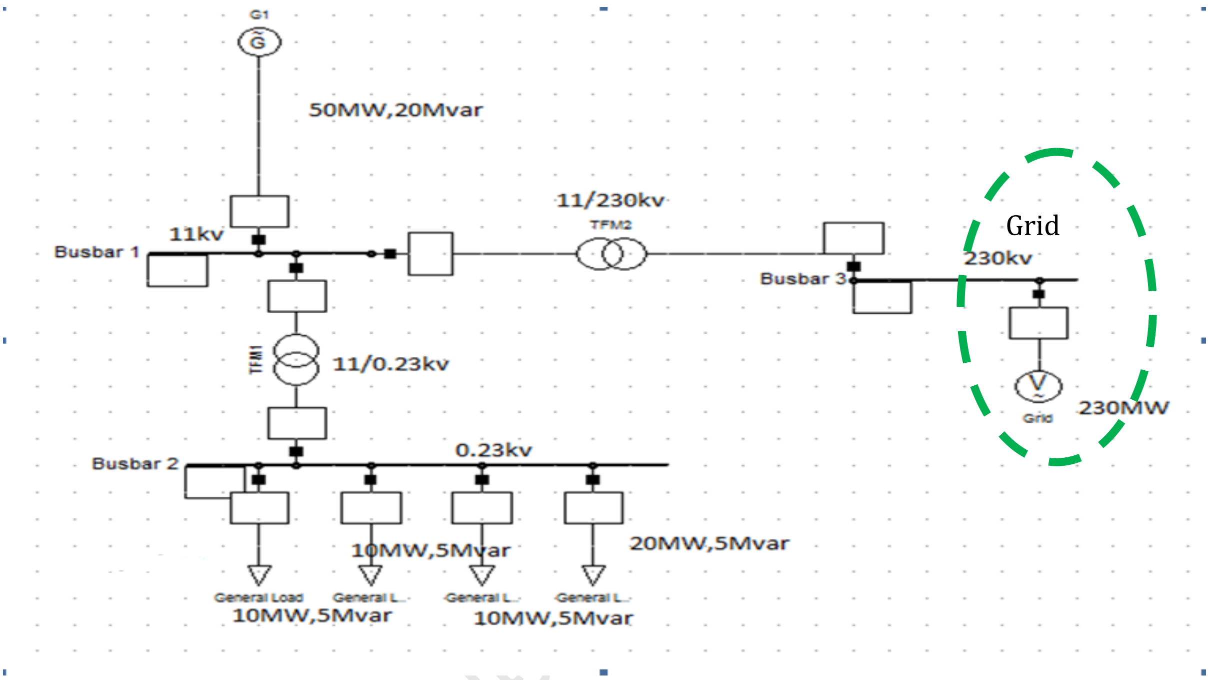

Figure 3-35 Grid connected

3.19.5 Simulation

A three phase fault short circuit was applied on bus bar 1 at 4 seconds and cleared at 4.5 seconds. This was done for both grid and micro grid (standalone) modes. The following parameters were investigated:

- Active and reactive power
- Frequency
- Voltage
- Speed

The parameters of the simulated network had the parameters shown in Table 3-14 and Table 3-15.

Table 3-14 Input Parameters used for Grid and Standalone system [117]

Components name	Voltage levels(kv)	Load	
		MW	***MVAr***
Bus bar 1	11	0.00	0.00
Bus bar 2	0.23	(10; 10;10; 20)	(5; 5; 5; 5)
Bus bar 3	230	0.00	0.00
TFM1	11/0.23	0.00	0.00
TFM2	11/230	0.00	0.00
Generator	11	50	20
Grid	230	230	1000

Table 3-15 Parameters of the Synchronous generator for stability studies [117]

Resistance	0.03	Tq'	5
X1	0.13	Xd'	0.25
Xrl	0	Xq'	0.25
Xd	5	Td	5
Xd''	0.22	Xq''	0.25

3.20 Load Profile

A load profile is an estimate of the amount of energy demanded by a given number of consumers from a power system. It is a two dimensional diagram showing energy demand (kWh) versus time in hours. A load profile gives a clear indication of how the load profile of a number of consumers changes with time. Load profile estimation is important in the estimation of the size of power systems generation capacities. For example the size of battery storage depends on the amount of energy that will be drawn by the loads. Determination of a load profile is also very important in efficiency calculations [113]. This helps in loss estimation on a power system as explained by Equation 3-34 below.

$$\%\,\text{Efficiency} = \frac{\text{Total energy generated/stored} - \text{Total energy delivered}}{\text{Total energy generated/stored}} \quad \textbf{....... 3-34}$$

The following section describes how the load profile realisation of Lodwar was done.

3.20.1 Load profile for Lodwar

In the load profile realisation of Lodwar it was assumed that each household had all the appliances listed in Table 3-16.

The monthly rating of each electrical appliance was estimated by Equation 3-35

$$\text{Monthly power consumption} = \text{power rating x period of usage x no of appliances} \quad \textbf{....... 3-35}$$

For example the amount of energy consumed by the 10 light bulbs is calculated as shown in Equation 3-36.

$$0.03X10X210 = 63kWh \quad \textbf{....... 3-36}$$

The monthly energy consumption of one consumer is **711.86kWh.**

Table 3-16 Estimated Household Monthly Power Consumption in a Rural Kenyan Household [95, 96,107]

Appliance	Power rating (kW)	No, of appliances	No of hours per day	No of hours per month (30 days)	Average kWh per month
light bulbs	0.03	10	7	210	63
Cell charger	0.008	2	2	32e-3	1
Fridge	0.208	1	6	180	34
Oven	0.8	1	6	180	144
Microwave	1	1	0.16	5	5
Washing machine	0.5	1	0.5	5	2.5
Domestic heaters	2	2	12	360	380.7
TV	0.3	1	6	180	54
Laptop	0.03	1	6	180	54
Geyser	0.486	1	7.5	225	114.3
Total					**711.86**

3.20.1.1 Realisation of the Load Profile

The amount of energy consumed hourly was estimated according to Table 3-17. A day was divided into periods. Each period had its own electricity consumption schedule. For example the amount of energy consumed from 12am to 5am in the morning assumed only the security lights are on, geysers and fridge. The total amount of energy consumed by these gadgets at that time span is added. An average of the sum is then taken to obtain hourly energy consumed for one consumer. The hourly average is multiplied by 17,000 to obtain the total demand at that particular hour. In this calculation it is assumed that all the consumers use energy equally at the same period.

Table 3-17 Load Profile Realisation

Period	Time of Day	Comments	Consumption of 17000 Households(kWh)
1	12am-5am	Night time with 0 irradiation,medium load consumption from security lights,geysers and Fridge	20060
2	6am-7am	Time when most consumers wake up to use boilers,geysers and other elecrical appliances.	27880
3	7am-4pm	Working hours when most consumers are not at home	1360
4	5pm	The periods when lights are turned on along with 24/7 electrical appliances	13430
5	6pm	Consumers back at homeand using geyser at a higher consumption rate,cooking stoves,heaters etc in the assumption that all the consumers are using the hot water.	52190
6	7pm-11pm	The period when TV and computer devices are switched on , heaters and other 24/7 electrical appliances	35190
7	8am-4pm	Consumers at home watching tv ,playing games,washing clothes,cooking. This excludes lights.	13430(Same as period 4)

Hourly consumption of each household was broken as shown in Table 3-18 and Figure 3-36.

Table 3-18 Hourly daily load profile breakdown for each Consumer for Lodwar Kenya

Time of day	Consumption(kWh)	Time of day	Consumption(kWh)
0000	1.18	1200	0.08
0100	1.18	1300	0.08
0200	1.18	1400	0.08
0300	1.18	1500	0.08
0400	1.18	1600	0.08
0500	1.18	1700	0.79
0600	1.64	1800	2.07
0700	0.08	1900	3.07
0800	0.08	2000	2.07
0900	0.08	2100	2.07
1000	0.08	2200	2.07
1100	0.08	2300	2.07

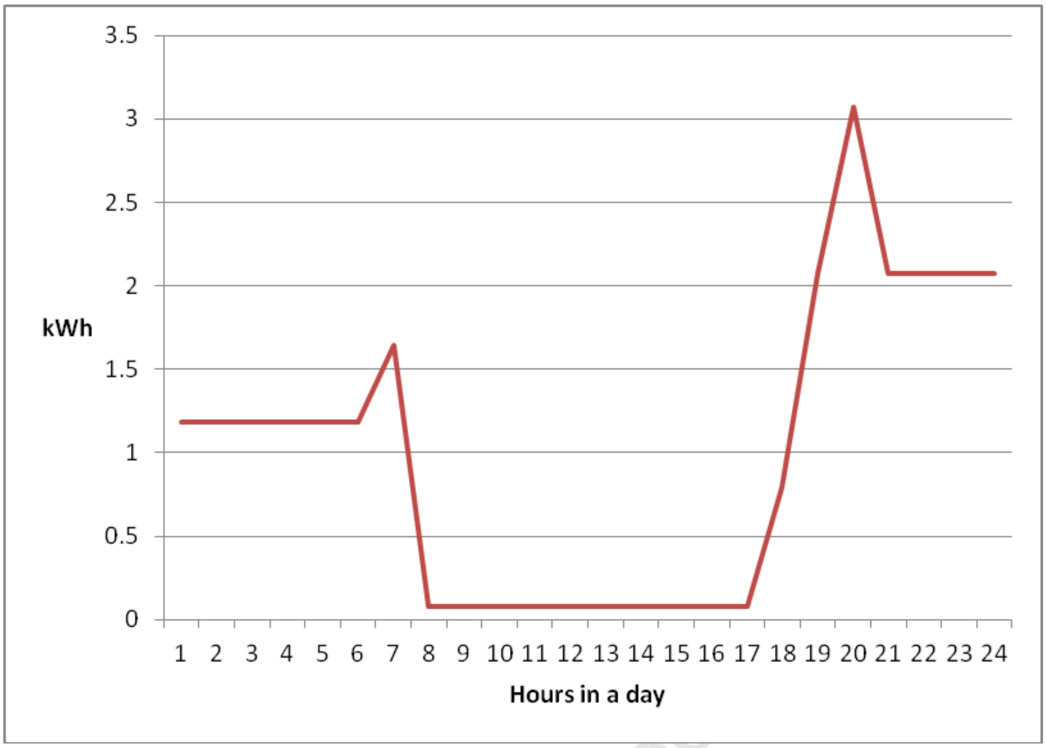

Figure 3-36 Daily Residential Consumption Profile of one consumer in Lodwar Kenya

Table 3-19 and Figure 3-37 shows the monthly demand of 17000 households of Lodwar Kenya. Consumption of electricity is low between 7am in the morning and 3pm as it is assumed that most people have left for work. Demand starts peaking from 5pm when everyone is at home cooking, washing clothes and dishes, watching television etc.

Table 3-19 Hourly Electricity consumption o 17,000 Households

Time of day	Consumption(kWh)	Time of day	Consumption(kWh)
0000	20060	1200	1360
0100	20060	1300	1360
0200	20060	1400	1360
0300	20060	1500	1360
0400	20060	1600	1360
0500	20060	1700	13430
0600	27880	1800	35190
0700	1360	1900	52190
0800	1360	2000	35190
0900	1360	2100	35190
1000	1360	2200	35190
1100	1360	2300	35190

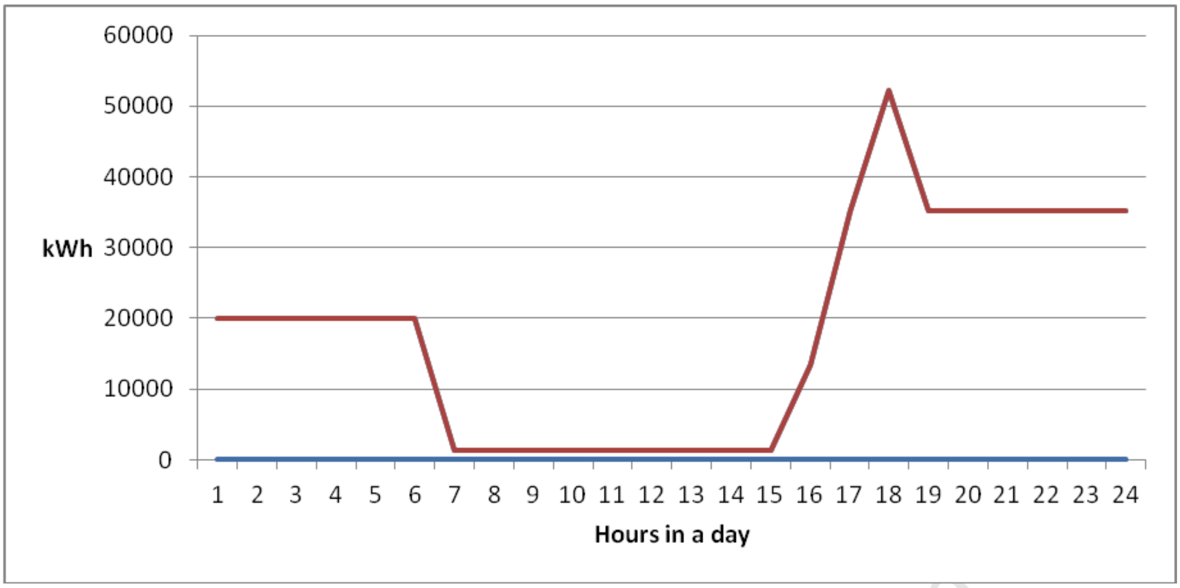

Figure 3-37 Hourly Electricity Consumption for 17000 Households

3.21 Cost

In SAM the total project costs are divided into two categories i.e. capital and operation & maintenance costs. The capital costs are incurred in the purchasing materials for building the plant. The capital costs are divided into direct and indirect costs. Direct costs are expenses used for purchasing a particular item, for example, buying collectors and receivers, installation of the TES system, turbine, generators, and transmission lines for connecting to the grid, motors and drives. Indirect expenses are service costs. These costs cannot be identified with buying of equipments. They include costs used for shipping, paying engineers and consulting, procurement costs, and paying workers during construction of the plant. SAM uses the total costs of the plant, both direct and indirect to calculate the LCOE, cash flows, net present value among other metrics.

Operation and maintenance costs are the costs incurred in the daily running of the plant after construction.

In the following section the total investments made on a dry and wet cooled plant are discussed and tabulated. They are used as inputs to the SAM cost user interface window.

3.21.1 Summary of Cost of dry cooled Model

The air cooling condensers add 5% to the capital cost which further increases the cost of electricity by 10% [67]. Dry cooled condensers uses air for heat dissipation at the condenser surface as was previously discussed in section 3.17.2. They are expensive to build and maintain. Table 3-20 shows the per item cost while Table 3-21 shows the total investment of the dry cooled parabolic trough.

Table 3-20 Per Item costs, Dry cooling [16, 69,101]

Item	Cost ($/kW)
Solar Field	315$/m^2
Fossil backup	0$/kW
Site Improvements	35$/m^2
HTF system	105$/m^2
Heat Storage	95$/kWh-t
Power Plant	1000$/kW
Cost of land	2% of total investment

Table 3-21 Dry Cooled Parabolic Trough Total Investment Costs

Item	Cost($)
Site Improvements	14,802,200
Solar Field	133,219,800
HTF System	44,406,600
Storage	166,136,724
Power Plant	55,000,000
Contingency costs	28,949,572
Owners and EPC costs	48,676,638
Land Cost	3,816,650
Sales Tax	17,700,595
Totals	**512,708,783**

3.21.2 Summary of Cost of wet cooling model

SAM calculates total investment cost on each item based on the unit price of each item.

Table 3-22 shows the per item cost of wet cooling parabolic trough plant. The total investment cost for the wet cooled parabolic trough was shown in Table 3-23.

Table 3-22 Summary of investment costs, Wet cooling [16, 69,101]

Item	Cost
Solar Field	295$/m^2
Fossil backup	0$/kW
Site Improvements	30$/m^2
HTF system	95$/m^2
Heat Storage	80$/kWh-t
Power Plant	940$/kW
BOP	0$/kWe
Cost of land	2% of total investment

Table 3-23 Wet Cooled Parabolic Trough Total Investment Costs

Item	Cost($)
Site Improvements	12,687,600
Solar Field	124,761,400
HTF System	40,177,400
Storage	139,904,610
Power Plant	51,700,000
Contingency costs	25,846,170
Owners and EPC costs	43,458,489
Land Cost	3,816,650
Sales Tax	15,803,087
Totals	**458,155,409**

3.21.3 Operations and maintenance costs of dry and wet cooled plants

The values of the plant's operations and maintenance costs for both wet and dry cooled plant are as shown in Table 3-24 below.

Table 3-24 Operations and Maintenance costs of Wet and Dry cooled Plants [16, 69,101]

Indirect cost category	Wet cooled plant	Dry cooled plant
Engineer/procure/construct	15%	20%
Project/land/management	3.5%	8.5%
Sales Tax (VAT)	7.75%	7.75%
Variable cost by generation	3$/kW	3.15$/kW
Fixed cost by capacity	70$/kW	73.5$/kW
PPA Escalation	1.2%	1.2%
Minimum required IRR	15%	15%
Availability	96%	96%
Salvage value	10%	10%
Contingency	10%	10%

3.21.4 Financials

In the simulation it is assumed that the plant is an independent power producer (IPP). IPP plants earn money through selling electricity to cover the total project costs incurred. In this simulation 40% of the total project cost is borrowed from the bank. The project lifetime is 30 years. Other base case assumptions made during modelling were as shown in Table 3-25.

Table 3-25 Cost Assumptions made During Modelling and Simulation of Parabolic Troughs [99,112]

Location	Lodwar Kenya
Project lifetime	30 years
Down payment (%)	20%
Loan interest rate in Kenya as per February 2012	19.9%
Equity rate of return	14%
Project discount rate	10%
Insurance	0.5% of installed cost
Internal rate of Return (IRR)	15%
Debt fraction	40%
Investment tax credit	10%
Tax rate (Kenya)	30%
Ownership	IPP
Depreciation(Straight line)	30 years
General inflation rate in Kenya as per February 2012	16%
Loan term	20 years

3.22 Chapter Review

This chapter modelled the parabolic trough defining all the parameters and assumptions made during simulation. The main blocks (Solar field, TES, steam Rankine cycle, power block) making up the parabolic trough plant were explained and their parameters tabulated. The cooling methods were introduced, defined and their input parameters tabulated. The simulation was run using SAM.

The following chapter discusses the results obtained in this research work.

4 Results and Analysis

4.1 Introduction

The results are discussed based on the impacts each cooling system has on the following:

- Energy production : In this section the following aspects are considered:
    - Energy production using wet cooling
    - Energy production using dry cooling
    - Efficiency of solar heat energy collection
    - Capacity factor
    - Solar Electric Efficiency
- Water consumption
- Land Usage
- Thermal energy storage (TES)
- Solar Multiple
    - Hourly analysis at different solar multiples
- ITD on dry cooling
- Approach temperature variation on wet cooling
- Sensitivity analysis of parabolic trough performance, costs and financing due to inflation
- Cash flow
- After tax cash flow of the parabolic CSTP plants: This is a measure of the project ability to generate revenue in its operations. It is obtained by adding back all the non-cash accounts which includes amortization, depreciation, costs incurred in restructuring and impairment to the net income.
- carbon dioxide avoided by a 50MWe parabolic trough CSTP plant and the impacts of dry and wet cooled parabolic trough plants on flora and fauna
- Stability analyses for grid-connected and stand-alone CSTP plant: This is used to show the stability of the CSTP system due to a fault on the parabolic trough CSTP plant. This is done for a standalone and grid connected modes of operations of a synchronous generator.

4.2 Energy production

In this section energy production of the parabolic trough CSTP plant with dry cooling and with wet cooling is discussed. The capacity of the plant is 50MWe with 12 hours of thermal storage at SM of 2. As was previously discussed on section 3.10.1 there exist relationships between the incident energy, absorbed energy and field output energy. The incident monthly energy is the amount of thermal energy received by the solar field (collectors) from the sun. It is a function of DNI and the collector surface area. This is shown by Equation 4-1.

$$\text{Incident energy} = \mathrm{I}_{\text{coll}} X\, A_{coll} \quad \textbf{....... 4-1}$$

where I_{coll} is the average DNI received in that day /month/year (Wh/m^2) and A_{coll} is the surface area of the collector field (m^2).

The absorbed solar thermal energy is the amount of thermal energy that is absorbed by the receivers. It is abbreviated as Q_{abs} (MWh). The net amount of energy absorbed by the receivers is called the field output energy (Q_{u}). It is the difference between the amount of energy absorbed by the receivers and the losses. The losses include emission losses of the receiver tube, and radiation losses, piping losses and amount of energy used for pumping HTF etc. This is shown in Equation 4-2.

$$\text{Field output thermal energy}(\mathrm{Q}_{\mathrm{u}}) = \text{absorbed thermal energy}(\mathrm{Q}_{\text{abs}}) - \text{losses}(\mathrm{Q}_{\text{loss}}) \quad \textbf{....... 4-2}$$

The monthly net output energy (kWh) is a function of the total mechanical power output of the turbine (E_{gross}) and the electric conversion efficiency of the synchronous generator as shown in Equation 4-3.

$$\text{Monthly net energy } (\mathrm{E}_{\text{net}}) = \mathrm{E}_{gross}\, \eta_{\text{el}} \quad \textbf{....... 4-3}$$

where

η_{el} = the power conversion efficiency from mechanical to electrical by the synchronous generator. Most steam generators have achieves an efficiency of 90%.In this simulation η_{el} used was 90%.

4.2.1 Energy Production with Dry Cooling

The monthly net energy, total amount of energy absorbed by the receivers, the amount of energy from the field, the amount of incident energy and monthly average DNI achieved by the dry cooled plant are shown in Table 4-1.

The highest net energy production is recorded in the month of September which corresponds to the month with the highest irradiance at design used in this research. This is attributed to the fact that in September DNI is highest hence a high thermal energy collection results in a high energy output from the plant.

From the definition of DNI as shown by Equation 4-4, the higher the DNI, the higher would be amount of thermal energy collected.

$$DNI_{design} = \frac{Q_{sf,design}}{A_{sf,design} x \eta_{sf}} \text{ .... } \textbf{4-4}$$

where;

DNI_{design}= Design irradiance (kWh/m^2/d)

$Q_{sf,design}$=heat input to the power plant (MWh)

$A_{coll,design}$=Solar field area (m^2)

η_{sf} = solar field efficiency

In Lodwar the cold season starts in March and ends in May; therefore DNI is low for that period and so is the energy production.

The amount of energy produced annually is found to be 128,094,592 kWh which is a summation of the monthly electricity generated by the dry cooled parabolic trough CSTP plant as shown in Figure 4-1.

The solar electric efficiency of the dry cooled plant is calculated using Equation 4-5[21].

$$\text{Solar Electric Efficiency} = \frac{\text{Annual Net power generation}}{\text{Annual Direct irradiance on aperture}} X100 \text{ .... } \textbf{4-5}$$

The kWh to BTU conversion used is given by Equation 4-6 [103].

$$1\,\text{kWh} = 3412.14\text{btu} \text{ .... } \textbf{4-6}$$

The calculation of solar electric efficiency of a dry cooled plant is done below.

$$Solar\ electric\ \ efficiency\ \ of\ \ dry\,cooled\ \ plant = \frac{128{,}094{,}592}{937{,}084{,}000}\,x100 = 13.66\%$$

Table 4-1Absorbed Energy, Field Energy, Incident Energy, Efficiency of Energy collection, Monthly Energy for Dry cooled CSTP plant

Month	Monthly net Energy (kWh)	Absorbed Energy (MWh)	Field Output Energy (MWh)	Incident Energy (MWh)	DNI(Wh/m^2)
January	1.137×10^7	48,002.6	45,647.4	84,431.5	222.46
February	1.067×10^7	45,048.1	42,825.2	74,371.4	216.95
March	9.9×10^6	42,496	40,393.6	68,523.9	180.55
April	8.835×10^6	37,755.2	35,948.3	61,932.6	168.62
May	9.35×10^6	39,520.6	37,675	67,688.3	178.35
June	1.07×10^7	44,225.3	42,302.3	77,099	209.92
July	1.17×10^7	48,679.9	46,487.6	83,285.3	219.44
August	1.35×10^7	55,497.8	53,330.3	89,972.4	237.06
September	**1.53×10^7**	**61,863.6**	**60,289.7**	**98,220.5**	267.42
October	1.19×10^7	48,818.1	47,482.2	80,629.6	212.45
November	9.22×10^6	39,237.3	37,766.9	70,035.4	190.68
December	1.05×10^7	43,922.1	42,028.4	80,869.8	213.08
Totals	**128,094,592**	**555,062**	**485,685**	**937,084**	**2516.98**

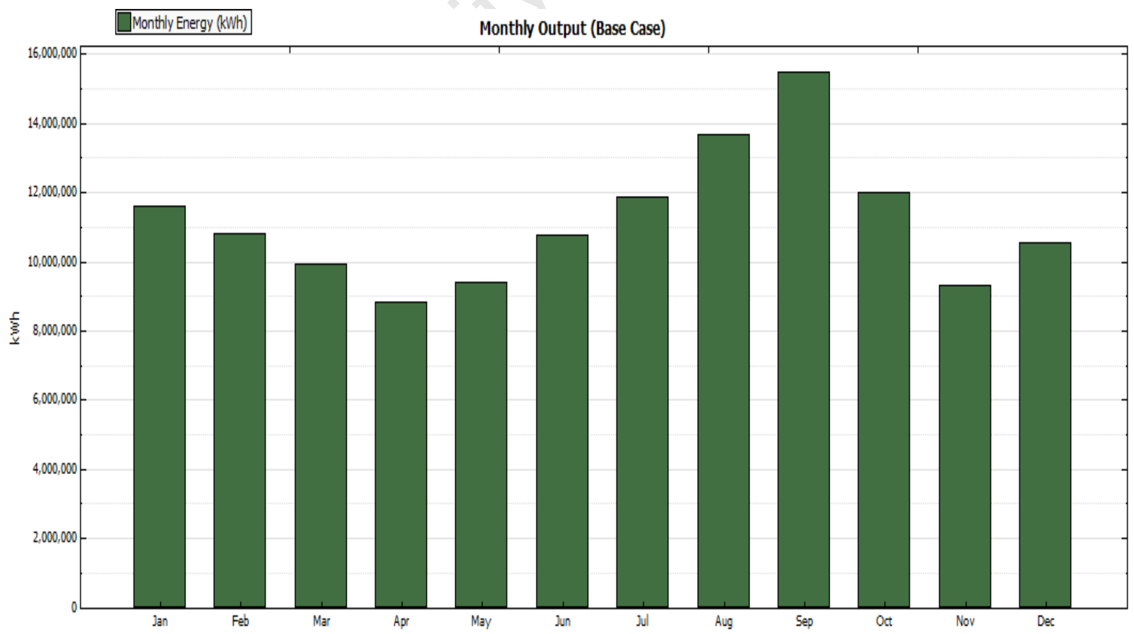

Figure 4-1 Monthly Energy Output for Dry Cooled Plant

4.2.2 Energy Production with wet Cooling

Table 4-2 shows the monthly incident energy, energy absorbed by the receivers, energy output from the solar field to the power block and the monthly net energy obtained from the power block for a wet cooled plant

Table 4-2 Absorbed Energy, Field Energy, Incident Energy, Efficiency of Energy collection and Monthly Energy for wet cooled Parabolic Trough

Month	Monthly net Energy (kWh)	Absorbed Energy (MWh)	Field output Energy (MWh)	Incident energy (MWh)	DNI(Wh/m^2)
January	1.43×10^7	48,013	45,679.5	84,431.5	222.46
February	1.34×10^7	45,059.3	42,849	74,371.4	216.95
March	1.25×10^7	42,505.5	40,421.3	68,523.9	180.55
April	1.10×10^7	37,763.4	35,968.6	61,932.6	168.62
May	1.16×10^7	39,528.3	37,696.4	67,688.3	178.35
June	1.33×10^7	44,234.3	42,328.3	77,099	209.92
July	1.46×10^7	48,688.3	46,511.4	83,285.3	219.44
August	1.68×10^7	55,507.4	53,352.9	89,972.4	237.06
September	$\mathbf{1.90} \times 10^7$	**61,887.1**	**60,332.3**	**98,220.5**	267.42
October	1.49×10^7	48,829.5	47,510.8	80,629.6	212.45
November	1.16×10^7	39,246.2	37,788.3	70,035.4	190.68
December	1.31×10^7	43,931.5	42,053.6	80,869.8	213.08
Totals	**135,447,558**	**555,190**	**532,487**	**869,006**	**2,516.98**

The energy produced for the wet cooled plant is 135,447,558kWh which is shown in Figure 4-2. Highest net thermal energy collection and highest net energy produced corresponds to the month of September.

The solar electric efficiency for the wet cooled plant is calculated using Equation 4-5, section 4.2.1, on and found to be 15.58%. The calculation of solar electric efficiency of a wet cooled plant is done below.

$$Solar\ electric\ \ efficiency\ \ of\ wet\ cooled\ \ plant = \frac{135{,}447{,}558}{869{,}006{,}000}\, x100 = 15.58\%$$

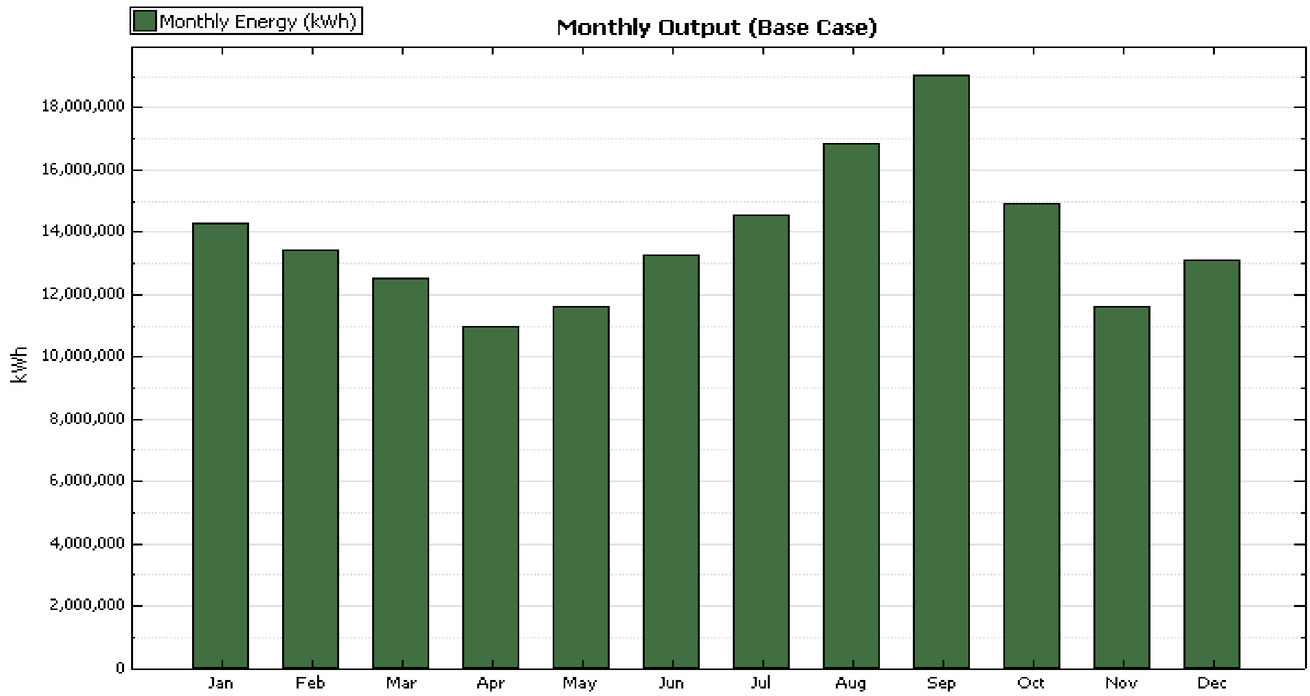

Figure 4-2 Monthly Energy Output for Wet Cooled CSTP Plant

4.2.3 Efficiency of thermal energy collection

Efficiency of solar thermal energy collection is defined by Equation 4-7.

$$\text{Efficiency of thermal collection} = \frac{\text{Q incident - Qabsorbed}}{\text{Q incident}} \quad \textbf{....... 4-7}$$

The monthly variations of thermal energy collection efficiency for both dry and wet cooled plants are shown in Table 4-3 and Figure 4-3.

Table 4-3 Monthly Variation of Thermal Energy Collection Efficiency

Month	Solar collection efficiency (%)(Dry Cooling)	Solar collection efficiency (%)(Wet Cooling)
January	43.15	43.13
February	39.42	39.41
March	37.98	40.96
April	39.02	39.01
May	41.61	41.6
June	42.64	42.63
July	41.55	41.54
August	38.31	38.31
September	**37.23**	**36.99**
October	39.45	39.44
November	43.97	43.96
December	45.68	45.68

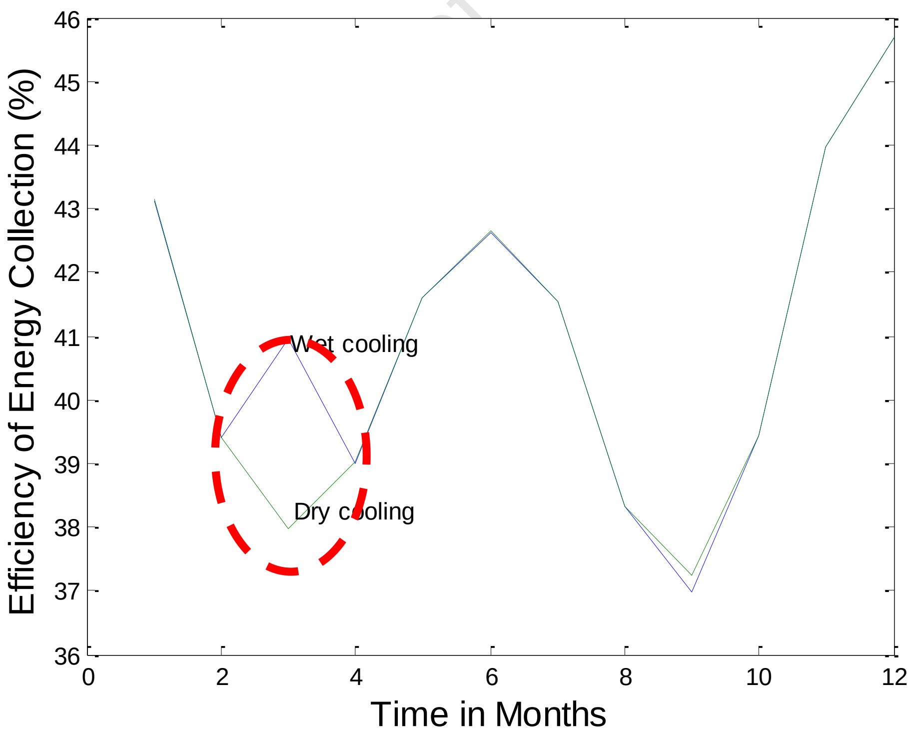

Figure 4-3 Comparison of Efficiency of thermal energy collection between Dry and Wet Cooling

It is shown that the values of thermal energy collection efficiency for the wet and dry cooled parabolic trough plants are different for the period of February to April. For the dry cooling the efficiency reduces while that of wet cooling increases. The reason behind this is that in Lodwar these months experience the highest dry bulb temperature. The initial temperature difference (ITD) between the condenser and the ambient air is lowered; hence the ability of the fans to reject all heat on the surface of the condensers is reduced. Initial temperature difference is the difference between the set dry condenser temperature and average dry bulb temperature in a given location. Most dry cooled condensers have an ITD of 16^0C [112]. The set condenser temperature is achieved by adding 16^0C to the average dry bulb temperature of the whole year. In Lodwar the average dry bulb temperature is 29.734^0C as shown on section 3.7.2, Table 3-2. The dry cooled temperature was hence set at 45.734^0C (29.734^0C $+16^0$C). This is illustrated on Figure 4-4 below.

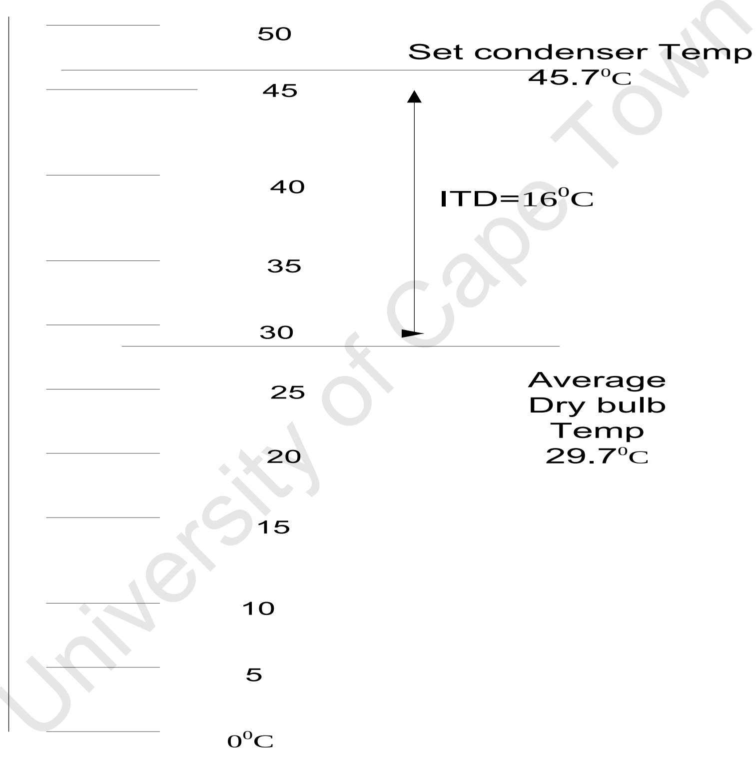

Figure 4-4 Representation of dry bulb temperature and Initial Temperature Difference

At higher dry bulb temperatures (ambient) more energy is drawn from the plant to cool a certain mass of exhaust steam arriving on the condenser from the steam turbine. This is illustrated by Equations 4-8 and 4-9.

$$H_1 = C_{p_{water}} X\, m_1\, X\, \Delta T_1 \text{.......} \textbf{ 4-8}$$

$$H_2 = C_{p,air}\, X\, m_2\, X\, \Delta T_2 \text{.......} \textbf{ 4-9}$$

where

H_1 = Amount of heat energy held by a certain quantity of heat arriving at the condenser (Joules)

H_2 =Amount of heat energy drawn by fans H_1

$C_{p_{water}}$ =specific heat capacity of water (4.2kJ/kg. ^{0}C)

$C_{p,air}$ = specific heat capacity of air (1 kJ/kg. ^{0}C)

m_1 =mass of exhaust on the surface of condenser per second. It depends on the amount of DNI. At high DNI values the amount of exhaust steam generated is higher.

m_2 =mass of air to cool H_1

ΔT_1 = Temperature difference between the set condenser temperature and ambient air (ITD).

ΔT_2 =Temperature difference between the set condenser temperature and the exhaust steam (ITD).

In this thesis the exhaust steam exits the turbine at 105^{0}C.

Therefore ΔT_2 =105^{0}C-45.734=59.266^{0}C.

For example the average temperature on the 8th and 15th January 2012 in Lodwar was 23.571^{0}C and 35.1097^{0}C respectively. Assuming m_1 =10kg/s, the amount of exhaust heat energy from the turbine per second will be 2,489J as calculated in Equation 4-10.

$$\mathrm{H1} = 4.2 \mathrm{X} 10 \mathrm{X} 59.266 = 2{,}489 \mathrm{Joules} \text{ .... } \mathbf{4\text{-}10}$$

The amount of energy drawn by the fans to counter this amount of heat on the 8th and 15th January is as calculated in Equation 4-11 where m is the mass of air

$$\text{Energy drawn on 15th} = 1 \mathrm{X}\, \mathrm{m}\, \mathrm{X}\, (45.734 - 35.1097) = 10.6243\mathrm{m} \text{ .... } \mathbf{4\text{-}11}$$

The mass of air drawn on 15th day of January to cool the condenser as calculated from Equations will therefore be 248.9kg as shown in Equation 4-12.

$$\mathrm{m}_{15\mathrm{th}} = \frac{2489\mathrm{J}}{10.6243} = 248.9kg \text{ .... } \mathbf{4\text{-}12}$$

On the 8th day of January the amount of energy drawn by the fans to cool the condenser is calculated as shown in Equation 4-13.

$$\text{Energy drawn on 8th} = 1 \mathrm{X}\, \mathrm{m}\, \mathrm{X}\, (45.734 - 23.571) = 22.163\mathrm{m} \text{ .... } \mathbf{4\text{-}13}$$

The mass of air drawn and re-circulated on the on 8th day of January to cool the condenser is 112.3kg as shown in Equation 4-14.

$$m_{8th} = \frac{2489J}{22.163} = 112.3kg \text{ .... } \mathbf{4\text{-}14}$$

This therefore implies that more energy is drawn by the fans to cool the plant at higher dry bulb temperatures. Cooling of a certain mass of steam at lower ITD therefore takes much more time than at higher ITD. This lowers the rate of heat exchange between water and the HTF. At this point the HTF reaches its maximum temperature to absorb the concentrated heat on the absorbers. The storage system has already reached its maximum heat capacity and cannot absorb more heat. The collectors therefore defocus to reduce the amount of energy absorbed by the receivers. This lowers the efficiency of energy collection for a dry cooled plant.

However during the same period from February to April, the wet cooled plant is seen to be more efficient than a dry cooled plant and its efficiency is found to increase. Wet cooling utilises water as the cooling medium which relies on the wet bulb temperature. Wet bulb is the temperature achieved by a moistened thermometer in flowing air. The approach temperature is the temperature difference between the wet bulb temperature and the incoming water for condenser cooling.

Most wet cooled condensers are designed with an approach temperature ranging from 3^0C - 10^0C [112]. In this thesis the wet cooled condenser designed, had an approach temperature of 10^0C. The average wet bulb temperature in Lodwar is 22.86^0C [72] and therefore a condenser temperature of 32.86^0C = (22.86^0C+10^0C), is used as shown in Figure 4-5.

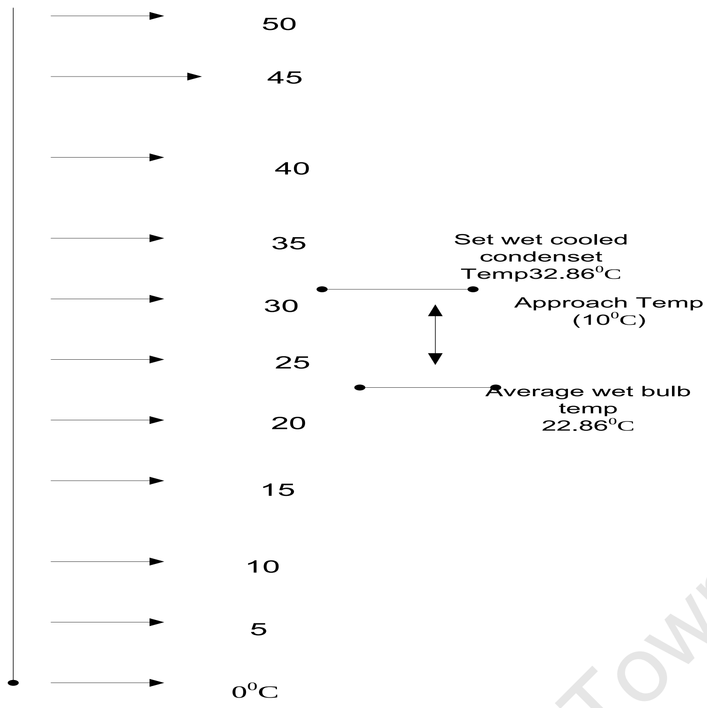

Figure 4-5 Representation of wet bulb Temperature and approach Temperature

Water is denser than air; hence it can remove heat from the condenser surface faster and more efficient than dry cooling. This allows for more heat absorption by the HTF and hence the efficiency of thermal energy collection is high for wet cooled plants.

In the month of September, the efficiency of dry cooled plant is found to be slightly higher than the wet cooled plant. This is because the dry bulb temperature is generally low in this month and does not approach the design point temperature difference (ITD). Therefore the amount of energy drawn by the fans to reject heat is minimal. In the wet cooled plant the temperature of the cooling water in the ponds rises as a result of re-circulation in the condenser. This leads to a lower approach temperature (temperature difference between the cooling water and the wet bulb temperature). It reduces the ability of the condenser to reject the incoming exhaust heat from the turbine. Therefore the amount of water re-circulated to cool a certain quantity of exhaust heat from the turbine increases. This further delays the process of heat exchange between water and HTF. The HTF reaches its maximum specific heat capacity, hence unable to absorb more heat from the collector field. This leads to collector defocusing which reduces the thermal collection efficiency.

4.2.4 Capacity Factor

Capacity factor of a plant is defined as the ratio of the actual energy produced in a given period of time to the hypothetical maximum energy production possible by the plant, i.e. running full time at rated power output [59].

For example the rated power output of the generator considered in this thesis is 50000kW. Hypothetically if the generator is operated for 24hours a day for 365days, the total energy production would be:

$$(50{,}000)\ \mathrm{X}\ (365\mathrm{X}24) = 4.38\mathrm{X}108\ \mathrm{kWh\ per\ year.}$$

If the actual energy produced is $2.25\mathrm{X}10^{8}$ kWh then the capacity factor is calculated as

$$\frac{2.25\mathrm{X}10^{8}}{4.38\mathrm{X}10^{8}}\ X100 = 51.4\%$$

This can be expressed as shown in Equation 4-15.

$$\text{Annual Capacity Factor} = \frac{\text{Actual Energy produced}}{\text{Days in a year x 24 x plant capacity}} \quad \textbf{....... 4-15}$$

The same capacity factor can also be calculated for a month by replacing ‘‘Days in a year” by “Days in a month” in Equation 4-15. For example in the month of January (for the dry cooled plant) the capacity factor is calculated as shown in Equation 4-16.

$$\frac{1.137\mathrm{x}10^{7}}{31\ \mathrm{x}\ 24\ \mathrm{x}\ 50000} = 31.5\% \quad \textbf{....... 4-16}$$

Equation 4-15 was applied to the rest of the values to obtain Table 4-4.

Figure 4-6 shows that the monthly values of capacity factor for dry cooled are lower than those for wet cooled plant. However, for both cases peak capacity factor occurs in September, the month with the highest DNI.

Table 4-4 Comparison of Capacity Factors of Dry and Wet Cooling

Month	Capacity Factor (%) (Dry Cooling)	Capacity Factor (%) (Wet Cooling)
January	31.5	39.72
February	29.63	37.22
March	27.5	34.72
April	24.54	30.55
May	25.97	32.22
June	29.72	36.94
July	32.5	40.55
August	37.5	46.6
September	42.5	52.77
October	33.05	41.38
November	25.61	32.2
December	29.16	36.4

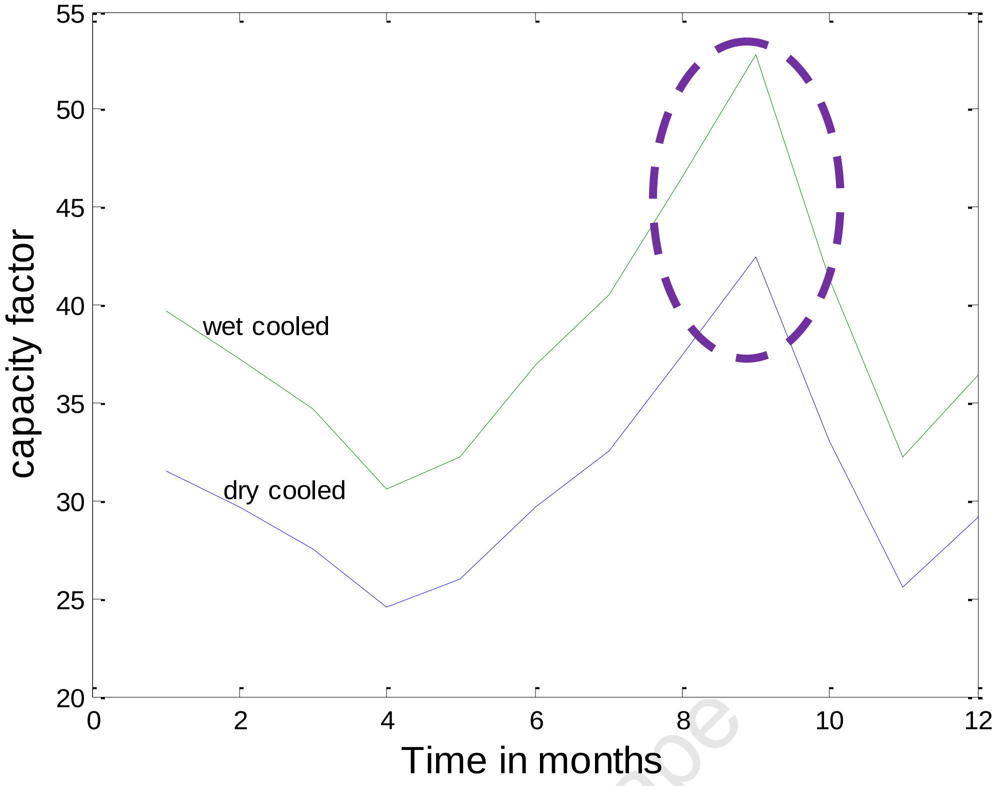

Figure 4-6 Capacity Factor of Dry and Wet Cooled Plants

The findings regarding the energy collected, efficiency of energy collection, capacity factor for both dry and wet cooled plants are summarised in the following section.

4.2.5 Summary of findings on Energy collected, efficiency of energy collection and capacity factor of dry and wet cooled parabolic trough plants

The total energy produced from a dry cooled plant is slightly less than in a wet cooled plant. This is associated with insufficient cooling of the turbine exhaust steam and usually manifested in the steam turbine back pressure. Steam turbine back pressure means that all available thermal energy at the inlet has not been used to generate energy. The cooling fans in a dry cooled plant reported in literature review are responsible for this energy reduction as compared to the wet cooling. The other major contributor towards decreased energy production in dry cooled plants is the dry bulb temperature which increases in different periods of the year. This decreases the difference between the dry bulb temperature and the cooling air on the surface of the condenser. The fans therefore draw more energy from the plant to cool the system.

The heat collection efficiency is a function of absorbed heat by the receivers and the incident energy on the collectors. It increases with decrease in temperature of the cooling media. For dry cooling the fans draws minimal energy from the plant to cool the condenser when the ambient temperatures were low. The amount of energy used for pumping water to the condenser for a wet cooled parabolic trough CSTP is low when the temperature of the coolant (water) is low.

In both dry and wet cooling when the coolant is at lower temperatures the efficiency of heat collection is improved.

Taking wet cooling as the reference point, it is seen that applying dry cooling to a parabolic CSTP plant of the same capacity and load would reduce the net electricity production by approximately 5.42% as shown below. Equation 4-17 is used in this calculation.

$$\frac{Wet\ cooling\ energy\ output - Dry\ cooling\ energy\ output}{Wet\ cooling\ energy\ output} x100 \qquad \textbf{....... 4-17}$$

$$= \frac{135{,}447{,}558 - 128{,}094{,}592}{135{,}447{,}558} x100 = 5.42\%$$

Trieb et al [21] in their studies reported that the wet cooled parabolic trough CSTP plant has an annual solar electric efficiency of about 15%. This is closely related to 15.58% obtained in this research work as was calculated on section 4.2.2. The dry cooled plant achieves a solar electric efficiency of 13.66% compared to the 12% obtained in reference [21]. The slight difference in these results could be attributed to the difference in geographical locations and type of data used. Trieb et al used DNI of 2000kWh/m^2/yr, 6 hours of TES and SM of 2 on a 50MWe plant, while in this thesis a DNI of 1836 kWh/m^2/yr, 12 hours of TES, 50MWe capacity and SM of 2 were used. The efficiency values obtained in this thesis were higher because TES hours are more by half. This increases the plant operation hours.

The percentage annual electric efficiency is higher for the wet cooled plant because air has a lower capacity for heat carriage than water. This therefore inhibits efficient heat rejection on the condenser surface in the case of a dry cooled plant making it less efficient as compared to wet cooling.

The main factor that lowers the capacity factor of a dry cooled plant is the solar field availability. The solar field availability accounts for the percentage of the solar field tracking hours. Dry cooled plants are affected by high temperatures because the difference between the initial temperature difference and ambient temperature is lowered and hence the fans draw more energy from the plant to remove the heat on the surface of condensers. This reduces the total net hourly energy production thereby reducing the capacity factor of the plant. The massive pipes used to remove heat on the surface of the condensers also consume part of the electricity generated for dry cooling [23]. The pipes are metallic in nature which loose energy inside the cooling tower through conduction. This prompts the fans to draw more energy from the plant for cooling.

Water consumption is a big issue for parabolic trough CSTP plant deployment. The following section discusses the amount of water consumption for a dry and wet cooled parabolic trough CSTP plant.

4.3 Water consumption: Comparison between dry and wet cooling for a 50MWe Plant

Table 4-5 shows the volume of water consumption of dry and wet cooled parabolic trough CSTP plants. Water consumption is found to be the highest in September. In dry cooled plant, water is used for mirror washing and steam generation while in a wet cooled plant water is used for steam production, mirror washing and cooling. The total annual water used for wet cooled and dry cooled plants of 50MWe are 718,633 m^3 and 40,902 m^3 respectively.

Table 4-5 Water Consumption between Dry and Wet Cooling

Month	Dry cooling Water consumption (m^3)	Wet Cooling Water Consumption (m^3)
January	1,578	59,341
February	1,478	55,765
March	1,399	53,262
April	1,243	47,542
May	1,303	49,667
June	1,466	55,465
July	1,609	60,576
August	1,841	69,064
September	2,080	78,071
October	1,645	62,371
November	1,309	49,965
December	1,450	55,041
Totals	**40,902m^3**	**718,633m^3**

The following section summarises the findings of the water usage in dry and wet cooled parabolic trough CSTP plants.

4.3.1 Summary of findings of water usage in dry and wet cooled parabolic trough plants

About 2% of the total water used in wet cooled plants is used for washing mirrors and steam production [101]. Therefore a total of 704,260 m^3 was used for cooling. Dry cooled plants use 3.1% of the water consumed by wet cooled plants which saves up to 97% of the volume of water consumed.

It is reported that changing from wet to dry cooling reduces water usage by more than 93% in all thermoelectric plants [101]. In both dry and wet cooling, the amount of water consumed is highest in September for Lowdar. The reason is that since more DNI is received in September in Lowdar, more water needs to be used in cooling and steam production. The amount of money spent on cooling of a 50MWe CSTP plant is estimated to be \$14.8 per 1000 gallons [94]. This value is used to estimate the amount of money spent on cooling as listed in Table 4-6. The conversion used from gallons to cubic metre of water is given by Equation 4-18[73].

$$1\,m^3 = 264\,gallons \quad \text{....... 4-18}$$

Table 4-6 Cost of Water for Dry and Wet Cooling Methods

Cooling type	Amount of Money spent on Water (\$) per year
Wet Cooling	$\frac{718{,}633x14.8x264}{1000} = 2{,}807{,}842$
Dry Cooling	$\frac{40{,}902x14.8x264}{1000} = 159{,}812$

The amount spent on water consumption per MWh of electricity generated is calculated using Equation 4-19.

$$\$/\text{MWh} = \frac{\text{Amount Spent per year(\$)}}{\text{Annual net Energy generated per year (MWh)}} \quad \text{....... 4-19}$$

Table 4-7 lists the money spent in US\$ for each MWh energy generated for dry and wet cooling parabolic trough plants.

Table 4-7Amount Spent per MWh Energy Generation

Cooling type	Amount of money spent per MWh generated annually (\$/MWh)
Wet Cooling	$\frac{2{,}807{,}842}{135{,}447.558} = 20.73$
Dry Cooling	$\frac{159{,}812}{128{,}094.592} = 1.25$

The amount of money spent on water in wet cooling is 16.62 times that used in dry cooling. The cost of water used for wet cooling per year shown in Table 4-6 and 4-7 could rise to the point where the cost of energy for a plant with wet cooling equals the cost of energy from a plant with dry cooling.

The water usage per MWh electricity generated annually (water intensity) is calculated using Equation 4-20.

$$\text{Water used per MWh/year} = \frac{\text{Annual Water usage}(m^3)}{\text{Annual net Energy generated}(MWh)} \quad \textbf{....... 4-20}$$

Table 4-8 shows the amount of water consumed per MWh electricity generated annually for dry and wet cooled parabolic trough plants.

Table 4-8 Water Consumption per MWh Energy Generation

Cooling type	**Amount of water used per MWh generated annually (m^3/MWh)**
Wet Cooling	$\frac{718,633}{135,447.558} = 5.31$
Dry Cooling	$\frac{40,902}{128,094.592} = 0.319$

It is reported that the average water consumption for dry and wet plants is 0.3m^3/MWh and 3.5m^3/MWh respectively [101]. Other wet cooled steam Rankine cycle such as those in coal and nuclear power plants consume about 2.2m^3/MWh and 3.2m^3/MWh respectively.

Compared to these plants, water consumption for the wet cooled parabolic trough plant is higher because of the lower steam Rankine cycle efficiency (37%) and the intermittent nature of the sun which forces the plant to frequently start and stop. The steam Rankine cycle efficiency of coal and nuclear plants is about 42% [116]. In this case dry cooling uses the least amount of water per MWh of generated electricity. The higher amount of water usage for the wet cooled plant occurs because of three major losses that occur in the cooling tower i.e. evaporation, blow down and drift.

Water loss through evaporation is the largest. The wet cooled plant utilises the wet bulb temperature for cooling the condenser. The cooling water is used to eject heat of the incoming exhaust heat from the turbine by absorbing it (latent heat of evaporation). The temperature of the cooling water rises and is lost through evaporation. The water is continuously re-circulated in the cooling tower and hence some more water must be added to make up the portion lost through evaporation. Concentration of the dissolved minerals in the water increases when water is evaporated without replenishing. The amount of water drained from the cooling ponds to avoid the minerals build up is called "blow down water". Water loss also occurs through drift, i.e. some water droplets are carried out by air before condensation occurs [101].

Parabolic trough CSTP plants need large portion of land for thermal energy collection. The following section compares the land usage for dry and wet cooled parabolic trough CSTP plants.

4.4 Land usage: Comparison between Dry and Wet Cooled Parabolic Trough

Land use refers to the total area directly occupied by the plant.

Table 4-9 shows the land usage for both dry and wet cooling of the solar parabolic plant. Wet cooled plant uses a much larger area when SM is used for sizing the field. Solar multiple is the ratio of the of the energy supply by the solar field to the turbine design heat input as described in section 4-8, Equation 4-30. This allows the solar field to oversize or undersize itself depending on the amount of DNI at a particular hour of the day to match the load on the turbine. Over-sizing the solar field means that all the collectors are tracking the sun i.e no collector is being defocused.

Under-sizing the solar field means that some collectors have been defocused because the amount of energy collected exceeds the turbine design thermal input. As was previously shown in section 4.2.3 the wet cooled plant has higher heat collection efficiency than a dry cooled plant. Thus for the same capacity size of 50MWe, wet cooling is more flexible to increase its area of heat collection because the turbine is able to convert the heat to electricity at a much faster rate than dry cooling. Also as reported by [101] dry cooled plants use less land per MWh of energy generated than wet cooling. The reason behind this is that unlike dry cooling the area covered by the water surface for wet cooling add to the total land occupied.

Table 4-9 Land Usage for wet and dry cooling

Land Usage for wet and dry cooling Type of cooling	Dry cooled	Wet cooled
Land area in m^2	1.56 X10^6	1.79 X10^6

4.4.1 Land use factor

Land use in CSTP plants refers to the land directly occupied by the power plant structures. The exact land usage for the parabolic trough considered in this research was 427,280m^2. The land use factor is defined shown in Equation 4-21.

$$land\ use\ factor = \frac{area\,(m^2)}{MWh/year} \text{ .... 4-21}$$

The land use factor for the dry cooled plant and wet cooled plants were calculated as shown in Equations 4-22 and 4-23 respectively. The dry cooled plant uses a slightly larger area than the wet cooled plant to generate 1MWh. The land use factor is lower for both dry and wet cooled plants in Lodwar compared to 1.1m^2/MWh reported in Spain for a 50MWe. This is mainly because of the high DNI in Spain of about 2136kWh/m^2which results into high energy production per square area [21]. The land use factor is 5.41% higher in dry cooling than in wet cooling.

$$land\ use\ factor, dry = \frac{427{,}280}{128{,}094.592} = 3.33m^2/MWh/year \text{ .... 4-22}$$

$$land\ use\ factor, wet = \frac{427{,}280}{135{,}447.558} = 3.15m^2/MWh/yr \text{ .... 4-23}$$

The following section describes the impacts of TES on LCOE and capacity factor of dry and wet cooled plants.

4.5 Impact of TES size on Capacity factor and LCOE

This section provides an analysis of energy production of TES and its effects on the LCOE and capacity factor. The addition of TES to a parabolic trough plant has some advantages.

Firstly, unlike plants without storage that must sell electricity whenever it's available, a CSTP plant with TES has the ability of shifting energy production from periods of low demand to periods of high demand. Most of the times the periods with high demand correspond with the highest electricity tariffs.

Secondly addition of TES replaces the use of conventional fossil fuels used for heating the HTF to maintain stable steam production at the turbine for constant electricity generation. These fossil fuels are a source of greenhouse gases when combusted.
Thirdly, addition of TES provides spinning reserves. Spinning reserve is the amount of energy that is always available and accessible to the system and can be activated on request by the system operator to run the turbine. It can also be defined as an additional generating capacity that provides power quickly, say in the span of 30 minutes after request by system operator.

The storage capacity of TES is the number of hours the hot storage tank can be charged to its maximum thermal capacity. In this thesis 12 hours was used. It has been assumed that the number of hours without the sun was 12 and that the energy available in TES after sunset was able to meet the load demand. This is similar for discharging in the assumption that the tank is 98.5% efficient [78].

The size of TES has an effect on land size. The more the number of hours of TES the bigger is the size of the storage system and so is it occupies a larger space than plants with small size of TES. The cost of energy for plants with higher storage capacities is higher than plants with lower capacity sizes. This is because the land occupied and the extra capital to hold larger capacities of TES has value in them. Therefore for a given plant capacity the optimisation to determine the minimum number of hours to run the turbine at design heat intake is paramount. At higher solar field area with lower TES size the collectors defocus more often. This is because the TES size reaches its maximum charge rate faster because of the huge thermal collection area. This therefore under-utilises the collector field. The correct TES size should be able to provide energy when the sun is not there and also have the capability to store excess energy generated by the collector field.

The following section discusses the results of the impacts TES has on LCOE and capacity factor of a wet cooled parabolic trough CSTP plant.

4.5.1 Wet cooled plant

Table 4-10 and Figure 4-7 show the capacity factor and LCOE values calculated for the wet cooled parabolic trough plant for different TES hour values. It is observed that for a wet cooled plant both the LCOE and the capacity factor increase as the number of TES hours increase. The reason for this finding for LCOE is that both land requirement and volume of TES increase with increase in TES hours and hence this raises the costs associated to them. It is seen that LCOE increases by 10.65% from the minimum LCOE at TES=3 to maximum at TES=12.

The lowest LCOE for the wet cooled plant is found to be $79.5493cent/kWh which occur at TES=3. At 0 hours the wet cooled plant records the highest LCOE and the least capacity factor. With no energy storage the plant operates only when irradiation is sufficient to run the turbine. This therefore decreases the amount of energy generated by the plant which leads to an increase in LCOE as shown in Equation 4-24. The capacity factor decreases because of the low energy generated [72].

$$LCOE = \frac{\text{a x Annualized capital costs} + \text{Annual O \& M costs}}{\text{Annual Energy generated x plant availability}} \quad \textbf{....... 4-24}$$

Where a is the annuity factor defined by equation 4-25.

$$a = \frac{K_d(1+K_d)^n}{(1+K_d)-1} + K_{insurance} \quad \textbf{....... 4-25}$$

K_d : Interest rate= 19.9% as per Kenya 2011, shown in Table 3-25, section 3.21.4

$K_{insurance}$: Annual plant insurance=0.5%, shown in Table 3-25, section 3.21.4

n : Depreciation period in years: This simulation assumed a 30 years plant life time, shown in Table 3-25, section 3.21.4

O &M: operation and maintenance costs.

Table 4-10 Capacity Factor and LCOE for a Wet cooled plant

Parameterized Input(s)	LCOE($cents/kWh)	Capacity Factor (%)
Full load hours of TES=0	88.3585	26.5498
Full load hours of TES=3	79.5493	31.0496
Full load hours of TES=6	81.614	31.497
Full load hours of TES=9	84.7999	31.4677
Full load hours of TES=12	88.0218	31.427

Capacity factor increases sharply from 0 hours of TES to about 3 hours of TES after which the increase is gradual. It starts to level off from 6 hours of TES. As was previously defined in section 4.2.4, Equation 4-15, the capacity factor increases with the increase of energy generated. Addition of TES to a parabolic CSTP plant stores extra thermal energy collected in the field to be dispatched later when the sun is not there or in case of low irradiance hours. This increases the total amount of energy generated by the plant and therefore the capacity factor increases. At lower hours of TES the amount stored is less hence the actual energy generated is low. This leads to lower capacity factor.

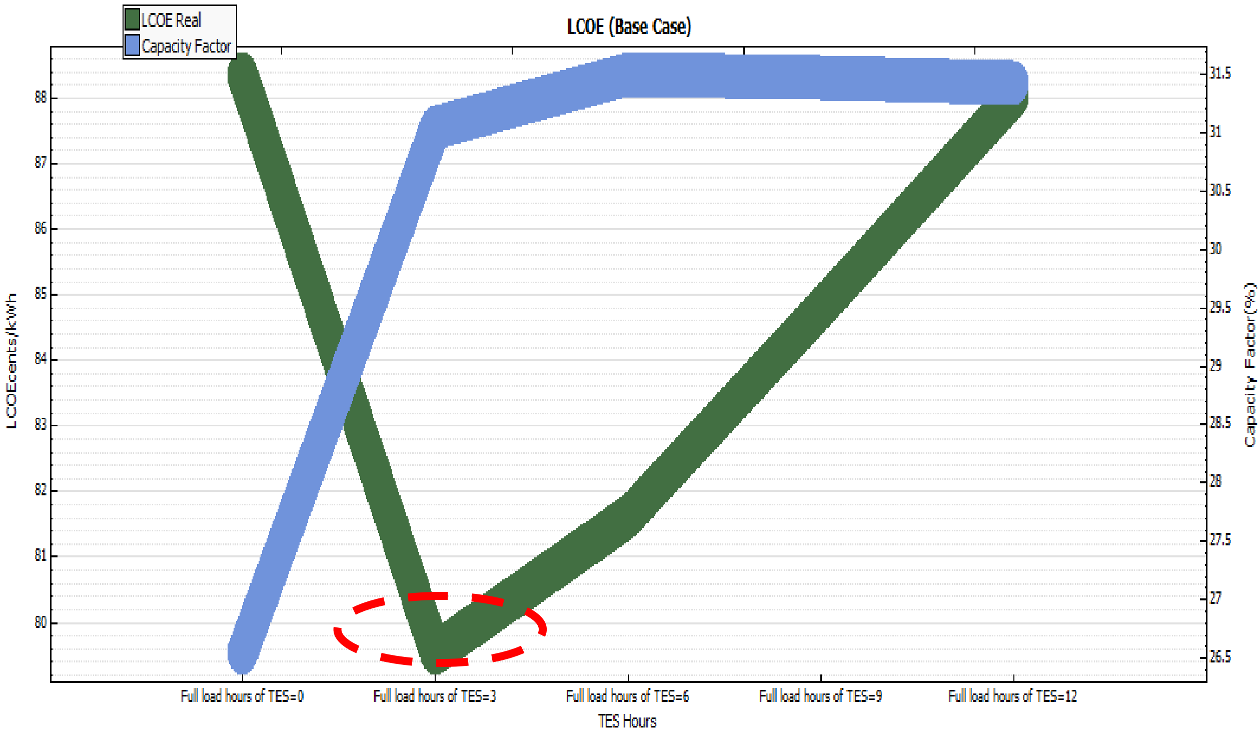

Figure 4-7 Impacts of Capacity Factor and LCOE with Increasing Hours of TES for Wet cooled Plant

The following section discusses the impacts TES has on capacity factor and LCOE of a dry cooled parabolic trough CSTP plant.

4.5.2 Dry cooled plant

Table 4-11 and Figure 4-8 shows the variation of capacity factor and LCOE with the increase in the hours of TES. The increase in LCOE from the minimum at TES=3 to maximum at TES=12 is 12.5%. The optimal LCOE value for the dry cooled plant is found to be $86.0814cent/kWh which occurs at TES=3. The investment cost of a dry cooled plant is 5% higher to cater for the complexity of the dry cooling system [90]. The air cooled condenser (ACC) of a dry cooled plant is more capital intensive than that of the wet cooled plant. As described by the definition of LCOE on section 4-5-1, Equation 4-24, increasing the capital costs of a plant will lead to an increase in the LCOE. On the other hand, the capacity factor of the dry cooled plant increases as TES increases. The reason is that at higher TES hours the amount of energy stored is higher hence increasing the actual amount of energy generated by the plant.

Table 4-11 Optimized Capacity Factor and LCOE for a Dry Cooled Plant

Parameterized Input(s)	LCOE ($cents/kWh)	Capacity Factor (%)
Full load hours of TES=0	94.9628	25.1526
Full load hours of TES=3	86.0814	29.3845
Full load hours of TES=6	88.8389	29.8106
Full load hours of TES=9	92.8191	29.7849
Full load hours of TES=12	96.8523	29.7441

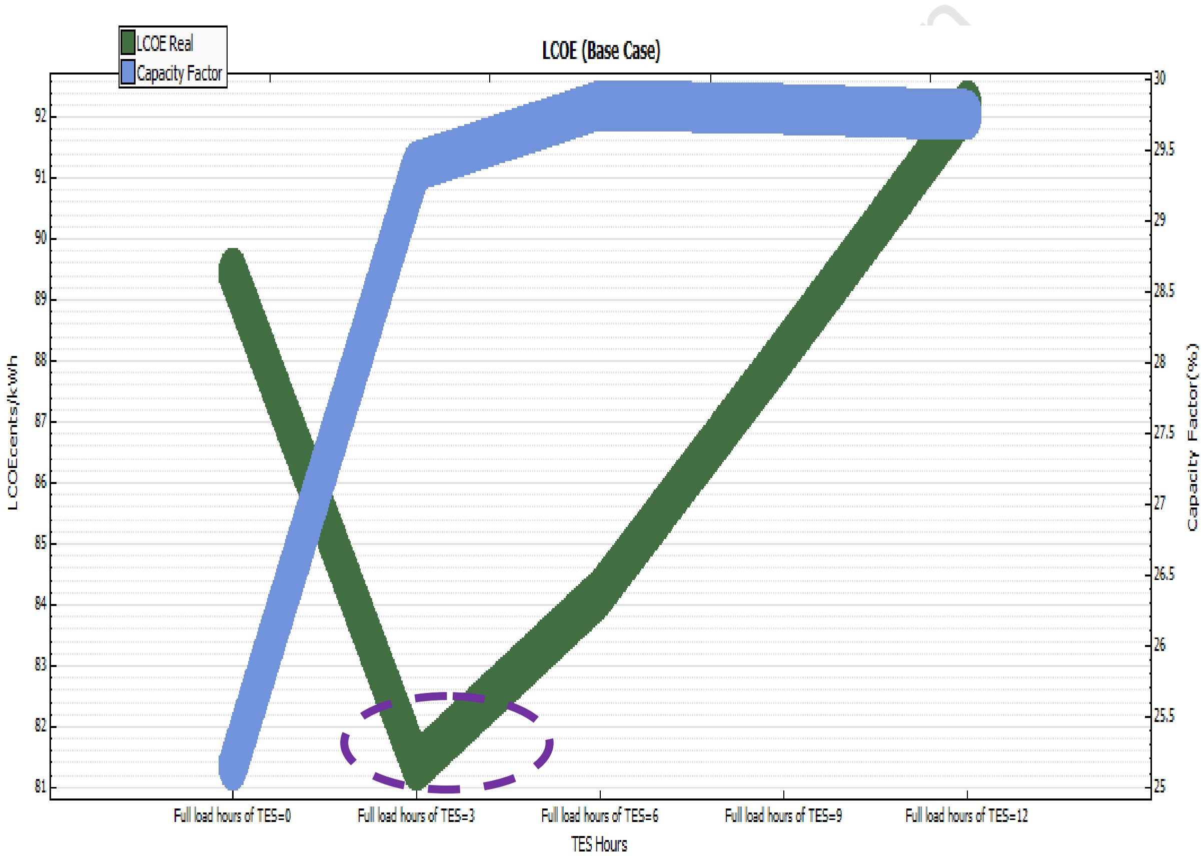

Figure 4-8 Impacts of Capacity Factor and LCOE with Increasing Hours of TES for Dry cooled Plant

As shown in Table 4-12 the LCOE of a dry cooled plant is higher than that of wet cooled plant for each hour each hour of TES considered. The increase is highest at 12 hours of TES (9.12%) and lowest at 0 hours of TES (6.95%). The reason behind this is that as the number of TES hours increases the physical size of the TES tank structures increases which attracts a higher cost.

The capacity factor of a wet cooled plant of a dry cooled plant is lower than that of the wet cooled plant. This is because of the low energy production associated with dry cooled plants. The lower energy production is owed to the fact that dry cooling uses air for cooling the exhaust steam from the turbine which is less efficient as was described in section 4.2.5.

Table 4-12 Comparison of Dry and wet cooled plant in terms of LCOE and Capacity Factor

Parameterized Input(s)	% increase of LCOE of dry cooled plant	% decrease in Capacity Factor of dry cooled plant
Full load hours of TES=0	6.95	5.26
Full load hours of TES=3	7.59	5.36
Full load hours of TES=6	8.13	5.35
Full load hours of TES=9	8.64	5.34
Full load hours of TES=12	9.12	5.35

The findings of the impacts of TES on capacity factor and LCOE for dry and wet cooling parabolic trough plant are summarised and compared in the following section.

4.5.3 Summary of findings of impacts of TES on LCOE and capacity factor on wet and dry cooled parabolic trough plants

As shown in Figure 4-9, inclusion of TES in both dry cooled and wet cooled plants increases the cost of TES. This increase results from the inherent design of parabolic trough plant with molten salt-based TES which implies that adding TES increases the size and inventory of the salt tanks. Adding TES increases the field size required for thermal energy collection. This is because the excess thermal energy collected but unused in the power block can be stored for later energy generation when demand rises or when solar irradiation is not there as discussed in section 4-7.

In both dry and wet cooling, as the capacity factor of the plant increases the LCOE decreases from 0 hours of TES to the optimized value at 3 hours of TES after which it starts rising again. At the maximum number of TES hours (12) for both wet cooled and dry cooled plants the LCOE of a dry cooled plant is higher by 9.12% as calculated in Equation 4-26.

$$\frac{96.8523-88.0218}{96.8523}X100=9.12\% \text{ .... } \textbf{4-26}$$

At 0 hours of TES for both dry and wet cooled plants they record the highest LCOE and the least capacity factor. The high LCOE is brought about by the lower energy generation without support generation from TES. This reduces the plant availability (the number of hours the plant is operating expressed as a percentage of hours in a year (8760) and hence the LCOE reduces as shown in section 4.5.1, Equation 4.24. The capacity factor at 0 hours of TES is low because the plant does not have any back up power. This reduces the total amount of energy generated and hence the capacity factor reduces. This is explained in section 4.2.4, Equation 4-15.

In Mojave desert California, USA it is reported that the dry cooled plants have higher LCOE than wet cooled plant by 5% [90].The difference in the results could be brought about by the difference in geographical location of Lowdar and California and by the different DNI patterns in these two places. The DNI of Lowdar is about 1836kWh/m^2/yr and that of California is 1965.2kWh/m^2/yr. The difference could also be attributed to the difference in the level of inflation rate in the two regions which is about 2.5% in USA and 16% in Kenya.

Parasitic losses are lower in plants with TES because of the higher annual generation and lower percentage of offline parasitic electric consumption. The electric parasitic losses refer to the amount of energy consumed by the plant itself to generate electricity. This includes amount of energy drawn by the fans and pumps for cooling, energy drawn by motors for defocusing the collectors, energy used for starting up the plant, energy consumed for TES pumping and the solar field HTF pumps used for pumping HTF to and from collector field.

These types of energy losses are minimised by TES incorporation. For example when there is enough energy storage the excess heat collected by the solar field is stored, hence the plant controller does not turn the motors on to defocus some collectors. This saves energy. One of the advantages of TES that was mentioned in section 4.5 is the provision of spinning reserves which ensures that there is enough energy to run the turbine at all times without drawing the generated energy from the plant. This saves time because the plant operator does not always have to wait until the solar irradiance generates enough energy to run the turbine.

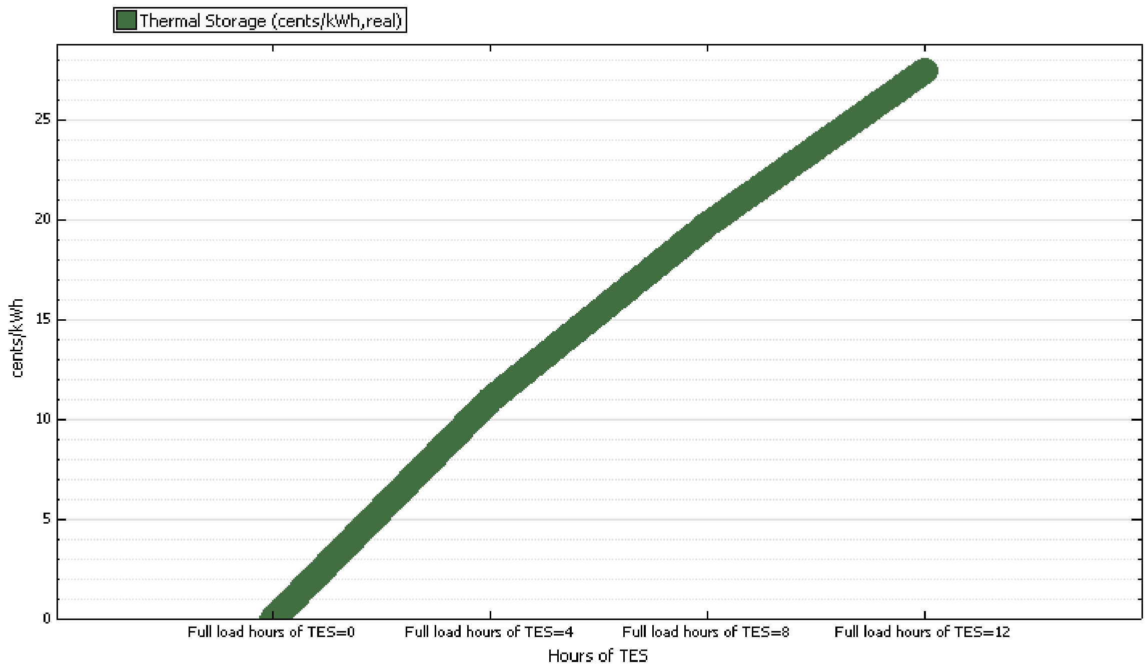

Figure 4-9 Rise in Cost of Thermal storage as hours of TES increases

The following section discusses the value of TES addition to both dry and wet cooling.

4.5.4 Addition of value by TES through Dispatch ability:

The energy value of TES for both dry and wet cooled plants is derived from two sources. These are:

4.5.4.1 Resizing the collector field

If the amount of energy collected during the day by the solar field at high solar hours is more than the turbine design gross input, TES stores the extra energy for later use.

This therefore enables the plant to use an oversized solar field because the excess heat collected can be stored in the TES tank which is unlikely for plants without storage. Solar field area is specified by a solar multiple of 2 in this thesis. SM re-sizes the field according to the amount of DNI at a particular time. If the amount of energy collected is more than the turbine design load the excess energy is stored in TES. If TES is full the solar field then resizes by defocusing some collectors.

4.5.4.2 Generation shifting capability

The other value of TES is that it stores the thermal energy generated during solar radiation peak hours for later use when the solar radiation is unavailable [78]. In other words TES acts like a storage battery which can be used to meet energy demand at any specific time the owner wants. In most cases household energy demands are highest in the morning or in the evening when the sun is not there. Periods of high energy demand corresponds with higher energy prices.

4.5.5 Effects of TES size on LCOE and Energy Dumping

As was previously mentioned in section 4.5.4.1, collector defocusing occurs when solar field delivers thermal energy more than the turbine and thermal storage system can accommodate. The amount of incident energy varies from one collector to the other depending on the inclination of the sun at a particular time and also depending on the hour of the day as the shadow of some collectors falls on other collectors reducing the amount of incident energy.

Sequential defocusing is implemented in this thesis whereby collectors receiving the highest incident energy defocus first. The amount of energy lost through collector defocusing is called dumping. Mathematically SAM estimates the dumped energy by multiplying the incident energy falling on the defocused collectors with the defocusing factor. Defocusing factor varies from 0 to 1.

A 0 means the collector is fully focussed on the absorbers while 1 means the collector is totally defocused. The values between 0 and 1 imply the collectors are partially defocused. A partially defocused collector does not receive the whole incident energy. Table 4-13 and Table 4-14 show the variation of energy prices (LCOE) as generated by SAM at different TES hours and the respective energy that goes to dumping on wet and dry cooled parabolic troughs respectively. The amount of energy dumped in every hour of TES is higher for dry cooling than wet cooling. The reason is that the dry cooled plant is inefficient in heat energy collection and hence the collectors defocus whenever the generated thermal energy exceeds the turbine design thermal input and maximum charge rate of TES.

Figure 4-10 and Figure 4-11 show that as the hours of TES increases, the dumped energy decays while the LCOE rises. Physically as the number of TES hours increases the size of energy storage and hence holds more energy. Addition of TES hours therefore has the benefit of reducing dumped energy by storing it. Addition of TES from 0 hours to 12 hours of TES reduces the amount of energy dumped by 99.9% for both wet and dry cooling CSTP plants. The amount of energy that goes to dump at 12 hours of TES is higher for the dry cooled plant than the wet cooled plant by 9.63%.

The dumping of energy reduces the total energy collected by the solar field and hence the total energy generated by a CSTP plant. From the definition of LCOE (section 4.5.1, Equation 4-24), the lower the energy generated the higher is the LCOE. Since the dumped energy in dry cooling is higher, the LCOE is also higher for each hour of TES as compared to wet cooling. The LCOE at 12 hours of TES for dry cooled plant is slightly higher than the wet cooled plant by 9.17%.

The other reason for LCOE increase as dumping energy decays is that at higher TES hours more inventory costs are incurred in acquiring extra land to accommodate large storage structures and building them.

Table 4-13 Dumped Energy versus LCOE for a wet Cooled Parabolic Trough

Parameterized Input(s)	Q Dumped, Annual (MWh)	LCOE Real($cents/kWh)
Full load hours of TES=0	65,700.5	84.4147
Full load hours of TES=1	32,644.5	78.0735
Full load hours of TES=2	16,686.6	76.6221
Full load hours of TES=3	7,668.88	76.34
Full load hours of TES=4	2,994.46	76.7397
Full load hours of TES=5	1,213.24	77.5625
Full load hours of TES=6	358.511	78.5074
Full load hours of TES=7	104.657	79.5729
Full load hours of TES=8	66.531	80.6759
Full load hours of TES=9	66.0834	81.7746
Full load hours of TES=10	57.8108	82.8794
Full load hours of TES=11	35.7797	83.9839
Full load hours of TES=12	33.841	85.086

Table 4-14 Dumped energy Versus LCOE for a dry Cooled parabolic trough

Parameterized Input(s)	**Q Dumped, Annual(MWh)**	**LCOE Real($cents/kWh)**
Full load hours of TES=0	65,747.1	90.7672
Full load hours of TES=1	32,839.4	84.1657
Full load hours of TES=2	16,901.7	82.7633
Full load hours of TES=3	7,828.11	82.6592
Full load hours of TES=4	3,107.04	83.2557
Full load hours of TES=5	1,286.71	84.3108
Full load hours of TES=6	384.765	85.5195
Full load hours of TES=7	117.582	86.8378
Full load hours of TES=8	78.1841	88.2019
Full load hours of TES=9	78.1698	89.5783
Full load hours of TES=10	68.1952	90.9541
Full load hours of TES=11	59.6399	92.3183
Full load hours of TES=12	37.4448	93.6833

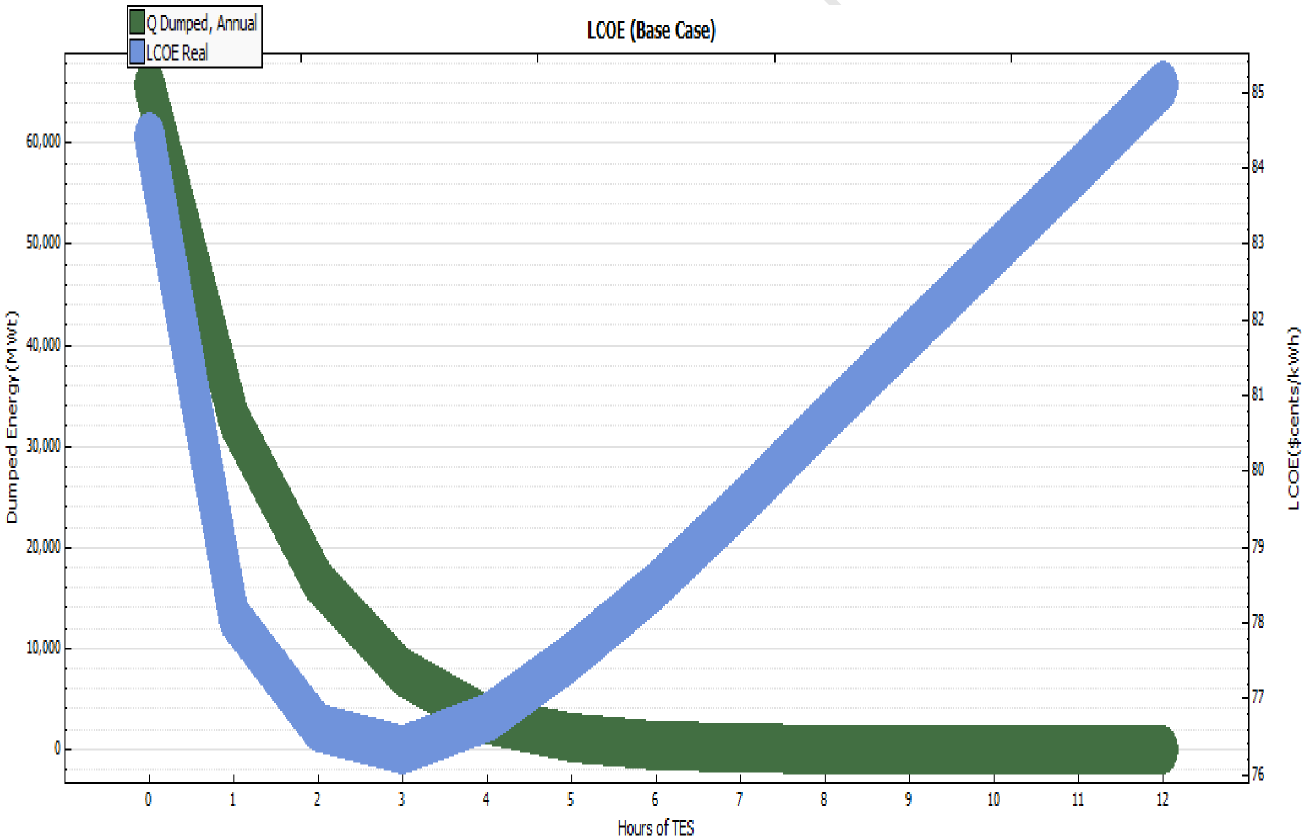

Figure 4-10 Effects of TES size on LCOE and Dumped on Energy (Wet cooled)

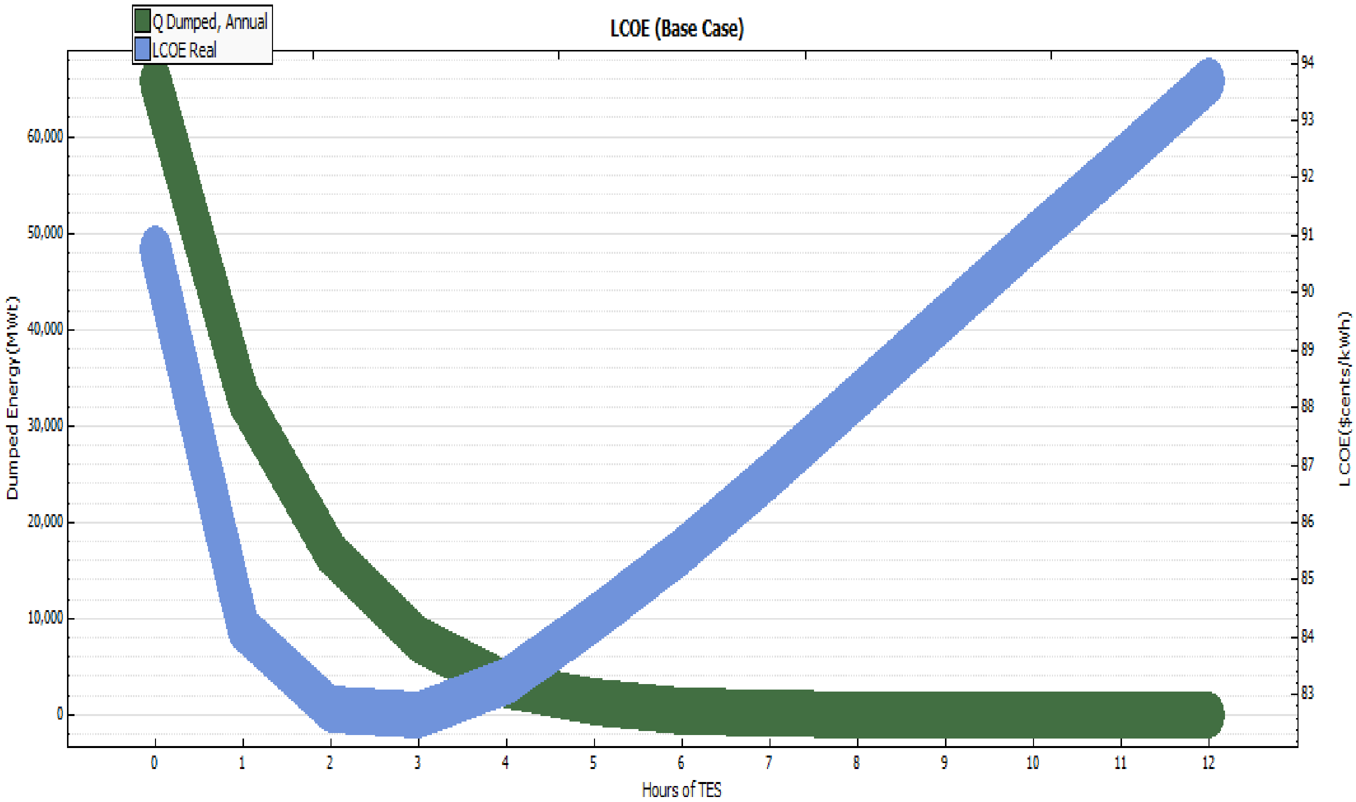

Figure 4-11 Effects of TES on LCOE and Dumped Energy (Dry Cooled)

4.6 TES dispatch scheduling

As was discussed on section 3.14 TES dispatch is the amount of energy drawn from storage to meet demand when the CSTP is not generating enough energy. TES dispatch is divided into two; namely, TES dispatch with solar and TES dispatch without solar. The amount of energy drawn from storage when the solar irradiation is not enough to generate enough energy is called TES dispatch with solar. The amount of energy drawn from storage when the solar irradiation is not available is called TES dispatch with solar. The amount of energy dispatched from storage is a fraction of the demand at that hour and the total energy in storage as described by the Equation 4-27 below. This fraction is called a period.

$$\text{Period} = \frac{\text{hourly energy demand}}{\text{Total Energy stored}} \quad \text{.......} \; 4\text{-}27$$

When the solar radiation is available but unable to meet demand at that hour the period is calculated as shown in Equation 4-28.

$$\text{Period, solar available} = \frac{\text{hourly energy demand - hourly energy generated}}{\text{Total Energy stored}} \quad \textbf{....... 4-28}$$

When there is no generation from the CSTP side the period is calculated as shown in Equation 4-29.

$$\text{Period , solar not available} = \frac{\text{Hourly energy demand}}{\text{Total Energy stored}} \quad \textbf{....... 4-29}$$

The assumption made in Equations 4-27-4-29 is:

- In the application of each of the dispatch periods there is enough energy in TES at that time.

The following section describes energy generation with TES. In this simulation the capacity of the plant considered is 50MWe with 12 hours of TES.

4.7 CSTP Energy Generation with TES Support

In this thesis the energy demand has been divided into six periods as shown in section 3.20.1.1, Table 3-17. The fractional amount of energy dispatched in each period is shown in Table 4-15.

Table 4-15 TES dispatch Periods of Wet and dry cooling in the 1st day of January 2012

Period	Energy demand	Fractional amount of energy dispatched from storage (wet cooling)	Fractional amount of energy dispatched from storage (dry cooling)
1	20,060	0.097	0.098
2	27,880	0.135	0.137
3	1,360	0.006	0.007
4	13,430	0.065	0.066
5	52,190	0.253	0.256
6	35,190	0.17	0.173

The amount of energy stored by TES depends on the DNI and the load on the turbine at any given time. During the night or low irradiance hours the stored energy is dispatched from storage. This is done through a heat exchange of its thermal energy with the HTF, which generates steam and runs the turbine for electricity production. The interaction of HTF and TES is described in section 3.14, Equations 3-16 to 3-19. The amount of energy dispatched depends on the Power block operating mode in the previous hour (charging or discharging), quantity of energy in storage in the current hour and the energy available from the solar field in the current hour.

Table 4-16 shows the hourly amount of energy generated by the wet and dry cooled plants and the hourly energy stored in each hour in the first day of January (1^{st} day of generation). As shown in this table the two plants (dry and wet cooled) starts generating energy at 8am to 6pm. Energy storage starts at 9am to 3pm. The amount of energy generated is able to meet the energy demand till 5 pm. From 6 pm until 11pm the demand of energy is met through TES dispatch because there is no DNI or is not sufficient to support generation. The total amount of energy stored for wet and dry cooled plant in the 1^{st} day of January is 206,586kWh and 203,897kWh respectively. Therefore as was described in section 4-6 the energy stored must be dispatched from storage to meet demand.

The blue values appearing in the table indicate TES dispatch from storage to the power block. The values in brackets at 6pm show the amount of energy dispatched from TES because in the current hour the energy generated by the CSTP is not enough to meet demand. The amount of energy dispatched from storage is the difference between the load demand and the amount of energy generated by the CSTP plant at a particular hour. For example at 6pm the amount of energy dispatched from storage for a wet cooled plant is the difference between the demand at that hour which is 35,190kWh and **22,561.7kWh** which is the amount of energy generated from the CSTP at that particular hour. This therefore means that **12,629kWh,** must be dispatched from storage to meet the energy demand at 6pm.

If zero energy is generated the by the CSTP plant, energy demand is met through TES dispatch from storage. As shown in Figure 4-12 and Figure 4-13 the amount of energy stored for a wet cooled plant is higher than for the dry cooled plant. This is due to the fact that dry cooled plant is less efficient in terms of energy collection and also the electric parasitic losses are higher as was described in section 4.5.3.

Table 4-16 Hourly Energy generated, Hourly DNI, hourly energy stored of wet and dry cooled plants against the hourly load profile (1st day of January)

Hour of the day	Energy generated by wet cooled plant(kWh)	Energy generated by dry cooled plant (kWh)	Energy stored wet cooled plant (kWh)	Energy stored dry cooled plant (kWh)	DNI(Wh/m^2)	Load profile kWh
0000	0	0	0	0	0	20,060
0100	0	0	0	0	0	20,060
0200	0	0	0	0	0	20,060
0300	0	0	0	0	0	20,060
0400	0	0	0	0	0	20,060
0500	0	0	0	0	0	20,060
0600	0	0	0	0	8	27,880
0700	0	0	0	0	246	1,360
0800	22,543.1	23,001.1	0	0	595	1,360
0900	52,129.8	51,770.4	9,493.69	9,493.69	605	1,360
1000	52,172.8	51,299.5	8,893	7,959.98	604	1,360
1100	52,301.6	51,033.8	2,884.63	2,237.41	586	1,360
1200	51,479.4	49,835.8	41,412.2	41,401.9	736	1,360
1300	51,272	49,454.1	54,401.1	53,470.7	775	1,360
1400	51,326.2	49,302	57,717.6	57,588.4	782	1,360
1500	52,053.7	49,940.1	31,784.5	31,745.9	683	1,360
1600	49,419.6	47,976.7	0	0	461	1,360
1700	39,489.3	40,070	0	0	328	1,3430
1800	**22,561.7(12,629)**	**25,042.6(10,148)**	0	0	38	35,190
1900	**52,190**	**52,190**	0	0	0	52,190
2000	**35,190**	**35,190**	0	0	0	35,190
2100	**35,190**	**35,190**	0	0	0	35,190
2200	**35,190**	**35,190**	0	0	0	35,190
2300	**35,190**	**35,190**	0	0	0	35,190
Totals	**486,056**	**478,033**	**206,586.7**	**203,897**	**6,447**	403,410

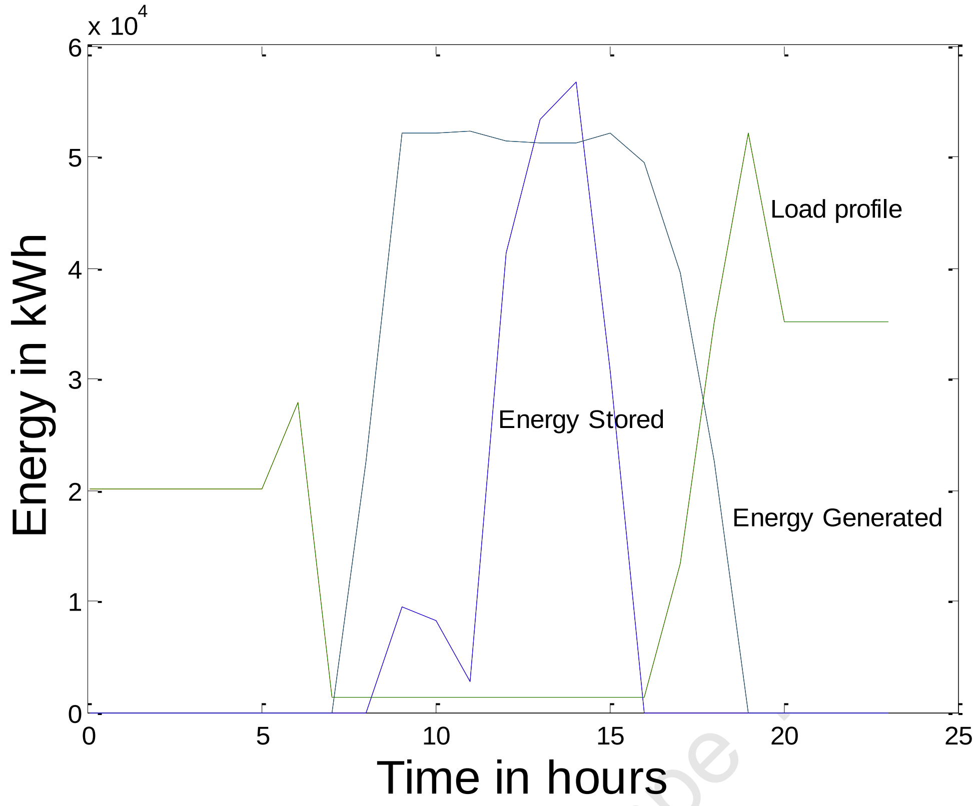

Figure 4-12 Hourly Amount of Energy generated, amount of energy stored and Load profile of a wet cooled plant

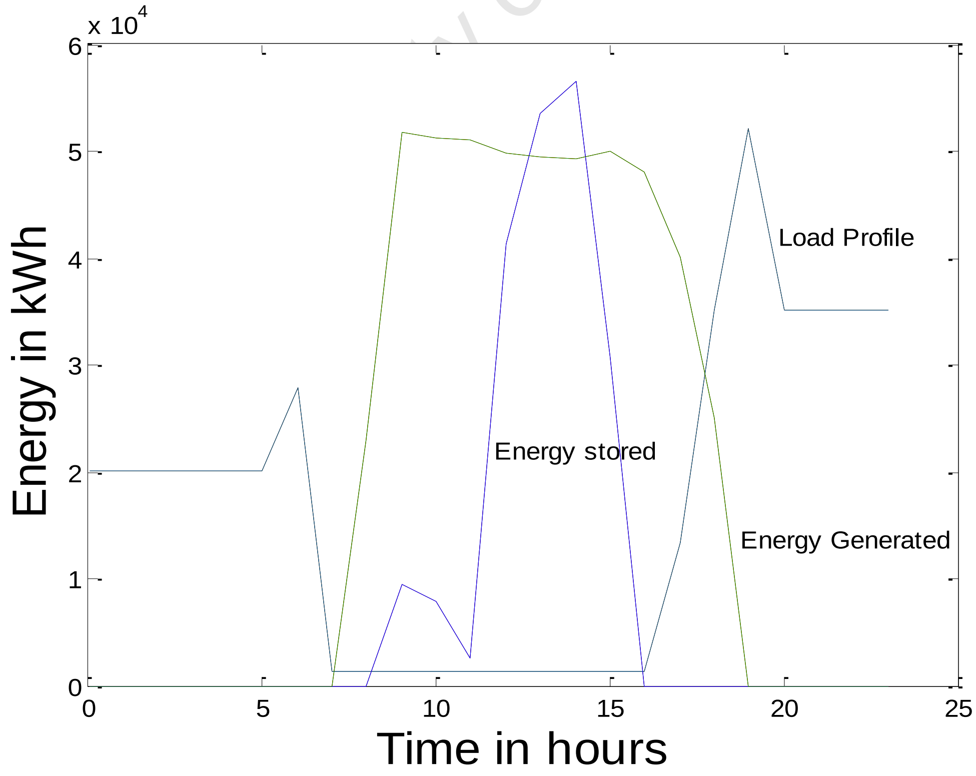

Figure 4-13 Hourly Amount of Energy generated, amount of energy stored and Load profile of a Dry cooled plant in the 1st day of January

As shown in Table 4-16 above the amount of energy generated from the CSTP plant after 6pm is not enough to meet the load demand. The load demand is hence met through TES dispatch. The circled section of Figure 4-14 and Figure 4-15 show the load met through TES dispatch for a non-uniform TES dispatch. Non uniform dispatch is able to fully meet the demand from 6pm up to 11 pm.

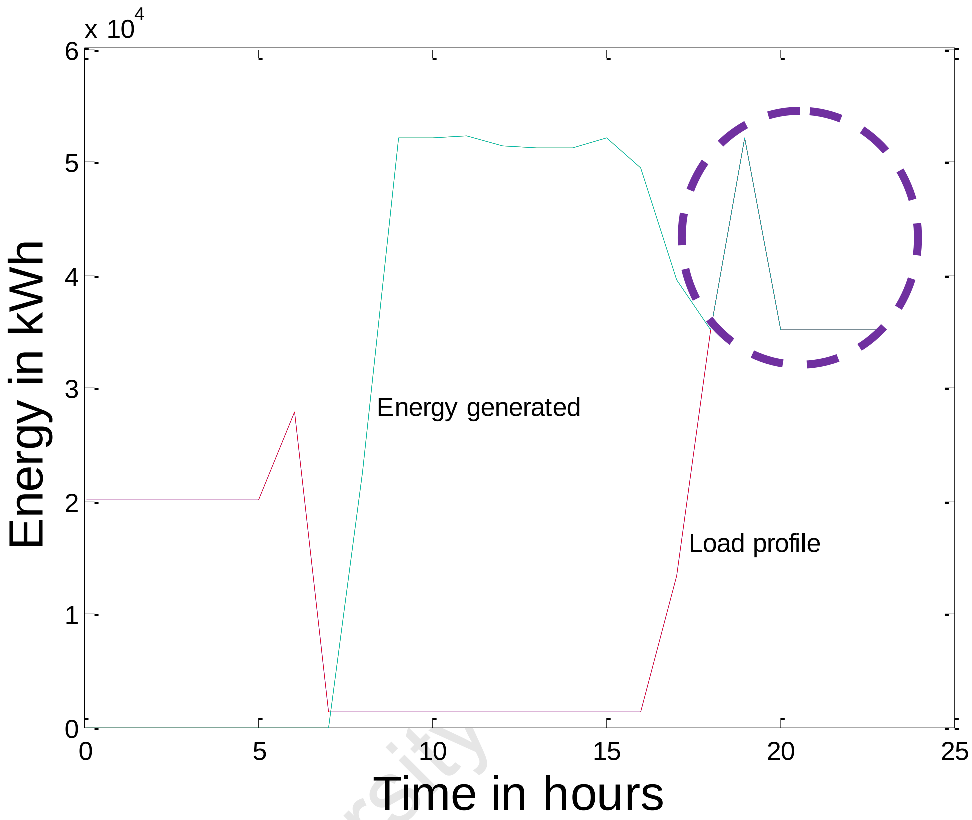

Figure 4-14 Hourly Energy generated with TES support to meet the Load demand (from 6pm to 11pm) for a wet cooled plant

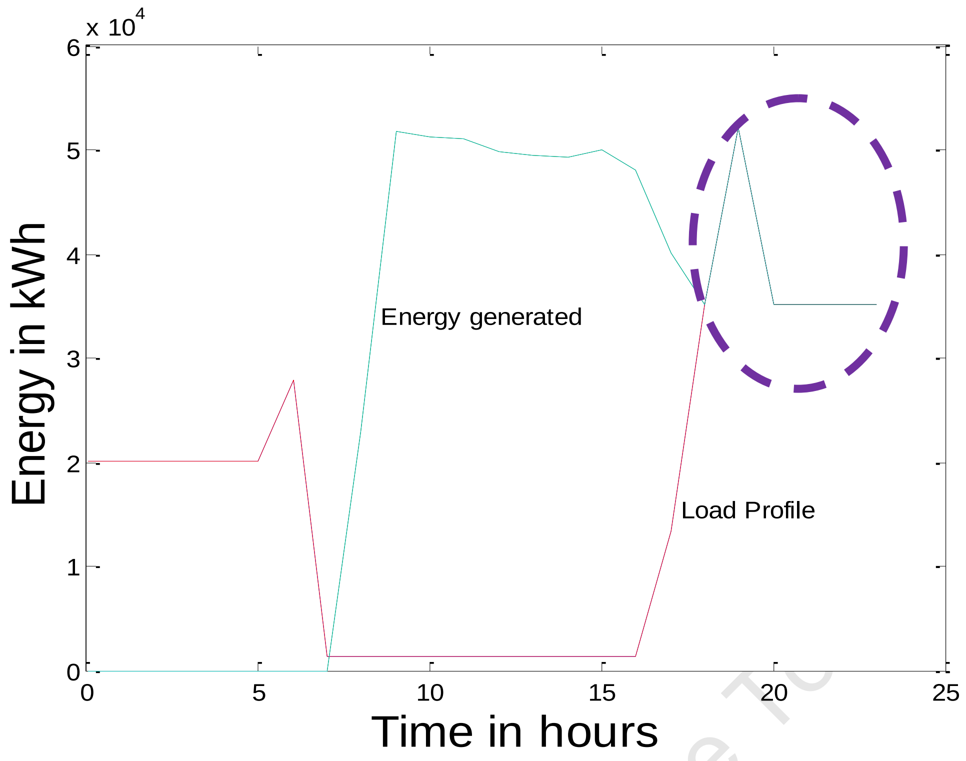

Figure 4-15 Hourly Energy generated with TES support to meet the Load demand (from 6pm to 11pm) for a Dry cooled plant

If uniform TES dispatch is adopted the amount of energy dispatched from storage to meet demand on each hour for wet and dry cooling is 8607.75kWh and 8495kWh respectively. This is done by dividing the total amount of energy in storage by the number of hours in a day. A plot of load profile, energy stored per hour and uniform TES dispatch for both wet and dry cooled plants are shown in Figures 4-16 and 4-17 respectively. As shown in these figures uniform dispatch evenly distributes energy in all hours regardless of whether there is generation from DNI or not. In this case TES dispatch is used as a base load power plant, which is the minimum amount of energy that a plant must make available to its clients to meet demand. The amount of energy uniformly dispatched in wet cooling is slightly higher than for dry cooling by 1.3%. The difference is brought about by the parasitic losses that occur when cooling the plant described in section 4.5.3.

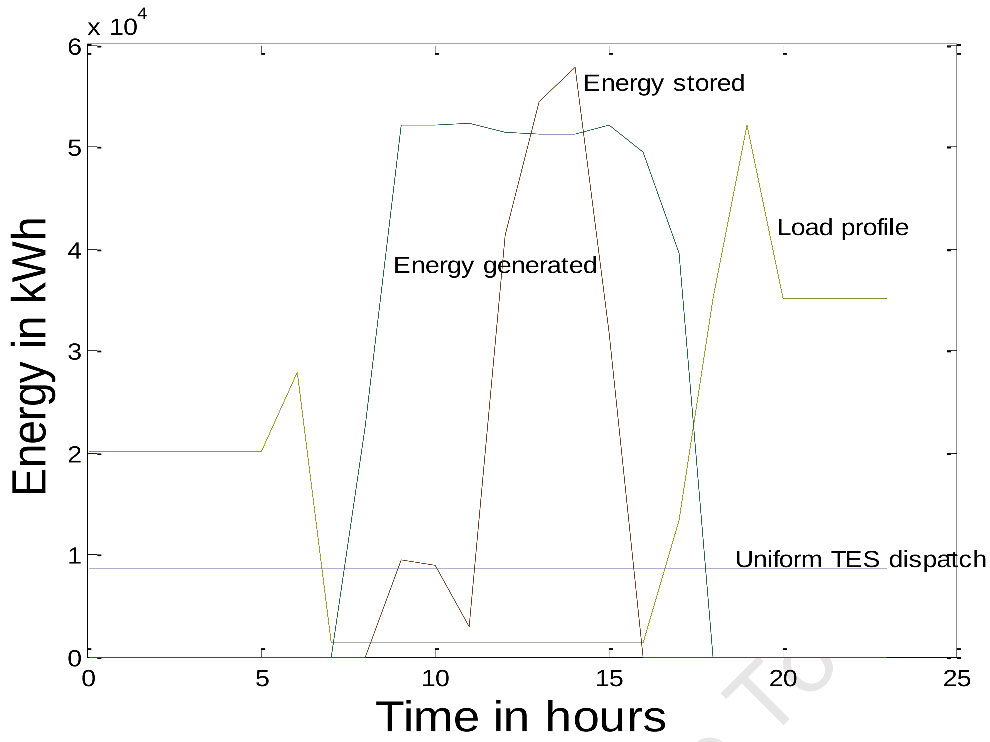

Figure 4-16 Hourly Energy Stored, Hourly Uniform TES Dispatch, Hourly energy generated and Load Profile of a wet cooled plant

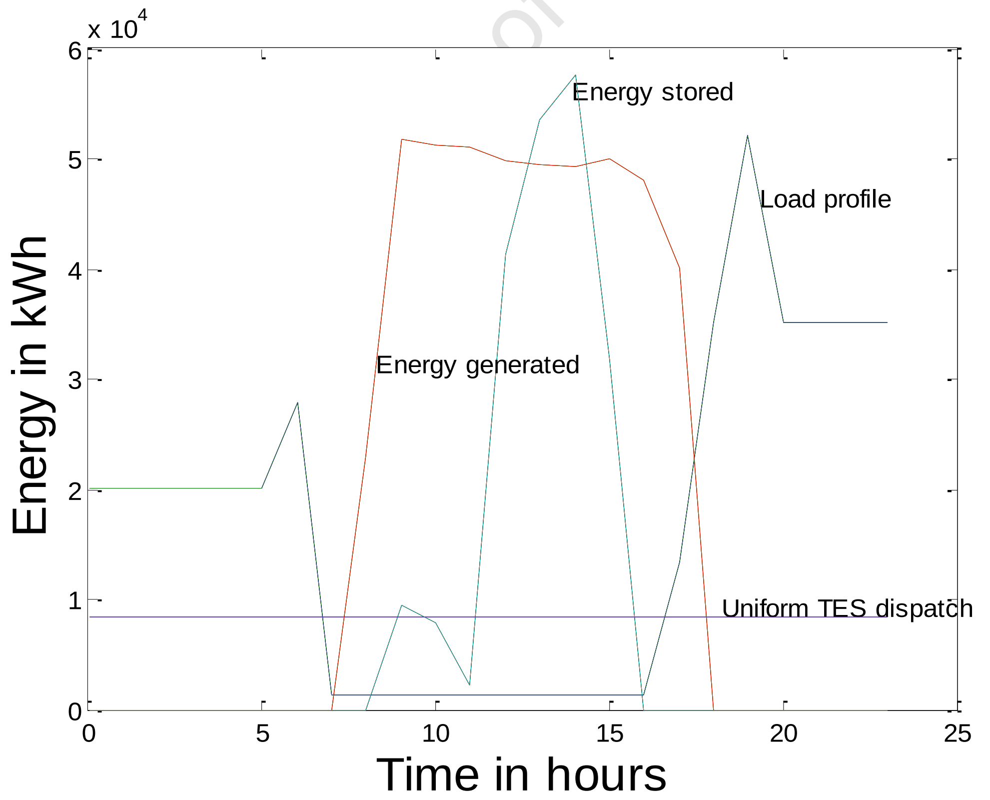

Figure 4-17 Hourly Energy Stored, Hourly Uniform TES Dispatch, Hourly energy generated and Load Profile of a dry cooled plant

The following section analyses the hourly energy production for both dry and wet cooled power plants. Due to the intermittent nature of solar energy operation on CSTP plants needs support from the grid when both the TES and the amount of energy collected from the DNI cannot support load demand.

4.7.1 Energy Generation of CSTP plants with Grid

As was shown in section 4.7 the CSTP plant is unable to meet the entire load demanded. This section discusses the hourly energy support the CSTP receives from the grid and also the amount of energy the CSTP sells to the grid in some hours of the day. In this case uniform and non-uniform TES dispatch has been considered. Non uniform TES dispatch supplies energy stored according to demand at that particular hour unlike uniform dispatch that uniformly distributes energy equally at all hours of the day. Table 4-17 shows the amount of energy supplied to and from the grid for a non-uniform TES dispatch. The amount of energy supplied to the grid from the CSTP plant is positive while the support energy obtained from the grid is indicated by a negative sign. The zero values indicate the hours the CSTP generates enough energy from TES to meet demand and does not require support from grid. The net energy is the difference between the energy supplied to the grid and obtained from the grid. In both cases as shown the plant is able to fully meet the load demand from 8am to 11pm after which the grid takes over.

Table 4-17 Energy supplied to and from the grid at the 1st day of January 2012 considering Non Uniform TES Dispatch

Hour of the day	Energy supplied to or from the grid, wet cooled plant (kWh)	Energy supplied to or from the grid, Dry cooled plant(kWh)	Load profile kWh
0000	-20,060	-20,060	20,060
0100	-20,060	-20,060	20,060
0200	-20,060	-20,060	20,060
0300	-20,060	-20,060	20,060
0400	-20,060	-20,060	20,060
0500	-20,060	-20,060	20,060
0600	-27,880	-27,880	27,880
0700	-1,360	-1,360	1,360
0800	21,183.1	21,641.1	1,360
0900	50,769.8	50,410	1,360
1000	52,172.8	49,939.5	1,360
1100	50,941.6	49,673.8	1,360
1200	50,119.4	48,475.8	1,360
1300	49,912	48,094.1	1,360
1400	49,966	47,942	1,360
1500	50,693.7	48,580.1	1,360
1600	48,059.6	46,616.7	1,360
1700	26,059.3	26,640	1,3430
1800	0	0	35,190
1900	0	0	52,190
2000	0	0	35,190
2100	0	0	35,190
2200	0	0	35,190
2300	0	0	35,190
Net Energy	**300,277.3**	**288,413.1**	403,410

The values of uniform TES dispatch (8607.75kWh for wet cooling and 8495 for dry cooling) obtained from section 4.7 were applied for a grid connected CSTP to obtain Table 4-18. Negative values indicate grid supply.

Table 4-18 Energy supplied to and from the grid at the 1st day of January 2012 considering Uniform TES Dispatch

Hour of the day	Energy supplied to or from the grid, wet cooled plant (kWh)	Energy supplied to or from the grid, Dry cooled plant(kWh)	Load profile kWh
0000	-11,452	-11,565	20,060
0100	-11,452	-11,565	20,060
0200	-11,452	-11,565	20,060
0300	-11,452	-11,565	20,060
0400	-11,452	-11,565	20,060
0500	-11,452	-11,565	20,060
0600	-19,273	-19,385	27,880
0700	7,247	7,135	1,360
0800	29,790.1	30,136.1	1,360
0900	59,376.8	58,905	1,360
1000	60,780.55	58,434.5	1,360
1100	59,549.35	58,168.8	1,360
1200	58,727.15	56,970.8	1,360
1300	58,519.75	56,589.1	1,360
1400	58,573.75	56,437	1,360
1500	59,301.45	57,075.1	1,360
1600	56,666	55,111.7	1,360
1700	34,666	35,135	1,3430
1800	-26,583	-26,695	35,190
1900	-43,583	-43,695	52,190
2000	-26,583	-26,695	35,190
2100	-26,583	-26,695	35,190
2200	-26,583	-26,695	35,190
2300	-26,583	-22,695	35,190
Net Energy	290,166	279,718	403,410

The comparison of the hourly amount of energy supplied to or from the grid for uniform and non-uniform TES dispatch is shown in Figure 4-18. The abbreviations in this figure stand for:

UDW-Uniform Dispatch with wet cooling

UDD-Uniform Dispatch with dry cooling

NUDW-Non Uniform Dispatch with wet cooling

NUDD-Non Uniform Dispatch with dry cooling

Uniform dispatch is preferred to non-uniform dispatch. This is because the hourly amount of energy obtained from the grid when the CSTP is not able to support load demand is less. This saves revenue for buying energy from the grid as some periods when the CSTP is not generating coincides with peak tariff charges.

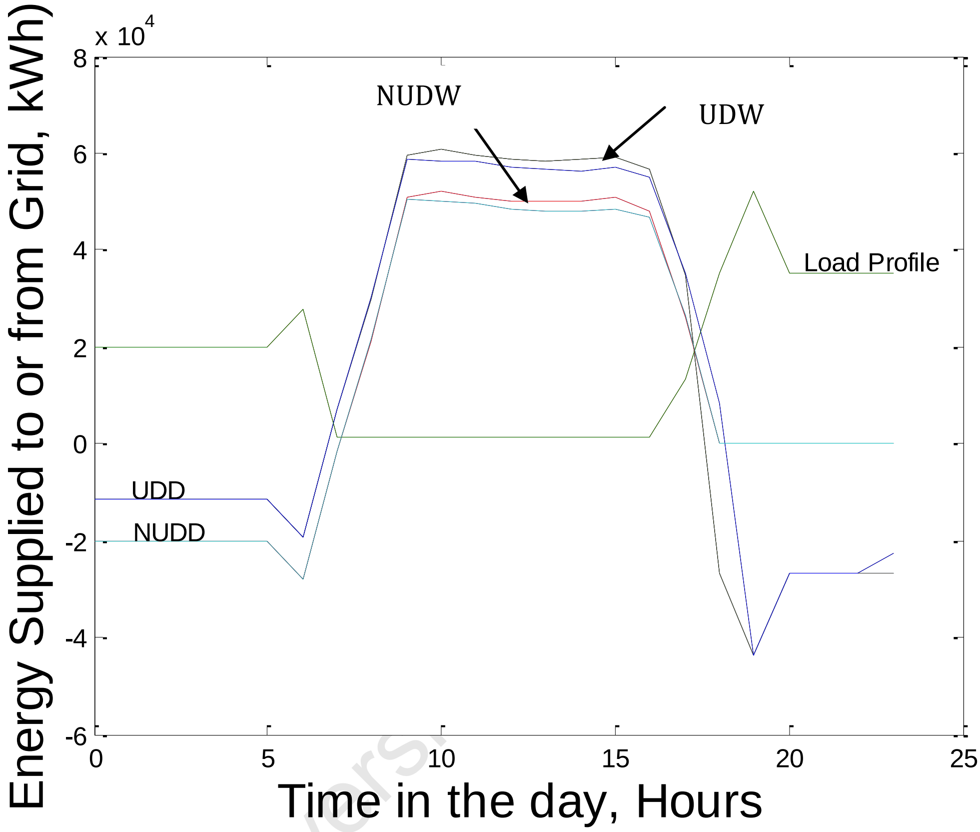

Figure 4-18 Uniform and Non Uniform TES dispatch used in a grid connected CSTP plant for a wet and dry cooled plant

The following section discusses the impacts of net energy production per hour on SM values of 1, 1.5 and 2. The size of the power block during this simulation was 50MWe and 12 hours of TES.

4.8 Impact of Solar Multiple on Dry and Wet cooling

Solar Multiple (SM) is the ratio of solar field thermal energy output to power block gross thermal energy demand at turbine design point conditions. The size of the solar field is specified by solar multiple. Solar multiple normalizes the solar field according to the size of the power block (capacity of the plant). For example a solar field with a specified solar multiple of 1 provides sufficient energy to run the power block under the reference conditions.

A solar field specified by a solar multiple of 3 covers thrice the solar field area covered by a solar multiple of 1. This then means that for smaller portions of land available, the amount of thermal energy collected is small which results to less energy generated as shown by Equation 4-30. The solar field thermal output is the amount of energy collectable from a certain field area specified by solar multiple. Its SI units are MWh.

The capacity of a CSTP power block is defined as the rated power to run the steam turbine. It is specified by the turbine maximum gross thermal input to the power block (MWh) or the rated output of the power block (MWe) [78]. The amount of DNI and the size of the solar field determine the amount of energy available to the power block. Sizing of both the power block and the solar field determines the capacity factor of the CSTP plant.

An undersized solar field will result to lower thermal energy delivery to the power block, hence reducing the capacity factor. Over sizing the solar field leads to energy dumping. This is because a lot of thermal energy is delivered to the turbine than its rated thermal input. The ability to oversize the solar field with respect to the power block is useful for parabolic trough plants which allow the plants to run at design point over a greater fraction of the year. For a plant with SM greater than 1 and without TES, the excess solar energy is reduced by defocusing of the collectors in order to match generation with the turbine gross thermal input.

If TES is available the excess energy can be diverted to TES and solar energy loss that might happen in case defocusing is minimised. This is the main reason why the plant with TES achieves higher capacity factor than plants without TES as discussed in sections 4.5.1, 4.5.2 and Figures 4-7 and 4-8.

In this simulation, a 50MWe capacity parabolic trough at a solar multiple of 2 is designed which matches the highest load demand estimated for Lodwar shown in section 3.20.1.1, Figure 3-37. The solar field (collector) maximum thermal output and power block thermal input is 291.468MWh and 145.734MWh respectively. The solar field thermal output is a function of DNI. In this thesis the irradiance at design used is 958.838W/m^2/day which was discussed in section 3.7.3. The actual area occupied by the plant is 428,914m^2.

$$SM_{Design} = \frac{Q_{Thermal,Field}}{Q_{Thermal,Power\,Block}} \quad \textbf{....... 4-30}$$

Where

$Q_{Thermal,Field}$ = the collector field maximum thermal output

$Q_{Thermal,Power\,Block}$ = the maximum thermal input of the turbine

The hourly amount of energy absorbed by the collectors, hourly solar field thermal output, and the hourly thermal energy to the power block in the 1st day of January 2012 is shown in Table 4-19 and Figure 4-19. They each increase with the amount of DNI collected. The field thermal output and thermal energy to the power cycle is zero between 6am and 8am though there is some DNI. In these hours the amount of energy collected is used for warming up the mirrors and heating up the HTF to its optimum temperature that can generate steam. The HTF used in this thesis has a maximum temperature of 393^{0}C. The HTF generates steam from water at 377^{0}C, at 100 bars of pressure. When the solar field thermal output is greater than the design power block thermal input the excess energy is stored in TES.

The total field thermal output is 11.77% lower compared to the total amount of energy absorbed by the HTF for a dry cooled parabolic trough. The loss is due to the parasitic losses in the HTF transit to the steam Rankine cycle. These parasitic losses include, thermal energy absorption by the pipes carrying HTF.

Table 4-19 Hourly Energy absorbed, Hourly Field thermal output, Hourly Thermal Energy to the Power block and Hourly DNI at a Solar Multiple of 2, 50MWe capacity size for a Dry cooled plant in the 1st day of January 2012

Hour of the day	Total Thermal Energy Absorbed MWh	Solar Field Thermal Output MWh	Thermal Energy to the power Block MWh	DNI(Wh/m^2)
0000	0	0	0	0
0100	0	0	0	0
0200	0	0	0	0
0300	0	0	0	0
0400	0	0	0	0
0500	0	0	0	0
0600	0	0	0	8
0700	25.9509	0	0	246
0800	153.082	0	0	595
0900	154.804	152.756	145.734	605
1000	153.448	153.016	145.734	604
1100	147.66	147.428	145.734	586
1200	189.012	186.546	145.734	736
1300	199.83	199.206	145.734	775
1400	202.378	202.199	145.734	782
1500	175.614	175.723	145.734	683
1600	113.903	113.895	145.694	461
1700	63.3208	62.723	137.595	328
1800	0	0	79.8201	38
1900	0	0	0	0
2000	0	0	0	0
2100	0	0	0	0
2200	0	0	0	0
2300	0	0	0	0
Total	**1579**	**1393**	**1383**	6447

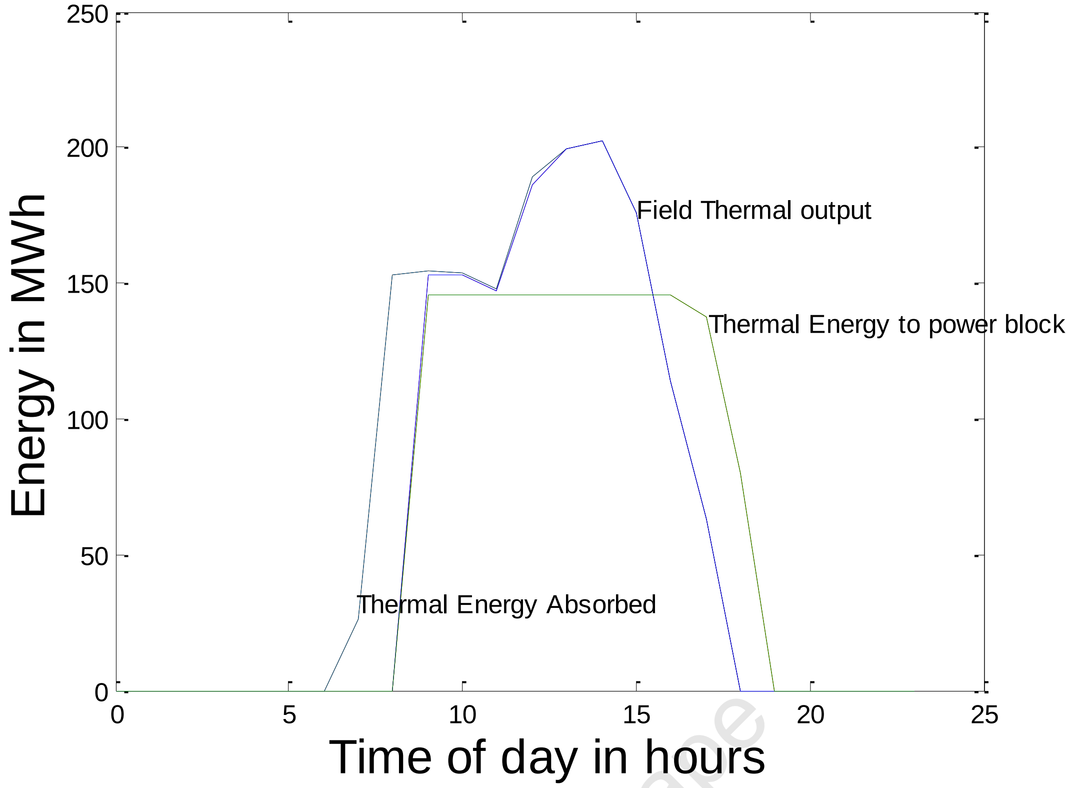

Figure 4-19 Hourly Energy absorbed, Hourly Field thermal output, Hourly Thermal Energy to the Power block at a solar multiple of 2 for a Dry cooled plant in the 1st day of January 2012

4.8.1 Hourly Net Energy at lower Solar Multiples values in a wet cooled

The hourly net electric output E_{net} of a CSTP plant is a function of the gross turbine energy output $E_{Grossturbine}$ and the parasitic energy losses $E_{parasitics}$ as shown in Equation 4-31[88].

$$E_{net} = E_{Grossturbine} - E_{parasitics} \quad \textbf{4-31}$$

As previously defined in section 4.8 Equation 4-30, SM expresses the solar field thermal energy output as a function of the power block capacity.

The data in Table 4-20 provide the Hourly AC electricity (kWh) generation from CSTP and the load demand for wet cooled plant at different SM values without TES or grid support. The hourly amount of energy stored for each SM values considered for wet and dry cooled plants is shown in Appendix A-1, Table 7-1 and Table 7-2 respectively.

Table 4-20 Hourly Net Energy Generation for Various SM values for a wet cooled plant 1st day of January

Hour	Energy at SM=1 kWh	Energy at SM=1.5 kWh	Energy at SM=2 kWh	DNI(Wh/m^2)	Load profile kWh
0	0	0	0	0	20,060
1	0	0	0	0	20,060
2	0	0	0	0	20,060
3	0	0	0	0	20,060
4	0	0	0	0	20,060
5	0	0	0	0	20,060
6	0	0	0	8	27,880
7	0	0	0	246	1,360
8	5,460.8	15,825.7	22,543.1	595	1,360
9	27,322.9	42,072	52,129.8	605	1,360
10	26,847.7	41,634.7	52,172.8	604	1,360
11	25,773	40,143.8	52,301.6	586	1,360
12	33,651.2	50,993.6	51,479.4	736	1,360
13	35,776.2	52,039.8	51,272	775	1,360
14	36,411.6	52,170.6	51,326.2	782	1,360
15	31,445.7	47,922	52,053.7	683	1,360
16	19,501.9	30,989.7	49,419.6	461	1360
17	0	14,451.2	39,489.3	328	1,360
18	0	0	22,561.7	38	13,430
19	0	0	0	0	35,190
20	0	0	0	0	52,190
21	0	0	0	0	35,190
22	0	0	0	0	35,190
23	0	0	0	0	35,190
Totals	**242,191**	**388,243**	**496,749**	**6447**	**403,410**

Figure 4-20 shows the amount of energy produced from a wet cooled plant at SM=1, 1.5 and 2 against the load profile on the first day of January.

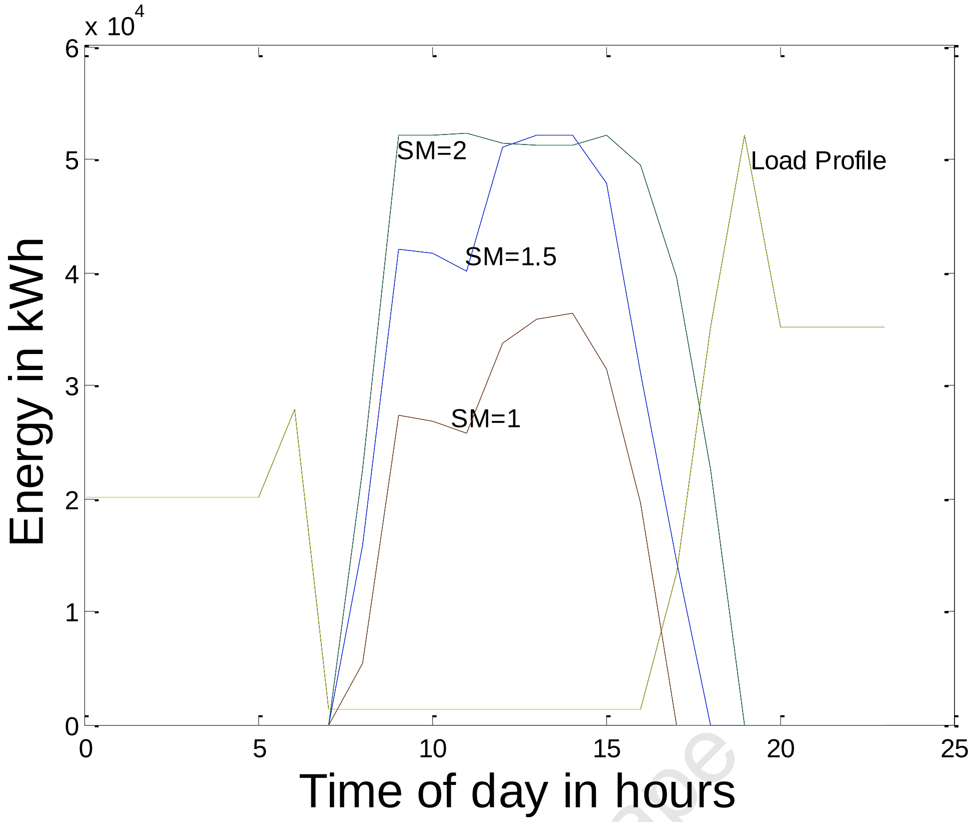

Figure 4-20 Load Profile against Energy in Varying SM values for Wet cooled Plant without grid or TES support

Table 4-21 shows the hourly amount of energy supplied to and from the grid by the CSTP plant with TES support at the different SM values. The negative values indicate the amount of energy supplied by the grid to the CSTP plant.

The net amount of energy supplied to the grid at SMs of 1, 1.5 and 2 is 65,561kWh, 223,214kWh, 300,277.3kWh respectively. The ability to meet the hourly load is highest for CSTP plants with SM of 2. This is owed to its ability to oversize its field to collect more energy as mentioned in section 4.8. The results of the dry cooled plant are shown in Table 7-3 and Figure 7-1 in Appendix A-1.

Table 4-21 Energy supplied to and from the grid at the 1st day of January 2012 at different SM values for a wet cooled plant with non-uniform TES dispatch inclusive

Hour	Energy supplied to or from the grid, wet cooled plant (kWh) SM=1	Energy supplied to or from the grid, wet cooled plant (kWh) SM=1.5	Energy supplied to or from the grid, wet cooled plant (kWh) SM=2	Load profile kWh
0	-20,060	-20,060	-20,060	20,060
1	-20,060	-20,060	-20,060	20,060
2	-20,060	-20,060	-20,060	20,060
3	-20,060	-20,060	-20,060	20,060
4	-20,060	-20,060	-20,060	20,060
5	-20,060	-20,060	-20,060	20,060
6	-27,880	-27,880	-27,880	27,880
7	-1,360	-1,360	-1,360	1,360
8	4,100.8	14,465.7	21,641.1	1,360
9	25,962.9	40,712	50,410	1,360
10	25,487.7	40,274.7	49,939.5	1,360
11	24,413	38,788.8	49,673.8	1,360
12	32,291.2	49,633.6	48,475.8	1,360
13	34,416.2	50,679.8	48,094.1	1,360
14	35,051.6	50,810.6	47,942	1,360
15	30,085.7	46,562	48,580.1	1,360
16	18,141.9	29,629.2	46,616.7	1360
17	-1,360	13,091.2	26,640	1,360
18	-13,430	-1,833	13,430	13,430
19	-35,190	-35,190	35,190	35,190
20	-52,190	-52,190	52,190	52,190
21	-35,190	-35,190	35,190	35,190
22	-35,190	-35,190	35,190	35,190
23	-35,190	-35,190	35,190	35,190
Totals	**65,561**	**223,214**	**300,277.3**	**403,410**

The following section investigates the impacts of initial temperature difference (ITD) on LCOE for a dry cooled plant.

4.9 Impacts of ITD Variation on LCOE for a Dry Cooled plant

As previously mentioned on section 4.2.3 initial temperature difference (ITD) is the difference in temperature between the set condenser temperature and the incoming ambient air for cooling the condenser. It applies only for the dry cooled parabolic trough. In this simulation a 50MWe capacity with 12 hours TES is considered. The inflation rate and discount rate were 16% and 10% respectively.

An air cooled condenser with a lower ITD is bigger in size and more capital intensive than condenser with a higher ITD [112]. This is illustrated on Figure 4-21 below where two condensers are considered.

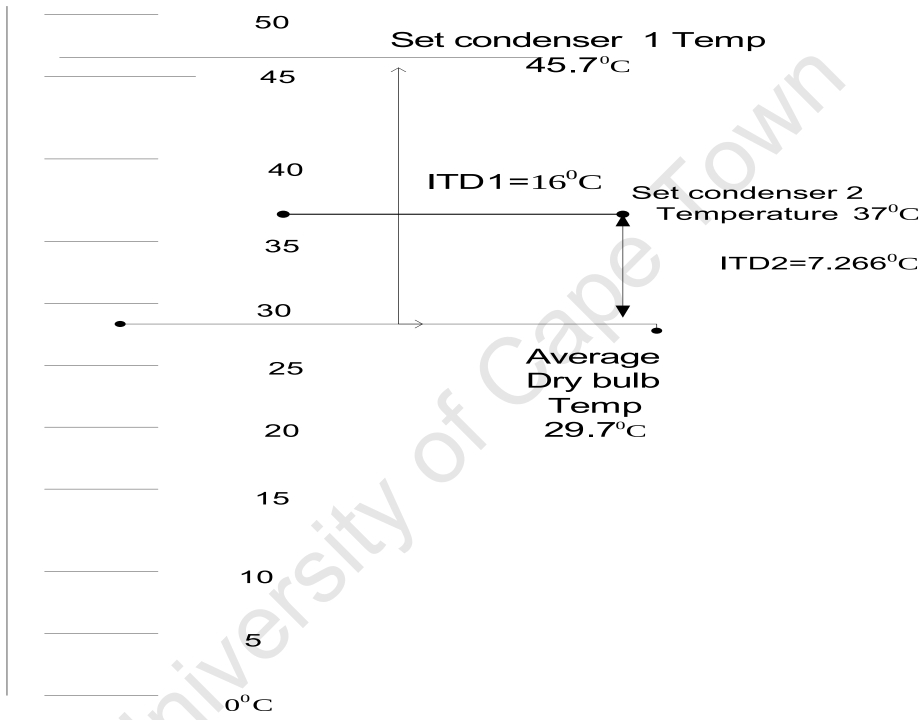

Figure 4-21 Illustration of ITD variation on two Condensers

Condenser 1 has a set temperature of 45.7°C while condenser 2 set temperatures was 37°C. The average dry bulb temperature for Lodwar as previously mentioned on section 4.2.3 is 29.734°C. Therefore ITD1 and ITD2 are each 16°C and 7.266°C respectively. If the dry bulb temperatures rises to 35°C and assuming the exhaust steam from the turbine is at 105°C the energy that the two condensers draw to eject this heat is calculated as shown in Equations 4-32 and 4-33.

$$H_{C1} = m_{steam} \text{ x } C_{p,\,water} \text{ x} \Delta T1 \text{.......} \mathbf{\ 4\text{-}32}$$

$$H_{C2} = m_{steam} \text{ x } C_{p,\,water} \text{ x} \Delta T2 \text{.......} \mathbf{\ 4\text{-}33}$$

where

H_{C1} =Energy used by condenser 1(Joules)

H_{C2} = Energy used by condenser 2 (Joules)

$C_{p,\,water}$ =Heat capacity of water (4.2kJ/kg. °C)

$\Delta T1$=Temperature difference between the exhaust steam and condenser 1(105°C-45.3°C) =59.7°C

$\Delta T2$ = Temperature difference between the exhaust steam and condenser 2(105°C-37°C) =68°C

m_{steam} =mass of steam on the condenser per second

The energy of the exhaust heat per unit time on condenser 1 and condenser 2 is ($250.74 x\, m_{steam}$) and ($285.6 x\, m_{steam}$) respectively.

The amount of energy drawn by the fans of the two condensers to counter the exhaust per unit time heat is as shown in Equations 4-34 and 4-35.

$$E_{C1} = m_1 x C_{P,\,air} \text{ x } ITD1 \qquad \textbf{....... 4-34}$$

$$E_{C2} = m_2 \text{ x } C_{P,\,air} \text{ x } ITD2 \qquad \textbf{....... 4-35}$$

where

E_{C1} =energy drawn by fans in condenser 1 to cool H_{C1}

E_{C2} =energy drawn by fans in condenser 2 to cool H_{C2}

m_1 =mass of air drawn by fans in condenser 1

m_2 =mass of air drawn by fans in condenser 2

$C_{P,\,air}$ = heat capacity of air (1kJ/kg °C)

For cooling to occur the heat drawn by the fans E_{C1} and E_{C2} must equal the exhaust heat H_{C2} and H_{C1}. The heat balance for the two condensers is shown in Equations 4-36 and 4-37.

$$m_{steam} \text{ x } C_{p,\,water} \text{ x}\Delta T1 = m_1 x C_{P\,air} \text{ x } ITD1 \qquad \textbf{....... 4-36}$$

$$m_{steam} \text{ x } C_{p,\,water} \text{ x}\Delta T2 = m_2 \text{ x } C_{P\,air} \text{ x } ITD2 \qquad \textbf{....... 4-37}$$

Therefore the mass of air drawn per unit time by condenser 1 and 2 is shown in Equation 4-38 and 4-39 respectively.

$$m1 = \frac{250.74x\ m_{steam}}{ITD1} = 15.625 kg / unit\ time \text{ .... } \mathbf{4\text{-}38}$$

$$m_2 = \frac{285.6x\ m_{steam}}{ITD2} = 37.285 kg / unit\ time \text{ .... } \mathbf{4\text{-}39}$$

The ratio of mass of air that must be drawn by the two condensers is shown by Equation 4-40.

$$\frac{m_2}{m_1} = 2.386 \text{ .... } \mathbf{4\text{-}40}$$

The fan in condenser 2 therefore must blow in air at a rate of 2.386 times more than condenser 1. Therefore condenser 2 must be bigger in size and stronger than condenser 1. It is therefore more capital intensive to operate condensers at lower ITDs. LCOE is a function of capital costs and annual energy production as shown in section 4.5.1, Equation 4-24. Therefore for lower ITD, the LCOE rises as shown in Table 4-22 and Figure 4-22. LCOE decreases by12% from 5°C to 10°C after which the decrease is gradual. The reason is that at lower ITDs capital costs are higher and hence the LCOE. The best (optimum) design ITD temperature of the dry cooled condenser was found to be 15°C. The variation of ITD with the investment costs is shown in Figure 7-2 in Appendix A-1.

Table 4-22 Impacts of ITD on LCOE

ITD (^{0}C)	LCOE ($cents/kWh)
5	103.893
10	91.4147
15	88.2248
20	89.5469
25	89.2186

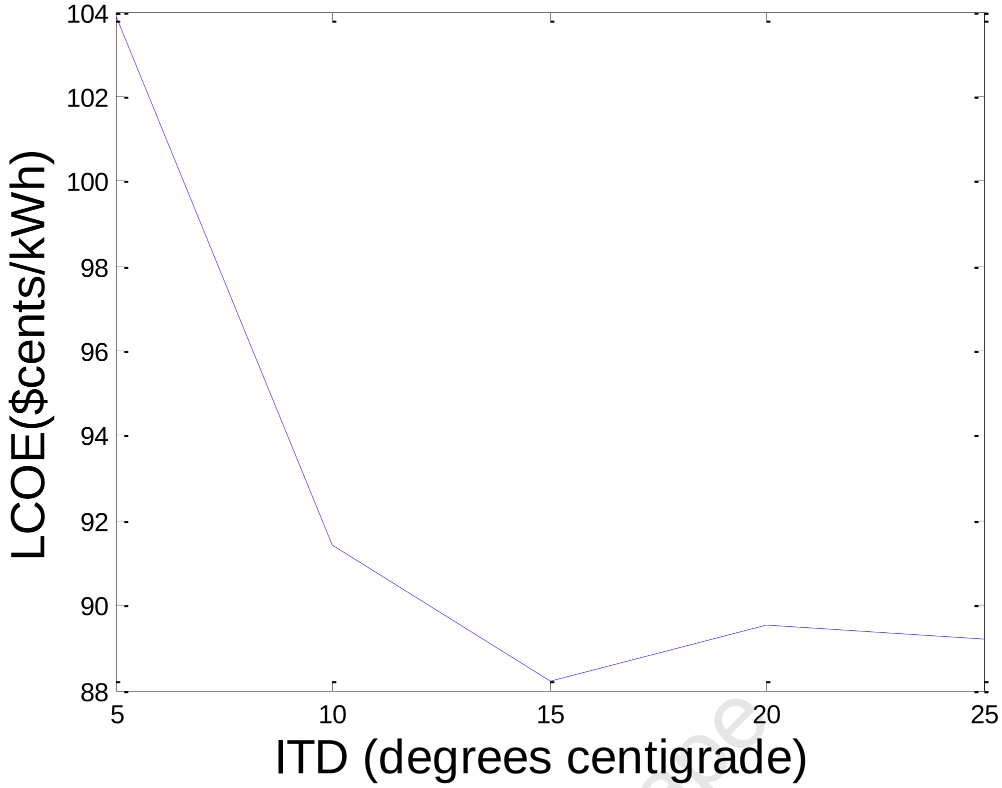

Figure 4-22 Variation of LCOE with increasing ITD

The following section discusses the impacts of varying the approach temperature on the amount of energy generation and the LCOE for a wet cooled plant.

4.10 Impacts of variation of Approach Temperature on LCOE for a Wet Cooled Plant

Approach temperature is the temperature difference between the incoming water for condenser cooling and the wet bulb temperature. Most wet cooled condensers are designed to handle an approach temperature between 3°C to -10°C [112]. In this simulation the approach temperature is varied between 3 -10°C. The capacity of the power block is maintained at 50MWe while the size of storage is 12 hours. The inflation and discount rate are each 16% and 10% respectively. The LCOE increases as the approach temperature of the cooling water increases while the capacity factor decreases as shown in Table 4-23 and Figure 4-23 . Both LCOE and capacity factor are a function of net energy produced. At higher approach temperatures the parasitic losses are high and hence the net energy yield is low as shown in Equation 4-41. Given a constant capital cost the LCOE increases according to the definition given in section 4.5.1, Equation 4-24. On the other hand the capacity factor decreases with reduction on the net energy as discussed on section 4.2.4, Equation 4-15.

$$\text{Net output energy (kWh)} = \text{Gross energy (kWh)} - \text{losses (kWh)} \quad \textbf{....... 4-41}$$

Table 4-23 Impacts of Approach Temperature on LCOE and Capacity Factor

Parameterized Input(s)	LCOE Real	Capacity Factor
Approach temperature=3	91.1256	31.2068
Approach temperature=5	91.1688	31.191
Approach temperature=7	91.2017	31.179
Approach temperature=9	91.2525	31.1605
Approach temperature=10	91.3115	31.139

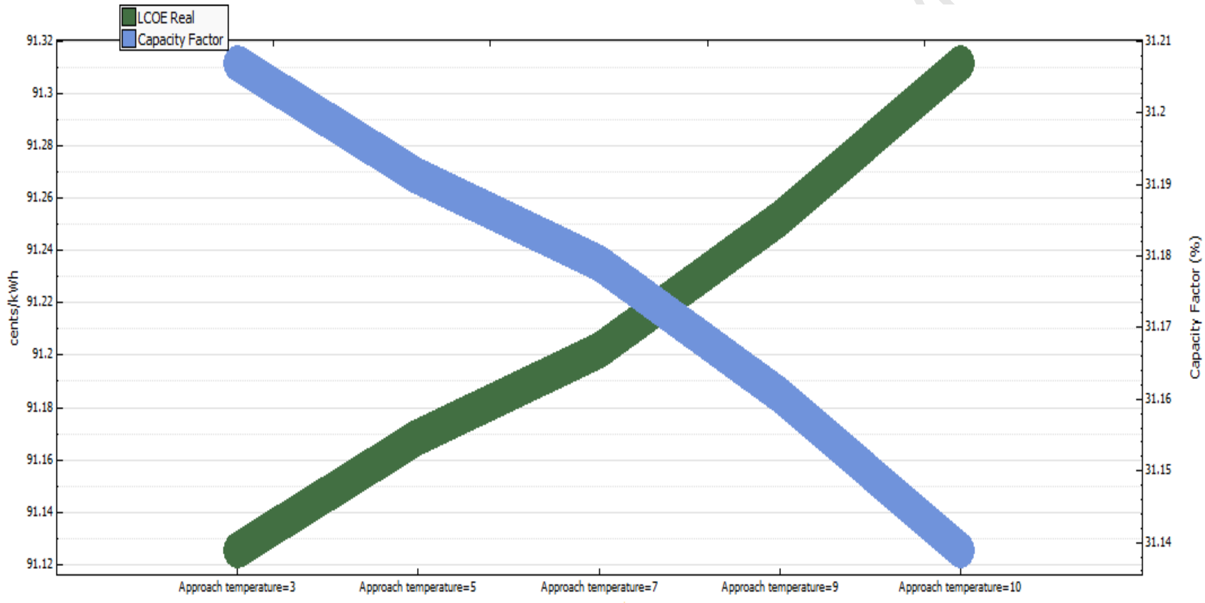

Figure 4-23 Approach Temperature and its Impacts on LCOE and Capacity Factor

In the following section stability of grid and standalone operation modes is discussed.

4.11 Stability analysis of a Grid and Standalone Operation Modes

This section investigated the stability of the synchronous generator operated both as a standalone and grid connected at SM=2.The parameters looked herein were real power, reactive power, voltage and frequency.

4.11.1 Impacts of a 3 phase on Active and Reactive power of a CSTP synchronous generator when operated as a standalone and grid connected modes

When a fault was applied on a standalone synchronous generator the active power decreased to about 20MW and rose again to 53MW after 5 s. It was restored back after about 40s. The reactive power decreased to zero and was restored back to normal after 20 s. The grid connected the system behaved like an induction motor by drawing large amounts of reactive power from the grid. The system was restored back to normal after 8s. Hence for grid connected the system was more stable. This was shown in Figure 4-24 and Figure 4-25.

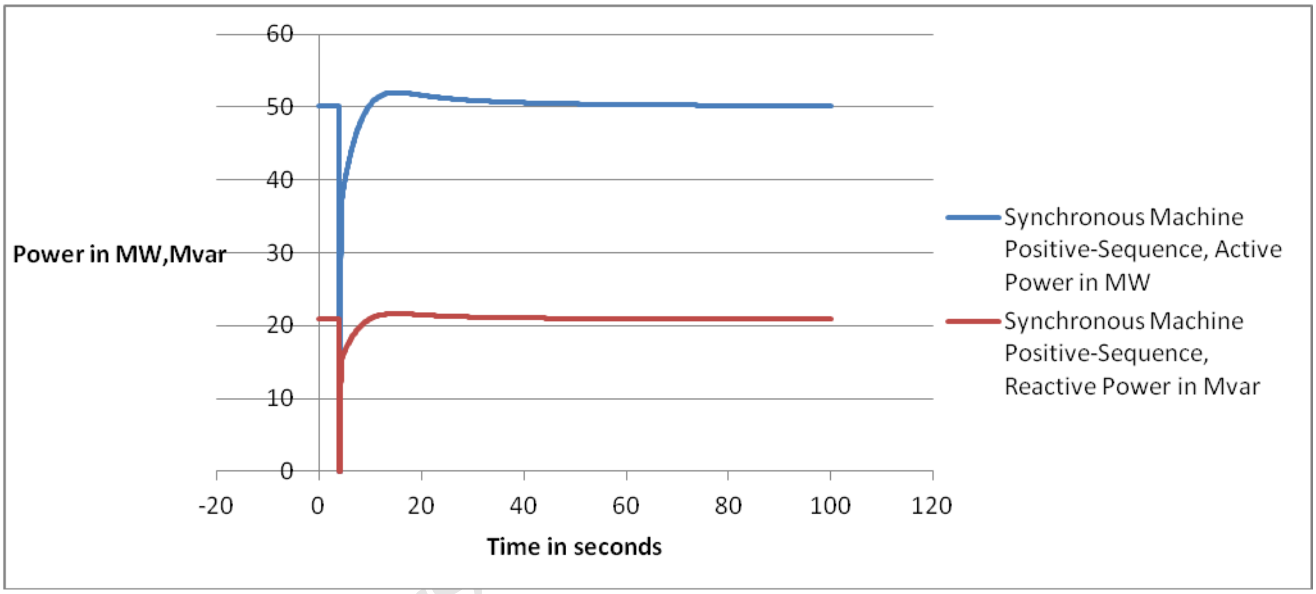

Figure 4-24 Output active and reactive power for Stand alone

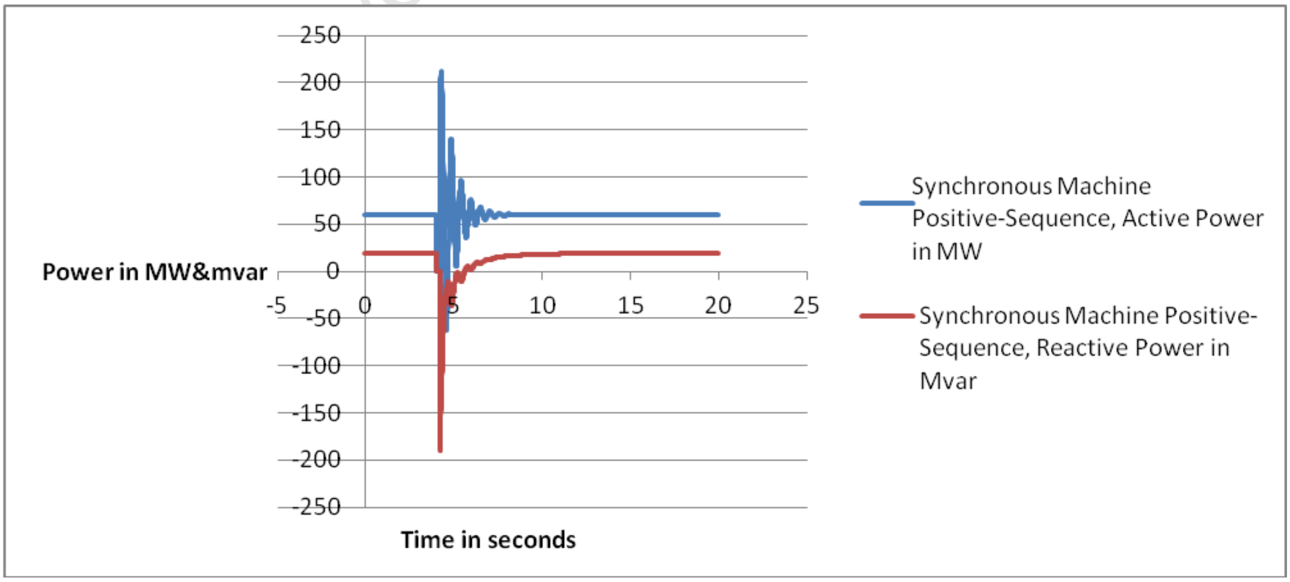

Figure 4-25 Output active and reactive power for grid connection

4.11.2 Impacts of a fault on Frequency of a CSTP synchronous generator when operated as a standalone and grid connected modes

After the fault was applied frequency restoration was faster for grid connected compared to standalone synchronous generator as shown in Figure 4-26 and Figure 4-27. Frequency restoration to 50 Hz occurred at 6 seconds for a grid connected system. Standalone system restored the frequency after 75 seconds. The system is hence more stable when connected to the grid.

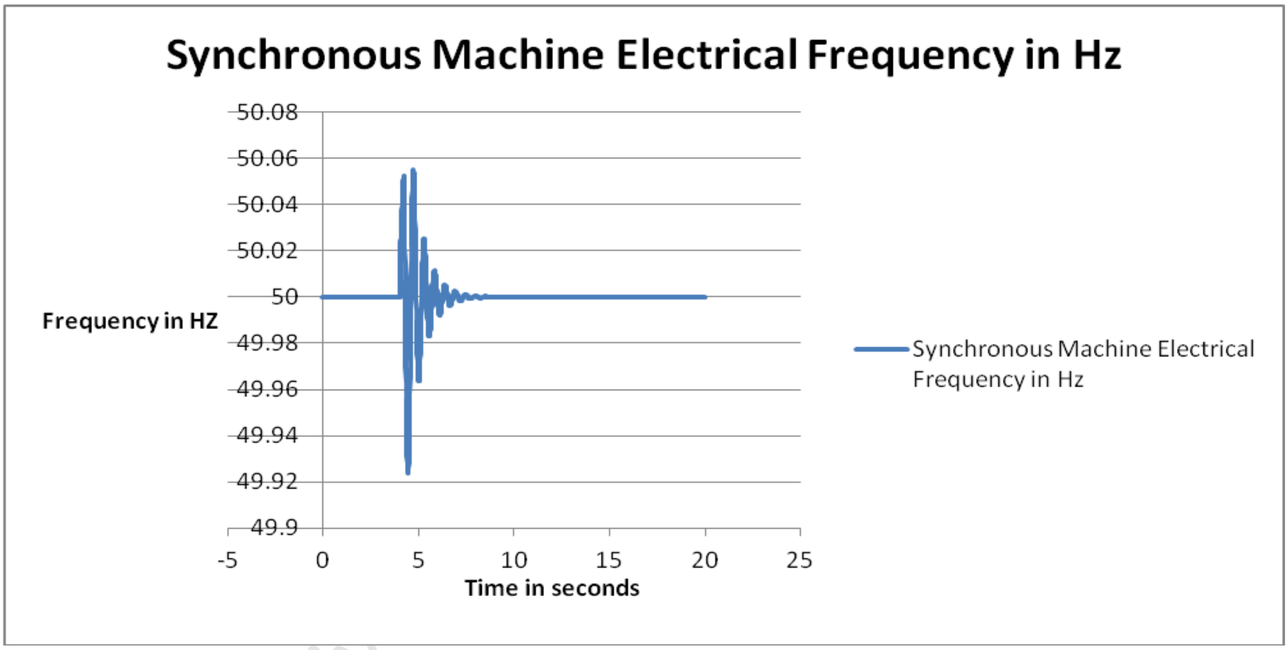

Figure 4-26 Frequency of Generator, Grid connected

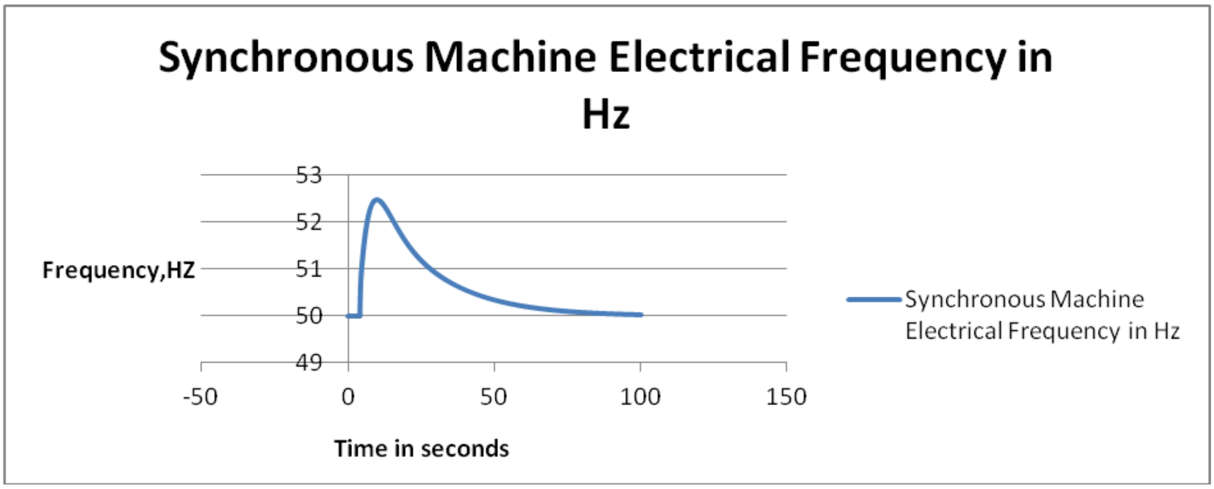

Figure 4-27 Frequency of Generator, Stand alone

4.11.3 Impacts of a fault on voltage of a CSTP synchronous generator when operated as a standalone and grid connected modes

Reactive power is used for voltage stabilisation. After fault application the reactive power reduced as previously discussed. This leads to voltage reduction as shown in Figure 4-28 and Figure 4-29. The collapse in voltage occurs because of reactive power deficiency [111].

Regaining of the normal voltage occurred at 6 seconds for a grid connected case. Standalone system took about 30 seconds to restore the voltage back to normal. The voltage levels during and after the fault was more sustained when the synchronous generator was connected to the grid.

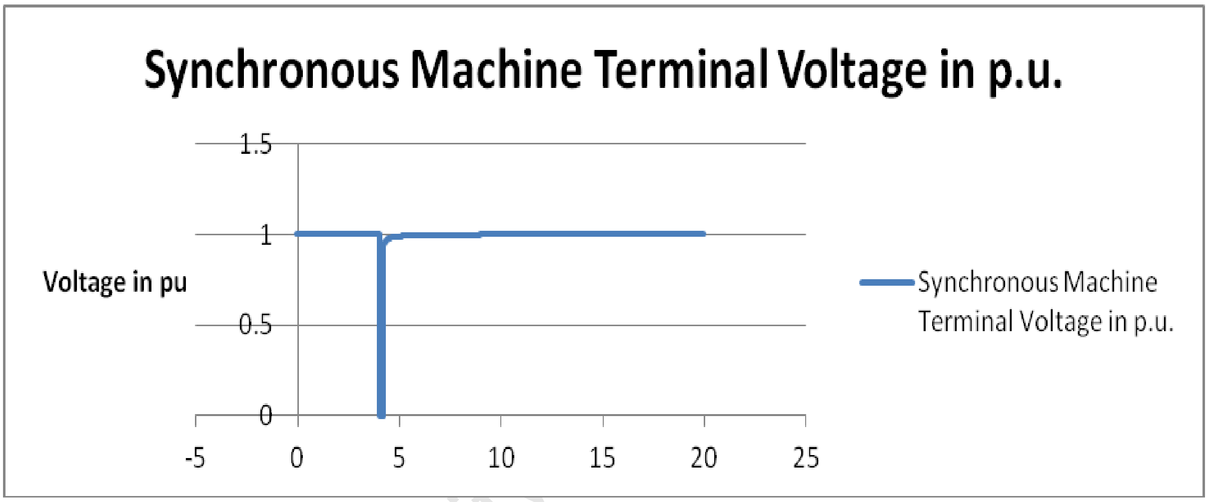

Figure 4-28 Terminal voltage, Grid connected

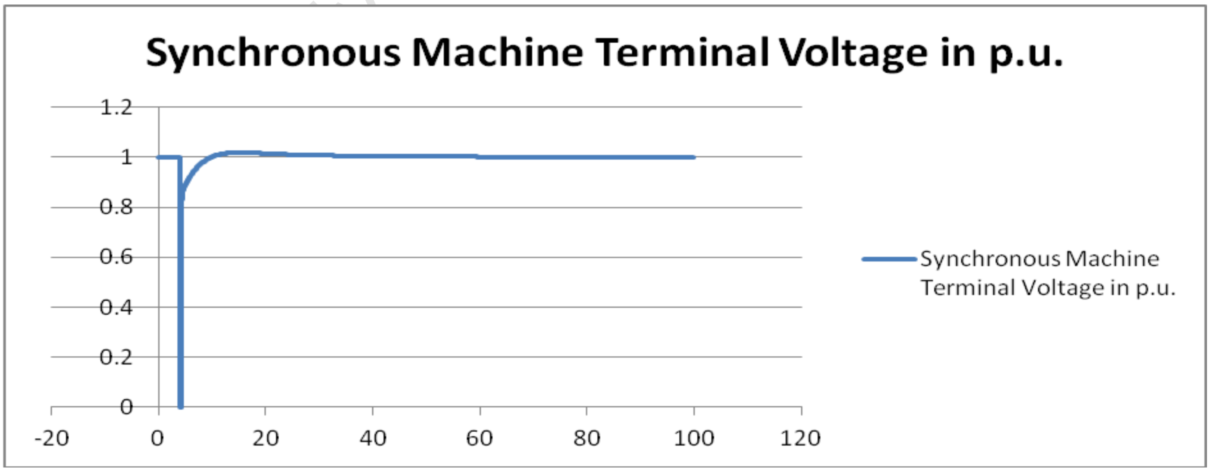

Figure 4-29 Terminal voltage, Stand alone

The results of the impacts of a 3 phase fault on speed of a standalone and grid connected CSTP synchronous generators are shown in Figure 7-4 and Figure 7-5 in Appendix A2. The simulation result diagrams of grid connected and standalone system are also shown in Figure 7-6 and Figure 7-7 in Appendix A2.

The following section discusses the environmental impacts of CSTP deployment on carbon dioxide avoided, land clearance (deforestation) and impacts of flora and fauna.

4.12 Determination of the annual carbon dioxide and Deforestation avoided by CSTP installation

Parabolic trough CSTP plants provide benefits by generating power without producing CO_2. In addition to this the use of a fixed cost renewable energy such as parabolic troughs helps decrease the use of fossil fuels. This reduces the emission and the high costs of fossil fuels such as paraffin.

In order to estimate the amount of carbon dioxide emissions prevention and deforestation brought about by the use of firewood the following assumptions were made:

- Burning 5.5kg of firewood emits 11kg $CO_{2.}$ In the rural areas of Kenya it is estimated that each person uses 0.55-0.83 kg of firewood per day. On average each household in the rural communities of Kenya has five members.
- Assuming that an acre of land accommodates 500 well spaced trees, each will take an average of 10-12 years to grow. A fully grown tree weighs [105,106].

Table 4-24 is used to show the total amount of deforestation and carbon dioxide reduction as a result of building a parabolic trough plants. The mass of firewood used by 17000 households is calculated using Equation 4-42.

$$Mass\,of\;\;firewood\,/\,yr = 17000*(0.83*5)*365 = 128,753,750kg \quad \textbf{....... 4-42}$$

The number of trees cut is estimated using Equation 4-43.

$$\text{Number of trees cut per year} = \frac{\text{Mass of firewood per year}}{\text{weight of one tree}} = \frac{128{,}753{,}750}{1000} = 128{,}754 \quad \textbf{....... 4-43}$$

The total amount of carbon dioxide produced per year is calculated using Equation 4-44. The

$$Total\ mass of\ CO_2 / yr = \frac{128753750}{5.5} * 11 = 257,507,500 kg$$ **....... 4-44**

Table 4-24 Estimated Deforestation and Carbon Dioxide Emissions

No of households	17,000
Firewood used by each person in a household	0.83kg
No of persons/household	5
Total Kg of firewood/yr	128,753,750kg
Total CO_2 emissions	257,507,500kg
Trees felled/year	128,754 trees
Area cleared per year	$1.05x10^6 m^2$.

About 80% of the homesteads in the rural communities do not have electricity and mainly depend on forest wood for their daily energy needs. About 70% of electricity generated in Kenya comes from hydro. However in the recent past rivers have dried up as a result of deforestation in the Mau forests in the search for energy which have forced the government of Kenya to intervene and vacate about 10,000 families from this area [4]. In the table above it has been shown that $1.05x10^6 m^2$ of land is cleared every year by the rural Kenyan communities. In a period of 12 years before a tree matures a total of $1.3x10^7 m^2$ will have been cleared. This therefore shows the environmental viability of parabolic trough deployment.

4.12.1 Impacts of parabolic trough on flora and fauna

Building of parabolic trough plants may have negative impacts on the flora and fauna. Mortalities on vertebrates are the main concern on the environmental impacts of the parabolic trough. Mortality occurs when the vertebrates collide with the mirrors or suffocate because of the generated heat on the mirrors. Birds can be visually impaired by the strong reflections causing casualties. In some climates vegetation can grow below the mirrors which can potentially cause fire.

The following section discusses the sensitivity analysis of parabolic trough net present value (NPV) and LCOE due to variation of inflation rate.

4.13 Sensitivity Analysis of parabolic trough NPV and LCOE due to inflation variation

Inflation is the rise in the general price level reported in rates of change of currency. This therefore means that the value of money goes down and it takes more money to purchase items for the plant. For example 4% inflation rate means the price level of that given year has risen by 4% from a certain reference year.

Net present value (NPV) is the sum of present values (PVs) of the individual cash flows of the same entity as shown by Equation 4-45 [100].

$$NPV = \sum_{n=1}^{N} \frac{R_n - C_{After\,Tax,n}}{(1+IRR)^n} + C_{After\,Tax,0} \quad \text{....... 4-45}$$

where

R_n = Annual Revenue for the year

IRR =Internal Rate of Return

IRR rate of return is the compounding rate which makes cumulative discounted cash flow equal zero at the end of a project. It is often used in capital budgeting that makes the net present value of all cash flows from a particular project equal to zero. Generally speaking, the higher a project's internal rate of return, the more desirable it is to undertake the project. As such, IRR can be used to rank several prospective projects a firm is considering. Assuming all other factors are equal among the various projects, the project with the highest IRR would probably be considered the best and undertaken.

$After\ tax, n$ =The net cash flow (the amount of cash, inflow minus outflow) after 30 years.

$C_{After\,Tax,0}$ = The net cash flow (the amount of cash, inflow minus outflow) in the first year.

As shown in Table 4-25 and Table 4-26 the inflation rate increases NPV and LCOE. They increase because of the following reasons:

- Inflation will mean higher costs of goods and higher selling prices.
- Inflation directly affects the financing needs and so it does to the capital cost.
- Since fixed assets like the generators, motors, collectors and other infrastructures supporting the parabolic trough increases with the decrease in money value, the same quantities must be financed by increasing amount of capital cost.

Table 4-25 Effects of inflation Rate on NPV and LCOE for a Wet cooled plant

Inflation rate	NPV($)	LCOE($cents/kWh)
15	561,950,770	67.49
16	677,097,434	82.91
17	873,231,975	126.49
18	1,113,363,066	124.99
19	1,165,098,443	129.95

Table 4-26 Effects of inflation Rate on NPV and LCOE on a Dry cooled plant

Inflation rate	NPV($)	LCOE($cents/kWh)
15	526,430,594	74.35
16	693,137,279	91.33
17	897,288,852	112.15
18	1,231,705,495	135.27
19	1,072,143,370	129.95

4.14 Sensitivity Analysis of parabolic trough NPV and LCOE due to sales tax variation

Increasing the sales tax (VAT-value added tax) increases both the LCOE and NPV. By definition the present value is the point at which the Net Present value (summation of the present values, PV, of the cash flows) for a project is zero. This is shown in Equation 4-46 and 4-47[110].

$$NPV = \sum_{t=0}^{n} PV = 0 \text{ .... 4-46}$$

where

$$PV = \frac{EBIT(1-T) + DEP - CAPEX}{(1+r)^t} \text{ .... 4-47}$$

And

EBIT=Earnings before interest and tax

DEP=Depreciation; Straight line method of depreciation is used in this research

CAPEX=capital expenditure

T=sales tax (%): A VAT of 16% was used for this simulation.

r=discount rate: In this simulation a discount rate of 10% is used.

As shown by Equations above increasing the sales tax leads to an increase in NPV and hence the LCOE. This is shown in Table 4-27 and Table 4-28.

Table 4-27 Impact of Sales Tax variation on NPV and LCOE, wet cooled plant

Sales Tax	LCOE Real ($cents/kWh)	Net Present Value ($)
0	88.1377	6.14019×10^8
4	89.0459	6.20741×10^8
8	89.954	6.27462×10^8
12	90.8622	6.34184×10^8
16	91.7703	6.40904×10^8
20	92.6784	6.47626×10^8

Table 4-28 Impact of Sales Tax variation on NPV and LCOE, Dry cooled plant

Sales Tax	LCOE Real ($cents/kWh)	Net Present Value ($)
0	97.6848	6.29294×10^8
4	98.7739	6.36908×10^8
8	99.8629	6.44523×10^8
12	100.952	6.52137×10^8
16	102.041	6.59752×10^8
20	103.13	6.67366×10^8

The following section presents the cash flow discussion of CSTP plants.

4.15 Cash flow

Cash flow is the movement of money in and out of a project, business or a financial project as shown in Table 7-4 in Appendix A-1. All the financial parameters assumed in this simulation are specified on Table section 3.21.4, Table 3-25. The main constituents of the cash flow are discussed below.

Energy (kWh): Energy is the total AC electricity generated by the CSTP plant per year. It is measured in kWh. The amount of energy generated per year is described by Equation 4-48.

$$\text{Energy in year one} = \text{Sum of simulation values x availability} \quad \textbf{....... 4-48}$$

Where

Sum of simulation values is the systems total electrical output (kWh)

Availability=number of hours the plant is producing energy expressed as a factor of the total number of hours in a year (8760).

Energy price ($/kWh): This is the price of energy per kWh. In this simulation it has been assumed that the energy price is escalated by 1.2% in each year to cover up the maintenance costs, salaries and taxes.

Energy value: This is the measure of electricity generated by a CSTP plant. It is calculated using Equation 4-49.

$$\text{Energy value (\$)} = \text{energy (kWh) x energy price(\$/kWh)} \quad \textbf{....... 4-49}$$

Operational and maintenance costs: These are the costs incurred in the proper running of the plant. They include salaries, wages and spare parts. These costs can be fixed O&M or variable O&M. Fixed operational costs are the costs incurred whether the CSTP plant is operating or not. They include salaries of the employees. Fixed costs in this model are calculated using Equation 4-50 below.

$$\begin{aligned}&\text{Fixed O \& M in year n}\\&= \text{Fixed annual cost (\$/year) x } (1 + \text{inflation rate} + \text{escalation rate})^{n-1}\end{aligned} \quad \textbf{....... 4-50}$$

Where

n = the year considered i.e. 1, 2, 3, 4…..30.

Variable costs are costs incurred when the plant is operating. They include consumables such as oil and water. These costs increase as the plant approaches its lifetime as shown in the cash flow table.

The variable O&M are calculated by Equation 4-51 [72].

$$\begin{aligned}&\text{Variable O \& M in year n}\\&= \text{variable cost by generation (\$/kWh) x}\\&\text{energy generated in year n x}(1 + \text{inflation rate} + \text{escalation rate})\end{aligned} \quad \textbf{....... 4-51}$$

Insurance

The plant insurance applies to all years considered. In this simulation the model calculates the insurance using Equation 4-52 below.

$$\text{Insurance in each year} = \text{Total installed plant cost (\$) x insurance (\%) x } (1 + \text{inflation rate})^{n-1} \quad \textbf{....... 4-52}$$

Debt balance: This is the debt portion of the investment costs.
The debt balance is calculated using Equation 4-53.

$$\text{Debt interest in year, n} = -\text{ debt balance in year n x loan rate} \quad \textbf{....... 4-53}$$

Total debt repayment

This is the total interests on the debts and the principal payments. It is calculated using equation 4-54.

$$\text{Debt total payment} = \text{Debt interest payment} + \text{debt repayment} \quad \textbf{....... 4-54}$$

The following section discusses the after tax cash flow of CSTP plants.

4.15.1 After Tax Cash flow

After tax cash flow is the measure of the financial performance that looks at the project's ability to generate cash flow through its operations. The after tax cash flow is defined by Equation 4-55 [100]:

$$\text{After tax cash flow} = \text{Net income} + \text{depreciation} - \text{non cash charges} \quad \textbf{....... 4-55}$$

where;

Net income: This is the profit obtained from the project through the sale of energy. Net income is the difference between the gross income obtained in a project and the revenues involved in the proper running of the project. These revenues include interest rates charged on borrowed loans, taxes, depreciation, salaries, wages and operation and maintenance costs.

Depreciation: Depreciation is the allocation of cost to fixed assets over their useful lifetime. Depreciation indicates the decrease in value of an asset. In this simulation straight line depreciation of 30 years is used. Straight line depreciation is a method of computing depreciation (amortization) of asset. It is done by dividing the difference between the capital costs and the useful life of a project.

Therefore the per year depreciation is calculated using Equation 4-56 below;

$$\text{Annual depreciation} = \frac{\text{Total capital costs - salvage value}}{\text{number of useful lifetime}} \quad \textbf{....... 4-56}$$

where

Salvage value is the value of a project after its useful life time. In this simulation a salvage value of 10% was used. For CSTP plants, some assets such as land can be sold after 30 years at a higher value than they were acquired at

Number of useful life: This is the number of years or duration during which a project is kept in a productive use in a business. In this simulation the number of useful life assumed is 30 years.

Non-cash charges: These are charges made against earnings. These charges are made against depreciation, amortization and depletion accounts of the projects profit and loss account [72].

The after tax cash flow of a dry cooled parabolic trough CSTP plant is shown in Table 4-29 and Figure 4-30 (*after tax cash flow of a wet cooled plant is shown in Figure 7-3 in Appendix A-1*). In the first years of operation all the equipments involved in generation such as collectors, generators, motors, condensers, TES and HTF are new and hence the operation and maintenance costs are low. As the plant approaches its operational lifetime the net income decreases because of degradation of the core generating components mentioned above. This is because as the main components of the CSTP plant ages, the variable costs, fixed operation and maintenance costs increase and hence reducing the net income earned. There is an increase in the after tax cash flow in the 21st year. It marks the end of payment of interests charged on loan borrowed for a term of 20 years assumed in this model.

Table 4-29 Dry Cooling plant after tax cash flow

0	-3.01958x10^{8}	16	2.00707x10^{8}
1	2.37479x10^{8}	17	1.97072x10^{8}
2	2.63628x10^{8}	18	1.92429x10^{8}
3	2.39386x10^{8}	19	1.86614x10^{8}
4	2.25421x10^{8}	20	1.79437x10^{8}
5	2.26904x10^{8}	21	1.98462x10^{8}
6	2.16552x10^{8}	22	1.88686x10^{8}
7	2.06017x10^{8}	23	1.76846x10^{8}
8	2.06952x10^{8}	24	1.62608x10^{8}
9	2.07636x10^{8}	25	1.45582x10^{8}
10	2.0802510^{8}	26	1.25315x10^{8}
11	2.08068x10^{8}	27	1.01282x10^{8}
12	2.07708x10^{8}	28	7.28748x10^{7}
13	2.06877x10^{8}	29	3.93873x10^{7}
14	2.05495x10^{8}	30	9.06256x10^{5}
15	2.03474x10^{8}		

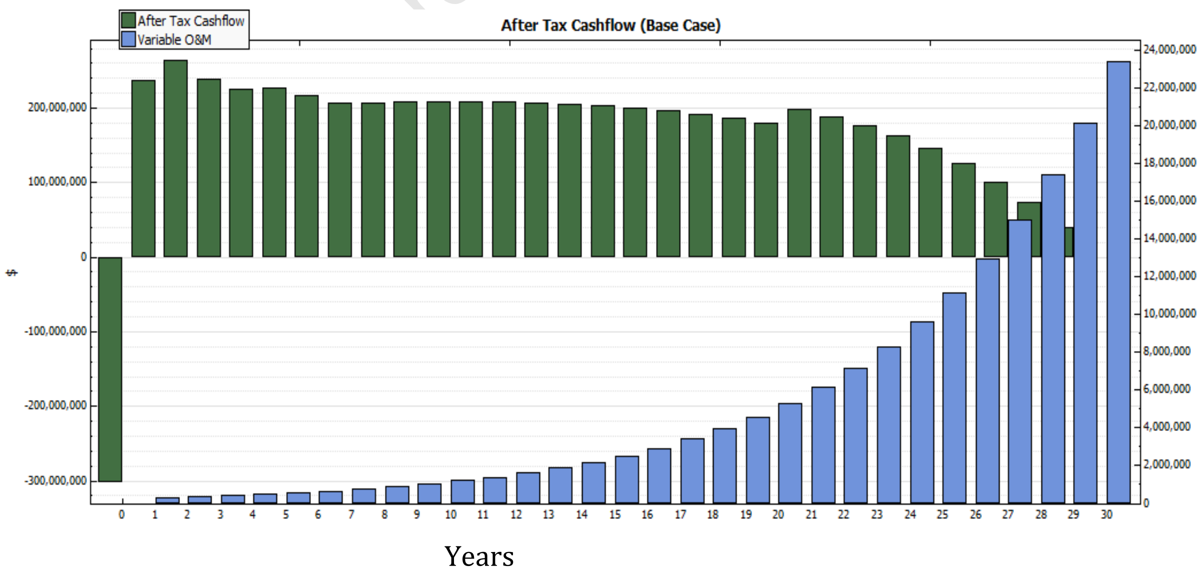

Figure 4-30 Dry cooling After Tax cash flow and variable costs

The following section presents the overview of the results and analysis of the research work.

4.16 Chapter Review

In this chapter wet and dry cooling parabolic troughs are simulated. The performance of a wet cooled plant in terms of energy production is slightly higher than the dry cooled plant. This is due to the fact that water is denser than air and hence has a higher capability of ejecting the exhaust heat on the condenser. Dry cooled plants are affected by high dry bulb temperatures which reduce ITD, hence reducing the fans capability to eject the exhaust heat from the condenser. This in turn reduces the thermal collection efficiency of the dry cooled plant by defocusing some mirrors. In wet cooled plants increasing the approach temperature reduces the energy output of the plant. The reason behind this is that more water is circulated in the condenser to cool a small volume of exhaust steam which increases the parasitic losses such as energy drawn by the pumps to pump water for cooling.

Addition of TES to a plant adds value to a parabolic trough such as spinning reserves and generation shifting. As the TES hours increases the LCOE increases due to land and inventory costs of building the structures. Uniform TES dispatch is more preferred to non uniform TES dispatch because it is able to dispatch the energy stored during the day evenly to all hours of the day. This reduces the amount of energy bought from the grid especially in peak hours when the electricity prices are high.

Water usage is the main drawback of the wet cooled plant especially in arid regions where water is scarce. Wet cooled plants use $5.31m^3$ for each MWh generated compared to a dry cooled plant which consumes $0.319m^3$ per MWh. This makes it hard for deployment of wet cooled CSTP plants in regions without water bodies.

The land use factor for a wet cooled plant is higher than for dry cooled plants due to the extra area occupied by the water ponds for cooling the plant.

At a solar multiple of 2 both dry and wet cooled plants are able to meet the load demand fully for three quarters of the day.

LCOE is sensitive to both inflation and VAT. Higher inflation rate and VAT decreases the value of money. This therefore prompts the increase of LCOE to cover up the effects of inflation and VAT. The after tax cash flow decreases as the years of useful lifetime diminishes. The main reason behind this is as the plant ages operation and maintenance costs increases.

Stability studies have shown that a grid connected system is more stable than a standalone system.

The following chapter discusses the conclusions, recommendations and further work of the research work.

5 Conclusions, Recommendations and Further work

The summary of conclusions and findings in this research work are discussed in the following dimensions: Plant performance capability, Water consumption, land usage, Energy production, LCOE, sensitivity analysis, environmental effects and stability analysis.

The performance criteria for both wet and dry cooling are reviewed in two main categories, i.e. efficiency and capacity factor. The ability of a cooling system to reject heat, wet or dry is dependent on the wet bulb temperature for a wet cooled plant and dry bulb temperature for a dry cooled parabolic plant. The rise of these temperatures results in lowering the ITD in an air cooled condenser (ACC) and lowering the approach temperature in a wet cooled condenser. When ITD decreases, the ability of the fans to dissipate heat on the condenser surface reduces. This is because the hot air is being used to expel hot exhaust steam from the turbine. The fans therefore draw more air to dissipate the exhaust heat from the turbine. This delays the rate of heat exchange between HTF and water. The HTF and storage reaches their maximum charge rate and cannot absorb more heat. The collectors therefore defocus reducing the thermal energy collection efficiency.

In a wet cooled condenser, lowering of the approach temperature lowers the ability of the cooling water to dissipate the exhaust steam from the turbine. This slows down the rate of heat exchange between water and HTF in the heat exchangers, and therefore the HTF reaches its maximum heat capacity. At this point also, TES has reached its maximum charge rate. The collectors defocus to reduce the amount of heat collected. This leads to lower thermal energy collection efficiency.

LCOE is a function of annual net energy and capital costs. At lower ITD the LCOE increases. The reason behind is that more energy is drawn by the fans (parasitic losses) from the plant to cool the condenser. This lowers the output energy and therefore LCOE increases. Lower ITD requires stronger condensers to dissipate the exhaust heat with minimum time possible. However such condensers are capital intensive. The best ITD design of the dry cooled condenser obtained is 15^0C. In a wet cooled plant the same principle applies. When the water in the ponds rises in temperature due to over circulation in the condenser, approach temperature increases which delays the process of heat exchange between water and HTF. This lowers the thermal collection efficiency. At high approach temperatures the pumps draws more energy to pump water to cool a unit mass of steam from the condenser. This decreases the energy production therefore increasing the LCOE. In both dry and wet cooled plants, the efficiency of thermal energy collection was highest in December which has the lowest ambient temperatures.

Capacity factor is the ratio of the actual energy produced by a plant to the hypothetical maximum energy a plant can produce. The overall capacity factor of a dry cooled plant is lower than that of a wet cooled plant. This is due to the lower solar field availability for the dry cooled plant. Solar field availability accounts for the number of hours all the collectors in the plants are tracking the sun. Due to the inefficient cooling in a dry cooled plant the number of hours all the collectors are operational is low. This together with the parasitic losses reduces the net amount of energy produced thereby reducing the capacity factor.

The net energy production of a wet cooled plant is 5.42% higher than a dry cooled plant. The solar electric conversion efficiency of both wet and dry cooled plants was 15.58% and 13.66% respectively. The annual solar electric efficiency is a function of incident energy on the aperture to the total net amount of energy collected per year. The incident energy on the collectors is affected by the availability of a collector to capture the energy. Due to the low cooling efficiency of dry cooled plants collector defocusing is inevitable. This lowers the incident energy falling on the collectors hence the total amount of energy absorbed is smaller. The lower electric conversion efficiency on the dry cooled plant is also associated with high the parasitic losses in its cooling system.

Water consumption is the major drawback of a wet cooled plant. Over 95% of the total water consumed by wet cooled parabolic trough can be eliminated by dry cooling. The cost of water for generating 1MWh is 16.62 times more expensive than used in dry cooling. The amount of water used for generating 1MWh in both wet and dry cooled plants is 5.31m^3 and 0.319m^3 respectively. The amount of money used for exporting water is therefore untenable in locations where water supply is limited. Coal and nuclear plants consume 2.2m^3 and 3.2m^3 of water per MWh energy generated respectively. In comparison with the wet cooled parabolic trough CSTP plant consumes more water per MWh of energy generated due to the lower steam Rankine cycle efficiency and the intermittent nature of the sun which forces the plant to frequently start and stop. The steam Rankine cycle efficiency of a parabolic trough is about 37% compared to 42% for coal and nuclear. Water loss in a wet cooled parabolic trough occurs through evaporation, blow down and drift. Water loss through evaporation is the highest.

CSTP plants occupy big stretches of land. The amount of land used for a wet cooled plant is 12.8% higher than for a dry cooled plant. The reason behind is that the wet cooled plant needs extra land covered by water surface for cooling. However the land use factor of a wet cooled plant is lower than that of a dry cooled plant. Land use factor in CSTP plants refers to the ratio of the total area occupied by the plant to the annual energy produced. Land use factor was 5.41% higher for dry cooling than wet cooling. The reason for higher land use factor in dry cooled plants is that the amount of energy collected annually is low.

Adding TES to both dry and wet cooled plants store the excess energy that would have been wasted. Therefore addition of TES allows for greater solar fields use and as a result more energy is collected. The stored energy is used for shifting generation from periods of low energy demand to periods of high energy demand. TES also eliminates the use of fossil fuels for heating the HTF to maintain steam production at the turbine for electricity generation. Addition of TES also provides spinning reserves. Plants without TES have higher LCOE due to lower energy production. However as the number of TES hours increase the LCOE increases. This is because of the added inventory cost of TES structures and land acquisition. Unlike the LCOE which decreases with TES hours, capacity factor increases. The reason is at higher number of TES hours more energy is generated by the plant.

The amount of energy stored during the day can be uniformly dispatched or non-uniformly dispatched. Uniform energy dispatch distributes energy equally to all hours of the day, while non uniform dispatch distributes energy according to demand. The amount of energy stored for both dry and wet cooled plants was able to meet the load from 6pm to 11pm for non-uniform TES dispatch. Using uniform TES dispatch the amount of energy dispatched from TES per hour from wet and dry cooled plant is 8607.75kWh and 8495kWh respectively. In this case, TES dispatch is used as a base load power plant. Uniform TES dispatch has been preferred to non uniform TES dispatch because it reduces the amount of hourly energy bought from the grid especially in the peak tariff hours. At a solar multiple of 2, a wet cooled CSTP plant of 50MWe, 12 hours of TES is able to fully meet the load for three quarters of the day supplying a net amount of 300,277.3kWh to the grid. At a solar multiple of 1 and 1.5 the plant supplied 65,561kWh and 223,214kWh to the grid in the 1st day of January 2012. At lower solar multiple values the plant relies on the grid support for a greater fraction of the day.

Inflation increases the NPV and LCOE. This increases the costs of good services in the plant operation. For example with inflation rate, the spare parts of the CSTP plant goes higher. The plant operator must therefore increase the price of electricity to cover up the high cost of operation and maintenance. Net present value is the sum of net present values of all cash flows. Inflation increases the in and out cash flows, hence the NPV also increases. The increase of the value added tax (VAT) also increases the LCOE and NPV.

After tax cash flow measures the financial performance of a project. The components of after tax cash flow include net income, depreciation and non cash charges. As the years of estimated lifetime nears, the after tax cash flow decays. In a parabolic trough revenues are earned by selling the generated selling energy. Depreciation of equipments such as collectors and motors increases the O& M which reduces the net income generated. The after tax cash flow is also affected by other factors such as interest on loan borrowed and tax.

The real power, reactive power, voltage and frequency restoration after a disturbance is quick in grid connected CSTP synchronous generator than standalone CSTP. A grid acts like an infinite battery storage to restore back the variations of any of the parameters at the least time possible.

The amount of carbon dioxide that can be avoided by installation of a 50MWe CSTP plant in a rural community consisting of 17,000 households is 257,507,500kg. This further prevents deforestation of $1.05x10^6 m^2$ of forests per year.

The dry and wet cooling of the parabolic trough has the following effects to the environment.

i. Noise

The noise in the wet cooled plant originates from the water falling on the cooling tower, fans and air motion. Dry cooled plants noise originates from the massive fans blowing air in the condenser. This noise is undesirable to the surroundings.

ii. Biological impacts

Wet cooling of the parabolic trough poses a great threat to the animals and plants that live in water. Any rise in temperature of water may terminate their lifecycle. As water is re-circulated into the condenser and the water body, its temperature is raised and its purity decreased, and may pose danger to the living organisms in the water. This is also brought about by the chemicals used to make the water pure for cooling which becomes concentrated after sometime thus requiring some water to be drained away to remove the particulates and salts.

Dry cooling blows and kills small insects and birds in the condenser.

iii. Pollution

The possibility of polluting the environment is higher for wet cooling than dry cooling. This is because the use of water has effects such as plume production, blow down, drift and evaporation. These have negative effects on the waste management, water discharge, hazardous materials, public health and soil for agriculture.

Table 5-1 summarizes the advantages and disadvantages of dry and wet cooling parabolic troughs.

Table 5-1 Characteristics Drawn from Wet and Dry Cooling Methods

Cooling Types	**Advantages**	**Disadvantages**
Wet cooling	• Lowest installed cost • Low parasitic loads • Gives high power cycle efficiency	• Higher water consumption • Water treatment and blow down disposal required which is an additional cost
Dry cooling	• low water consumption • No water treatment required	• More expensive equipment • Higher parasitic loads • Poorer cooling at high dry bulb temperatures.

5.2 Recommendations

In future energy problems in Kenya should be solved by first determining the potential of the available sources of energy versus the exploited energies. More emphasis should be based on increasing renewable energies by utilising available sources of solar energy and waste land.

The Energy Regulatory Commission of Kenya should introduce Renewable energy feed in Tariff (REFIT) to promote the stakeholders of CSTP energy. The government of Kenya should provide incentives through capital grants and reduce the cost of exporting energy from CSTP plants.

5.3 Scope for Further work

Significant efforts and integrity was devoted to this project with the sole aim to ensure an accurate and complete analysis of the topic. Yet, due to time constraints, other areas of interests were regrettably left out of the analysis.

Research and development should be done on the following:

i. **Investigate applicability of a single tank thermo-cline storage system**

The cost of a TES system depends on the storage material; the type of heat exchanger used for charging and discharging the system and the cost of land/space used. In this thesis a two tank indirect storage was used. This type of storage has a higher land usage hence adding to the total capital cost of the CSTP plant. Other storage types such as single tank thermo-cline system should be investigated. A single thermo-cline storage system utilises one tank. This therefore reduces the number of heat exchangers used, hence reducing the amount of capital used to assemble the structure and maintain it. Thermo-cline system depends on the density and temperature of the storage fluid. The difference in temperatures and densities of the hot and cold fluid stratifies the thermal storage media and forms a thermo-cline section between the two temperature gradients. This is illustrated in Figure 5-1.

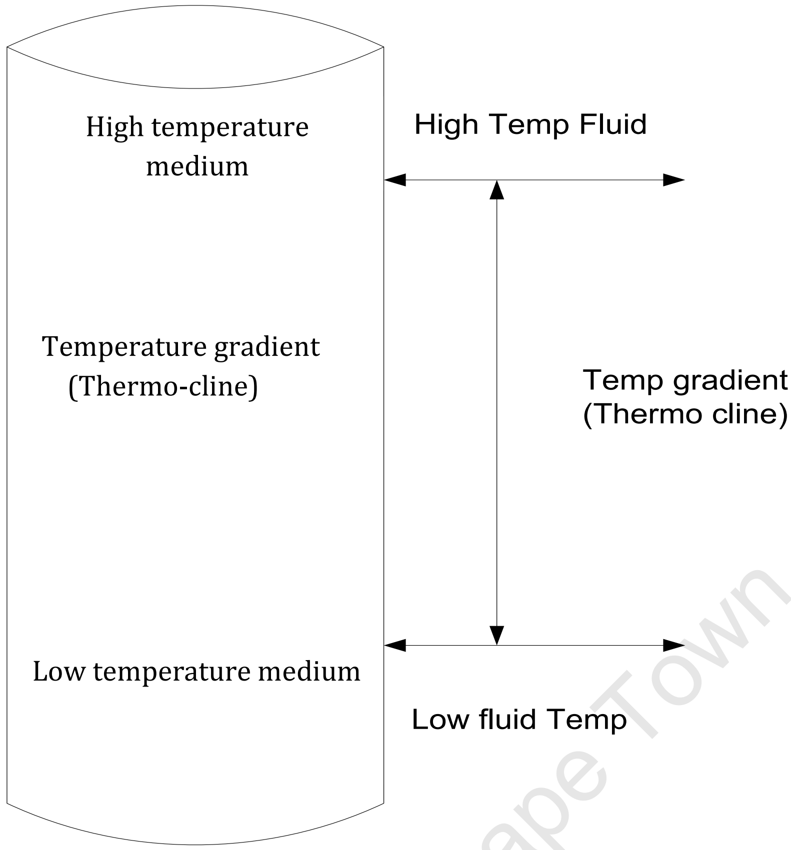

Figure 5-1 Single tank Thermo-cline system

ii. Better cleaning techniques for mirrors to reduce dust and improve efficiency

The reflectivity and absorbance degree of the mirrors are the main factors that affect the energy production of a parabolic trough CSTP plant. The reflectivity of the collectors is heavily affected by soiling caused by the wind. The methods applied in collector cleaning of the parabolic troughs are wet brushing and water jet cleaning. They both use water for cleaning the collectors. Further research and development should be done to investigate the economic viability of dry cleaning the collectors and absorbers using air. This will reduce the amount of water used in parabolic troughs which is the major constraint in the arid areas.

iii. Investigate the effects of incorporating hybrid systems

Huge interest has been generated in the prospect of hybrid systems incorporating two renewable technologies such as biomass and parabolic trough to increase the renewable energy fraction. This would help reduce the costs of TES used because the biomass plant will run throughout hence little thermal energy storage will be required. This would also reduce the solar field area and hence the cooling system.

iv. Investigate other steam Rankine systems

Research should be carried out to investigate the applicability of the Brayton high temperature steam Rankine cycle into the parabolic trough to reduce losses incurred during cooling. This will make dry cooling more suitable especially in desert areas where water is the main problem for the parabolic trough deployment.

v. **Hybrid Wet-Dry Cooling**

Investigate the application of the hybrid wet-dry cool in parabolic trough CSTP plants by studying its performance cost and its environmental effects.

vi. **Comparison of Parabolic Trough CSTP plant with other CSTP technologies**

Although parabolic trough has a good track record of performance over other CSTP technologies research should be done to investigate the performance of other CSTP technologies. As was reported in the literature review parabolic dish (for example) is most suitable for remote and water scarce areas.

vii. **Low power fans**

Investigate the different types of fans that can be applied in dry cooling to reduce the amount of fan power consumption.

viii. **Hybrid fans that use parabolic trough generated energy and wind**.

Investigate the potentiality of hybrid parabolic trough with wind fans for dry cooling. This would reduce the amount of power drawn by the fans. It will therefore increase the annual energy production.

6 References

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7 Appendix

7.1 Appendix A1: Results of SAM

Table 7-1 Energy stored at Different SM values for Wet Cooled Plant (kWh)

Hours	Energy (kWh) SM=2	Energy (kWh) SM=1.5	Energy (kWh) SM=1
0	0	0	0
1	0	0	0
2	0	0	0
3	0	0	0
4	0	0	0
5	0	0	0
6	0	0	0
7	0	0	0
8	0	0	0
9	9,493.69	0	0
10	8,193	0	0
11	2,684.63	0	0
12	41,412.2	0	0
13	53,401.1	4,749.83	0
14	56,717.6	6846.63	0
15	30,784.5	0	0
16	0	0	0
17	0	0	0
18	0	0	0
19	0	0	0
20	0	0	0
21	0	0	0
22	0	0	0
23	0	0	0
Net Stored	**202,786**	**11,596**	**0.00**

Table 7-2 Energy stored at Different SM values for a dry Cooled Plant (kWh)

Hours	Energy (kWh) SM=2	Energy (kWh) SM=1.5	Energy (kWh) SM=1
0	0	0	0
1	0	0	0
2	0	0	0
3	0	0	0
4	0	0	0
5	0	0	0
6	0	0	0
7	0	0	0
8	0	0	0
9	9,493.69	0	0
10	7,959.98	0	0
11	2,537.41	0	0
12	41,401.9	0	0
13	53,470.7	4,241.24	0
14	56,588.4	6,767.37	0
15	30,745.9	0	0
16	0	0	0
17	0	0	0
18	0	0	0
19	0	0	0
20	0	0	0
21	0	0	0
22	0	0	0
23	0	0	0
Net stored	**202,197**	**11,000**	**0.00**

Table 7-3 Energy supplied to and from the grid at the 1st day of January 2012 at different SM values for a Dry cooled plant without grid or TES support

Hour	Energy supplied to or from the grid, wet cooled plant (kWh) SM=1	Energy supplied to or from the grid, wet cooled plant (kWh) SM=1.5	Energy supplied to or from the grid, wet cooled plant (kWh) SM=2	Load profile kWh
0	-20,060	-20,060	-20,060	20,060
1	-20,060	-20,060	-20,060	20,060
2	-20,060	-20,060	-20,060	20,060
3	-20,060	-20,060	-20,060	20,060
4	-20,060	-20,060	-20,060	20,060
5	-20,060	-20,060	-20,060	20,060
6	-27,880	-27,880	-27,880	27,880
7	-1,360	-1,360	-1,360	1,360
8	3,609.1	13,289.1	21,641.1	1,360
9	24,436.7	39,569.9	50,410	1,360
10	24,551.1	38,747.5	49,939.5	1,360
11	23,230.3	36,947.2	49,673.8	1,360
12	30,580.6	46,963.7	48,475.8	1,360
13	32,551	47,773.2	48,094.1	1,360
14	32,949.8	47,633	47,942	1,360
15	28,214.3	43,622.6	48,580.1	1,360
16	16,919	27,720.6	46,616.7	1,360
17	-13,430	52.1	26,640	1,3430
18	-35,190	-24,190	0	35,190
19	-52,190	-52,190	0	52,190
20	-35,190	-35,190	0	35,190
21	-35,190	-35,190	0	35,190
22	-35,190	-35,190	0	35,190
23	-35,190	-35,190	0	35,190
Totals	**-174,128**	**-24,421**	**288,413**	**403,410**

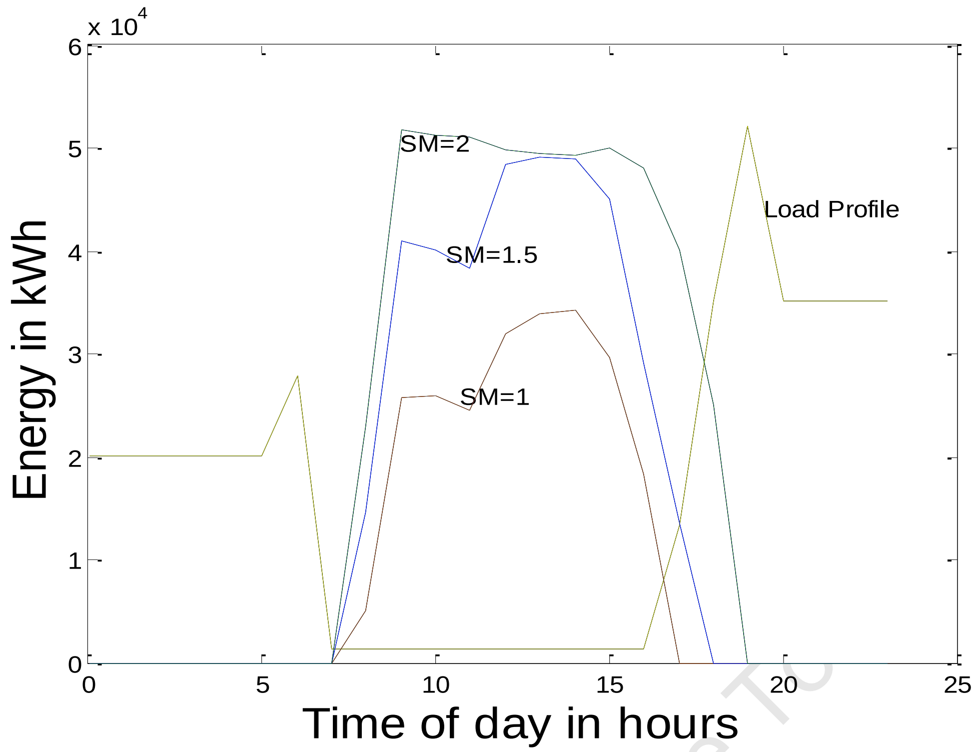

Figure 7-1 Load Profile against Energy generated by Varying SM values for dry cooled Plant without grid or TES support

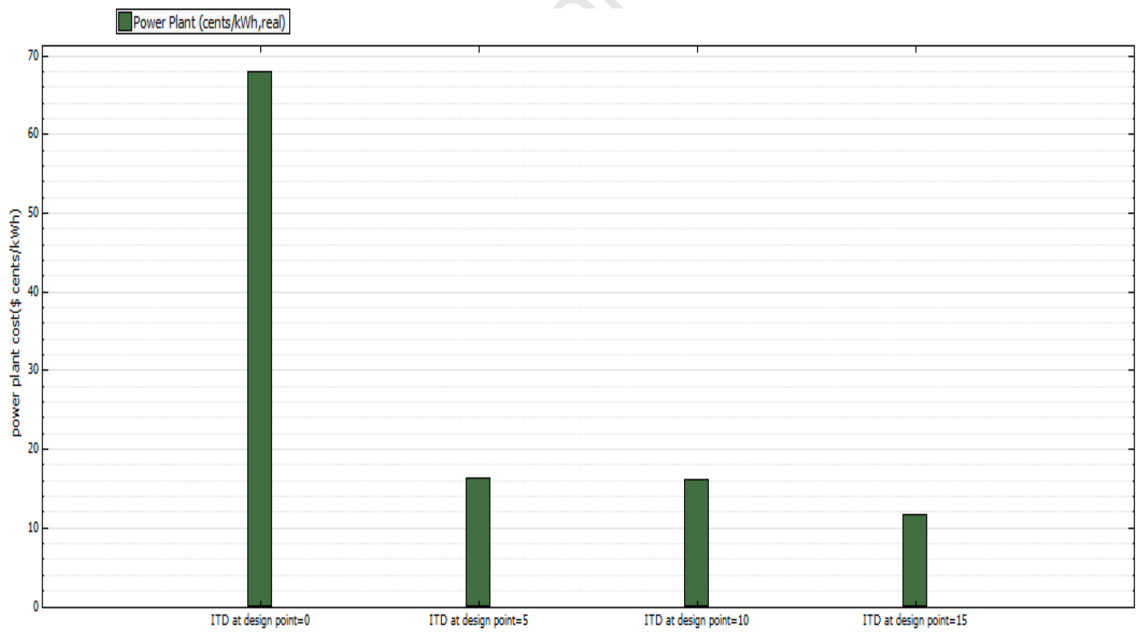

Figure 7-2 Variation of Plant cost with ITD

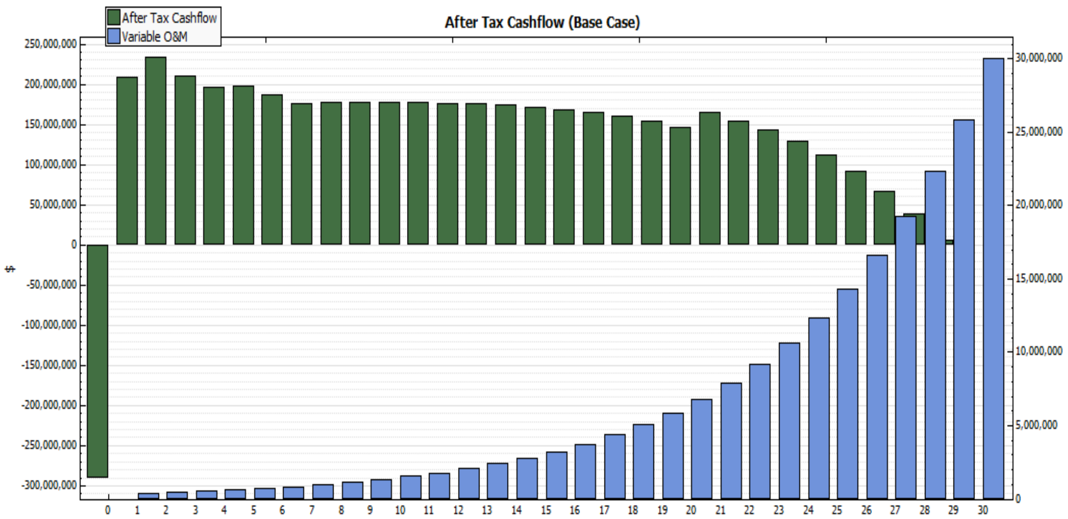

Figure 7-3 After Tax Cash flow for wet cooling

Table 7-4 Cash Flow, wet cooling parabolic trough CSTP plant

Year	0	1	2	3	4	5	6
Energy (kWh)	0	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558
Energy Price ($/kWh)	0	2.167	2.193	2.22	2.246	2.273	2.301
Energy Value ($)	0	293,138,581.97	296,656,244.96	300,216,119.9	303,818,713.33	307,464,537.89	311,154,112.35
Operating expenses							
Fixed O&M	0	3,465,000	4,019,400	4,662,504	5,408,504.64	6,273,865.38	7,277,683.84
Variable O&M	0	405,751.67	470,671.94	545,979.45	633,336.17	734,669.95	852,217.15
Insurance	0	2,524,234.49	2,928,112.01	3,396,609.93	3,940,067.52	4,570,478.32	5,301,754.85
Property Assessed Value	0	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1
Net Salvage Value	0	0	0	0	0	0	0
Total Operating Expenses	0	6,394,986.17	7,418,183.95	8,605,093.38	9,981,908.33	11,579,013.66	13,431,655.84
Operating Income	0	286,743,595.81	289,238,061	291,611,026.51	293,836,805.01	295,885,524.24	297,722,456.51
Financing							
Debt Balance	0	-257,446,675.69	-251,974,807.82	-246,051,510.87	-239,639,541.91	-232,698,585.51	-225,185,000.21
Debt Interest Payment	0	21,239,350.74	20,787,921.65	20,299,249.65	19,770,262.21	19,197,633.3	18,577,762.52
Debt Repayment	0	5,471,867.86	5,923,296.96	6,411,968.96	6,940,956.4	7,513,585.3	8,133,456.09
Debt Total Payment	0	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6
Tax Effect on Equity (State)							
State Depreciation Schedule (%)	0	3.33	3.33	3.33	3.33	3.33	3.33
Depreciation	0	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48
State Income Taxes	0	70,430,825.44	72,821,437.85	71,928,534.51	71,679,220.73	72,402,712.82	72,323,568.58
State Tax Savings	0	-70,430,825.44	-72,821,437.85	-71,928,534.51	-71,679,220.73	-72,402,712.82	-72,323,568.58
	0	177,125,354.31	182,252,020.84	179,679,907.71	178,569,188.13	179,684,707.42	178,880,166.74

Continuation of cash flow

Year	0	7	8	9	10	11	12
Energy (kWh)	0	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558
Energy Price ($/kWh)	0	2.328	2.356	2.384	2.413	2.442	2.471
Energy Value ($)	0	314,887,961.7	318,666,617.24	322,490,616.64	326,360,504.04	330,276,830.09	334,240,152.05
Operating expenses							
Fixed O&M	0	8,442,113.26	9,792,851.38	11,359,707.6	13,177,260.82	15,285,622.55	17,731,322.16
Variable O&M	0	988,571.89	1,146,743.39	1,330,222.33	1,543,057.91	1,789,947.17	2,076,338.72
Insurance	0	6,150,035.63	7,134,041.33	8,275,487.94	9,599,566.02	11,135,496.58	12,917,176.03
Property Assessed Value	0	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1
Net Salvage Value	0	0	0	0	0	0	0
Total Operating Expenses	0	15,580,720.78	18,073,636.1	20,965,417.88	24,319,884.74	28,211,066.3	32,724,836.9
Operating Income	0	299,307,240.92	300,592,981.14	301,525,198.77	302,040,619.3	302,065,763.8	301,515,315.15
Financing							
Debt Balance	0	-217,051,544.12	-208,247,077.91	-198,716,243.23	-188,399,114.69	-177,230,823.05	-165,141,147.35
Debt Interest Payment	0	17,906,752.39	17,180,383.93	16,394,090.07	15,542,926.96	14,621,542.9	13,624,144.66
Debt Repayment	0	8,804,466.21	9,530,834.68	10,317,128.54	11,168,291.64	12,089,675.7	13,087,073.95
Debt Total Payment	0	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6
Tax Effect on Equity (State)							
State Depreciation Schedule (%)	**0**	3.33	3.33	3.33	3.33	3.33	3.33
Depreciation	0	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48
State Income Taxes	0	72,188,945.99	72,744,287.98	73,218,597.16	73,595,774.24	73,857,016.12	73,980,374.21
State Tax Savings	0	-72,188,945.99	-72,744,287.98	-73,218,597.16	-73,595,774.24	-73,857,016.12	-73,980,374.21
Net cash		177,895,037.24	178,464,466.77	178,784,894.31	178,813,811.07	178,501,991.4	177,792,428.69

Continuation of cash flow

Year	0	13	14	15	16	17	18
Energy (kWh)	0	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558
Energy Price ($/kWh)	0	2.501	2.531	2.561	2.592	2.623	2.655
Energy Value ($)	0	338,251,033.88	342,310,046.28	346,417,766.84	350,574,780.04	354,781,677.4	359,039,057.53
Operating expenses							
Fixed O&M	0	20,568,333.7	23,859,267.09	27,676,749.83	32,105,029.8	37,241,834.57	43,200,528.1
Variable O&M	0	2,408,552.91	2,793,921.38	3,240,948.8	3,759,500.61	4,361,020.71	5,058,784.02
Insurance	0	14,983,924.2	17,381,352.07	20,162,368.4	23,388,347.34	27,130,482.92	31,471,360.18
Property Assessed Value	0	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1
Net Salvage Value	0	0	0	0	0	0	0
Total	0	37,960,810.81	44,034,540.54	51,080,067.02	59,252,877.75	68,733,338.19	79,730,672.3

Operating Expenses							
Operating Income	0	300,290,223.07	298,275,505.75	295,337,699.81	291,321,902.29	286,048,339.21	279,308,385.23
Financing							
Debt Balance	0	-152,054,073.4	-137,887,315.85	-122,551,800.81	-105,951,105.77	-87,980,853.39	-68,528,055.19
Debt Interest Payment	0	12,544,461.06	11,375,703.56	10,110,523.57	8,740,966.23	7,258,420.4	5,653,564.55
Debt Repayment	0	14,166,757.55	15,335,515.05	16,600,695.04	17,970,252.38	19,452,798.2	21,057,654.05
Debt Total Payment	0	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6	26,711,218.6
Tax Effect on Equity (State)							
State Depreciation Schedule (%)	**0**	3.33	3.33	3.33	3.33	3.33	3.33
Depreciation	0	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48
State Income Taxes	0	73,940,241.47	73,706,756.56	73,245,111.8	72,514,749.51	71,468,428.75	70,051,141.66
State Tax Savings	0	-73,940,241.47	-73,706,756.56	-73,245,111.8	-72,514,749.51	-71,468,428.75	-70,051,141.66
Net cash	0	176,619,102.03	174,905,546.41	172,563,195.31	169,489,459.29	165,565,498.36	160,653,639.31

Continuation of cash flow

Year	0	19	20	21	22	23	24
Energy (kWh)	0	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558
Energy Price ($/kWh)	0	2.686	2.719	2.751	2.784	2.818	2.852
Energy Value ($)	0	363,347,526.22	367,707,696.54	372,120,188.89	376,585,631.16	381,104,658.74	385,677,914.64
Operating expenses							
Fixed O&M	0	50,112,612.59	58,130,630.61	67,431,531.51	78,220,576.55	90,735,868.79	105,253,607.8
Variable O&M	0	5,868,189.46	6,807,099.78	7,896,235.74	9,159,633.46	10,625,174.81	12,325,202.78
Insurance	0	36,506,777.81	42,347,862.26	49,123,520.22	56,983,283.46	66,100,608.81	76,676,706.22
Property Assessed Value	0	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1
Net Salvage Value	0	0	0	0	0	0	0
Total Operating Expenses	0	92,487,579.87	107,285,592.65	124,451,287.47	144,363,493.46	167,461,652.42	194,255,516.8
Operating Income	0	270,859,946.35	260,422,103.89	247,668,901.43	232,222,137.7	213,643,006.32	191,422,397.84
Financing							
Debt Balance	0	-47,470,401.14	-24,675,490.63	0	0	0	0
Debt Interest Payment	0	3,916,308.09	2,035,727.98	0	0	0	0
Debt Repayment	0	22,794,910.51	24,675,490.63	0	0	0	0
Debt Total Payment	0	26,711,218.6	26,711,218.6	0	0	0	0
Tax Effect on Equity (State)							
State Depreciation Schedule (%)	**0**	3.33	3.33	3.33	3.33	3.33	3.33
Depreciation	0	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48

State Income Taxes	0	68,198,855.32	65,837,050.91	62,879,027.95	58,615,721.16	53,487,880.9	47,354,992.96
State Tax Savings	0	-68,198,855.32	-65,837,050.91	-62,879,027.95	-58,615,721.16	-53,487,880.9	-47,354,992.96
Net cash	0	154,594,381.37	147,202,924.31	164,976,361.36	155,028,645.52	143,063,684.91	128,753,613.05

Continuation of cash flow

Year	0	25	26	26	28	29	30
Energy (kWh)	0	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558	135,250,558
Energy Price ($/kWh)	0	2.886	2.92	2.955	2.991	3.027	3.063
Energy Value ($)	0	390,306,049.62	394,989,722.21	399,729,598.88	404,526,354.06	409,380,670.31	414,293,238.36
Operating expenses							
Fixed O&M	0	122,094,185.05	141,629,254.66	164,289,935.4	190,576,325.07	221,068,537.08	256,439,503.01
Variable O&M	0	14,297,235.23	16,584,792.86	19,238,359.72	22,316,497.28	25,887,136.84	30,029,078.74
Insurance	0	88,944,979.22	103,176,175.89	119,684,364.03	138,833,862.28	161,047,280.24	186,814,845.08
Property Assessed Value	0	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1	504,846,898.1
Net Salvage Value	0	0	0	0	0	0	50,484,689.81
Total Operating Expenses	0	225,336,399.49	261,390,223.41	303,212,659.16	351,726,684.62	408,002,954.16	422,798,737.02
Operating Income	0	164,969,650.12	133,599,498.8	96,516,939.72	52,799,669.44	1,377,716.15	-8,505,498.66
Financing							
Debt Balance	0	0	0	0	0	0	0
Debt Interest Payment	0	0	0	0	0	0	0
Debt Repayment	0	0	0	0	0	0	0
Debt Total Payment	0	0	0	0	0	0	0
Tax Effect on Equity (State)							
State Depreciation Schedule (%)	**0**	3.33	3.33	3.33	3.33	3.33	3.33
Depreciation	0	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48	18,258,629.48
State Income Taxes	0	40,054,034.59	31,395,872.82	21,161,086.52	9,095,119.92	-5,097,339.19	-7,825,106.48
State Tax Savings	0	-40,054,034.59	-31,395,872.82	-21,161,086.52	-9,095,119.92	5,097,339.19	7,825,106.48
Net cash	-2.9×10^{8}	111,718,043.52	91,515,666.07	67,634,498.02	39,480,575.96	6,364,838.04	477,100

7.2 Appendix A2 –Results of DIg SILENT

7.2.1 Impacts of a 3 phase on Speed of a CSTP synchronous generator when operated as a standalone and grid connected modes

The speed of a rotor restores back to 1pu after about 90s for standalone system as shown in Figure 7-4. In grid connected it took about 7s as shown in Figure 7-5. This therefore explains that a grid connected is more stable than standalone CSTP synchronous generator.

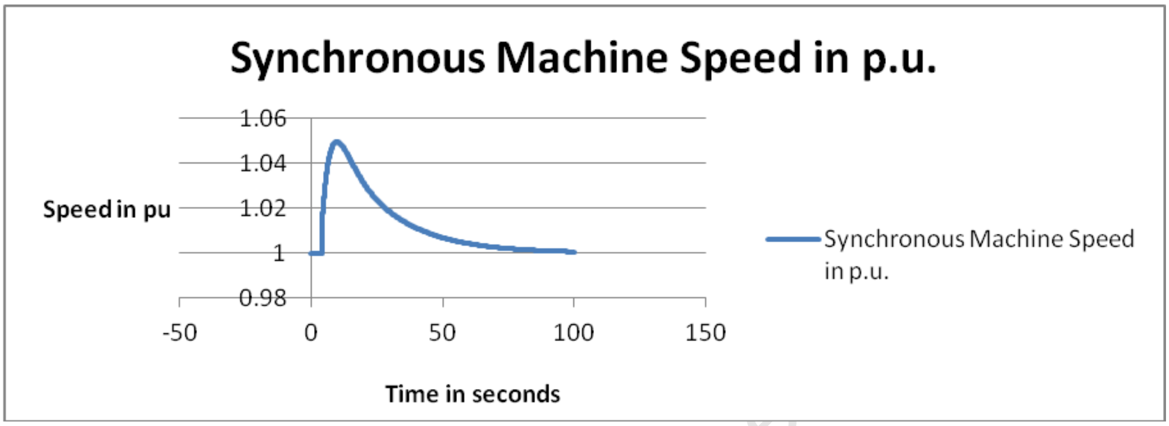

Figure 7-4 Speed of Rotor, Standalone

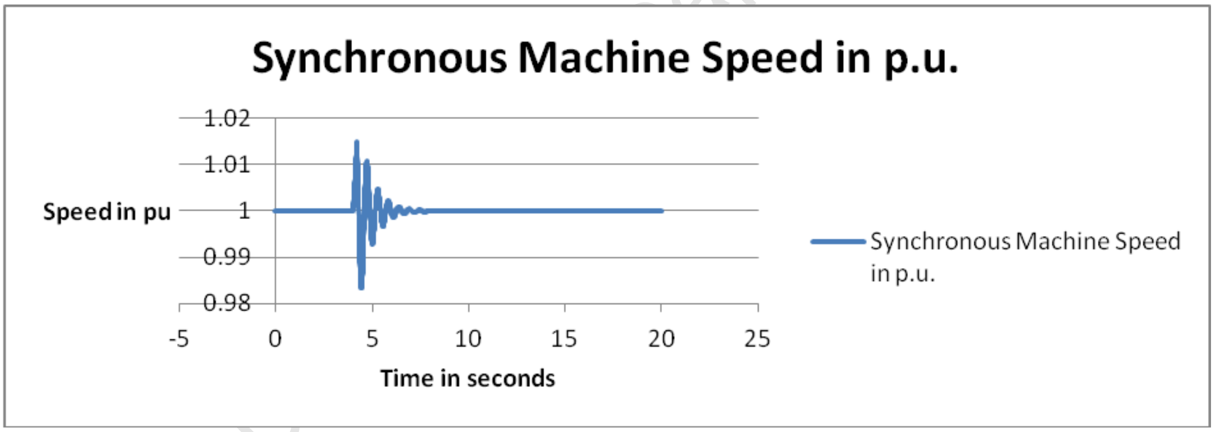

Figure 7-5 Speed of Rotor, Grid connected

7.3 System isolated from the main Grid (Standalone)

Table 7-5 Results of a Standalone system (Obtained from boxes of Figure 7-6)

Components	MW losses(MVAr)		MVAr	loading(%)	Voltage levels
Bus bar1	0.00	0.00	0.00		11kv,1pu,0.00degrees
Bus bar 2	10,10,10,20	5,5,5,5	0.00		0.23kv,0.99pu,-0.78degrees
Generator	50.17	20.87	54.34		11kv
TFM1	50.17	20.87	54.34	0.17	11/0.23kv

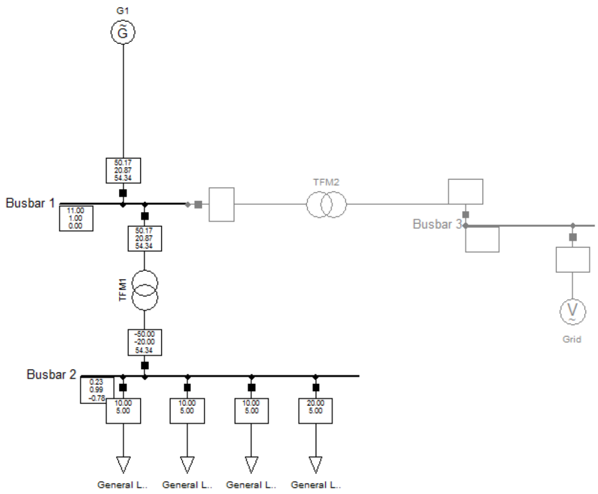

Figure 7-6 Results of a Standalone system

- DIgSILENT /info-element G1 is local reference in separated area of Bus bar 1
- DIgSILENT /info-calculating load flow
- DIgSILENT /info
- DIgSILENT /info start Newton Raphson Algorithm….
- DIgSILENT /info load flow iteration 1
- DIgSILENT /info load flow iteration 2
- Newton Raphson converged with 2 iterations
- Load flow calculation successful;

7.4 System connected to the Grid

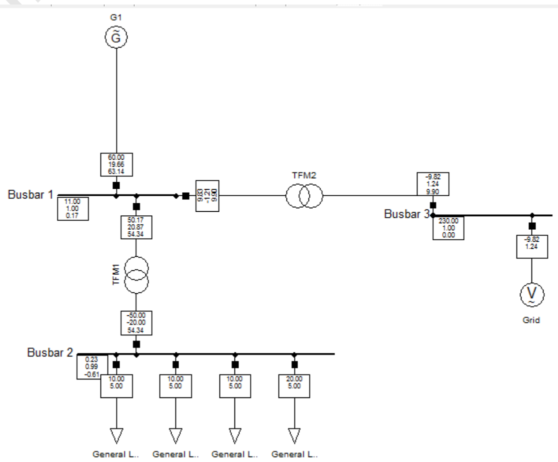

Figure 7-7 Grid connected system results

Table 7-6 Results of a Grid connected system (Obtained from boxes of Figure 7-7)

Components	MW	MVAr	Loading (%)	Losses (MW)	Voltage levels
Bus bar1	0.00	0.00	0.00	0.00	11kv,1pu,0.17degrees
Bus bar 2	10,10,10,20	5,5,5,5	0.00	0.00	0.23kv,0.99pu,-0.61degrees
Bus bar3	0.00	0.00	0.00	0.00	230kv,1pu,0.00degrees
Generator	60.00	19.66	63.14	0.00	11kv
Grid	-9.82	-1.24	0.00	0.00	230kv,1pu
TFM1	50.17	20.87	54.34	0.17	11/0.23kv
TFM2	9.83	-1.21	9.90	0.01	11/230kv

- DIgSILENT/info-element Grid is local reference in separated area of Bus bar 3
- DIgSILENT /info-calculating load flow
- DIgSILENT /info
- DIgSILENT /info start Newton Rap son Algorithm….
- DIgSILENT /info load flow iteration 1
- DIgSILENT /info load flow iteration 2
- Newton Raphson converged with 2 iterations
- Load flow calculation successful;

CURRICULUM VITAE

S.Kibaara was born in Kenya. After completing his high school at Nyandarua High School in 2003, he joined the University of Nairobi. He received his BSc Electrical and Electronic Engineering in 2010. He joined the University of Cape Town in 2011 where he did research on the concentrating solar thermal power under the supervision of Dr Sunetra Chowdhury and Professor SP Chowdhury. He was also a teaching assistant in electrical power systems.
In the support of this work the following papers were published by him.

Papers Published

1. S.K.Kariuki, G.T. Machinda, S. Chowdhury, S. P. Chowdhury, "Power Engineering Education: Kenyan Perspective", *45th Universities Power Engineering Conference*, Soest, Germany, pp1-7, September 2011.
2. S.K.Kariuki, G.T. Machinda, S. Chowdhury, S. P. Chowdhury, "Generation of Electricity from Biomass", *South Africa University Power Engineering Conference*, University of Cape Town, South Africa, pp1-7,July 2011.
3. T .Mashau, S. Kibaara, S.Chowdhury, S.P.Chowdhury, "Impact of Distributed Generation on Protection Coordination in a Radial Distribution Feeder," 45th *Universities Power Engineering Conference*, Soest, Germany, pp1-7, September 2011.
4. GT Machinda,S. Kibaara, S.Chowdhury,SP Chowdhury, "Concentrating Solar Thermal Power Technologies", *Annual IEEE India Conference, Indicon2011*, Bits Pilani ,pp1-7, December 2011..
5. S.K.Kariuki, S. Chowdhury, S. P. Chowdhury , " Solar Multiple Optimization and Dispatch Analysis of A Potential Parabolic CSP Trough In Kenya",*Paper accepted awaiting Publication at IEEE PES Transmission and Distribution Conference and Exposition*, Orlando Florida,pp1-7, May 2012.
6. S.K.Kariuki, S. Chowdhury, S. P. Chowdhury, "A Thermal analysis of Parabolic Trough CSP and Biomass Hybrid Power System", *Paper accepted awaiting Publication at IEEE PES Transmission and Distribution Conference and Exposition*, Orlando Florida, pp1-7,May 2012.
7. M.Ntshangase, S.K.Kariuki, S. Chowdhury, S.P.Chowdhury, "Voltage Stability Analysis of Electricity Networks with DFIG-based Wind Power Plants to the Submission Site", *Paper accepted awaiting Publication at IEEE PES General Meeting*, Santiago, California, pp1-7, July 2012.