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"THERMODYNAMIC ANALYSIS OF SOLAR-AIDED THERMAL POWER PLANT"

Submitted to Delhi Technological University in partial fulfillment of the requirement for the award of Degree of

Master of Technology

In

Thermal Engineering

By

ABHISHEK SEN

2K13/THE/01

Under the Supervision of

Dr. K . Manjunath

Asst. Professor

Department Of Mechanical Engineering

Delhi Technological University

Delhi-110042

Department of Mechanical Engineering
Delhi Technological University
(Formerly Delhi College of engineering)
Bawana Road, Delhi-110042

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I **Abhishek Sen**, hereby certify that the work which is being presented in the major project-II entitled "**Thermodynamic analysis of solar aided thermal power plant"** is submitted in the partial fulfillment of the requirements for the degree of **M.Tech** at **Delhi Technological University** is an authentic record of my own work carried under the supervision of **Dr K. Manjunath** and **Mr.Nausad Ansari**. I have not submitted the matter embodied in this major project-II for the award of any other degree. Also it has not been directly copied from any source without giving its proper reference.

ABHISHEK SEN

M. Tech (Thermal Engineering)

2K13/THE/01

CERTIFICATE

DELHI TECHNOLOGICAL UNIVERSITY
(Formerly DELHI COLLEGE OF ENGINEERING)

Date:-__________

This is to certify that report entitled **"THERMODYNAMIC ANALYSIS OF SOLAR-AIDED THERMAL POWER PLANT"** by **ABHISHEK SEN** is the requirement of the partial fulfillment for the award of Degree of **Master of Technology (M.Tech)** in **Thermal Engineering** at **Delhi Technological University**. This work was completed under our supervision and guidance. He has completed his work with utmost sincerity and diligence. The work embodied in this project has not been submitted for the award of any other degree to the best of our knowledge.

SUPERVISORS

Dr. K. Manjunath	**Mr. Nausad A. Ansari**
Asst. Professor	**Asst. Professor**
Department of Mechanical Engineering	**Department of Mechanical Engineering**
DELHI TECHNOLOGICAL UNIVERSITY	**DELHI TECHNOLOGICALUNIVERSITY**

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ABHISHEK SEN

(Roll No-2K13/THE/01)

ABSTRACT

In these days when every country in the world are very much serious about the economic and environmental issues, every country wants to grow up economically with increase of energy consumption and wants to keep the environment pollution free. At this position use of only coal based thermal power plant is not significant. Seeing the harmful effect to environment it is going to be completely obsolete in coming hundred years. At these days innovative techniques and better technical solutions are coming up to reduce pollution due to emission from any type of plants or vehicles. Which means many research works to reintegration of these thermal power plants with alternative renewable energy sources are growing up at this moment.

Out of many solutions this work indicates the possible way of generation of power from a coal based thermal power plant integrated with indirect solar heating by using solar heater to increase the annual power output from the steam power plant. Parabolic trough solar heaters are installed in position of water circuit to increase the quality of the steam generated. The net power output is calculated by commercially available 'EES' software upon mass & energy balance to each and every component of solar aided steam power plant. Annual power output of the solar aided thermal power plant is calculated and shows that although the investment cost for power generation is increased but the annual coal consumption is become decreased which saves fossil fuels and extra invested cost can be compensated within some years.

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NOMENCLATURE

APH	Air Pre-heater
BFP	Boiler Feed Pump
BRH	Bottom Ring Header
CEP	Condensate Extracted Pump
CRS	Central Receiver System
CSP	Concentrating Solar Power
D/C	Down Comer
DE	Dish/ Engine System
DSG	Direct Steam Generation
FD	Force Draft
GHG	Green House Gases
HPT	High Pressure Turbine
HRSG	Heat Recovery Steam Generator
HSGTPP	Hybrid Solar-Gas Thermal Power Plant
HSPP	Hybrid Solar Power Plant
HTF	Heat Transfer Fluid
IPT	Intermediate Pressure Turbine
ISCCS	Integrated solar combined cycle system
LF	Linear Fresnel
NPSH	Net Positive Suction Head
LPT	Low Pressure Turbine
PTC	Parabolic Trough Collector
SAPG	Solar Aided Power Generation
SAPS	Solar Aided Power System
SCA	Solar Collector Assembly
SEGS	Solar electricity generating system
SFEE	Steady Flow Energy Equation

SYMBOLS

Letters	Description	Unit	Subscript	Description
A	Area	m^2	**a**	Aperture
$\dot{m}$	Mass flow rate	kg/s	**r**	Receiver
$\dot{W}$	Power	W	**1**	Inlet
H	Efficiency	-	**2**	Intermediate
H	Enthalpy	kJ/kg	**3**	Exit(outlet)
P	Pressure	$kg/m\text{-}s^2$	**out**	Output
F	Actual Mass flow rate deviation to the reference Flow	-	**is**	Isentropic
Cp	Specific heat capacity	$kJ/kg\text{-}^oK$	**is,d**	Isentropic at design Condition
UA	Overall heat transfer Coefficient		**ref**	Reference
ΔP	Pressure drop	$kg/ m\text{-}s^2$	**d**	Design
K	Coefficient	-	**a,b,c**	Turbine part-load Coefficients
T	Temperature	^oC	**gen**	Generator Coefficient
N	Number of day	-	**sat**	Saturated condition
Q	Energy	KJ	**DEA**	Deaerator
P	Power	W	$\mathbf{f_w}$	Feed water
N	Number	-	**b**	Bled steam
P	Reflectivity	-	**c**	Returning condensate
H	Transmissivity	-	**cond**	Condenser
A	Absorptivity	-	**pump**	Pump
K	Incidence angle modifier	-	**ST**	Standard time
A,B,C,D	Collector heat loss Coefficients	-	**Loc**	Location
V	Velocity	m/s	**thermal**	Thermal
Ψ	Efficiency matrix	-	**Incident**	Incident
Γ	Fraction of field in track	-		
Z	Stefan-Boltzman constant		**loss,HCE**	Loss through heat collector element
L	Length	M	**aperture,ap**	Aperture
$\mathbf{P_{field}}$	Field density	-		
R	Radial distance from representative heliostat to	-		

CHAPTER 1

1. Introduction

The world's energy demand is highly depends on fossil fuels to fulfill the demand of rapid growing industries, vehicles, human requirements and commercial sectors. It is seen that coal is most wide used fossil fuel among the all fossil fuels in the world and it is estimated that the amount of coal remain in the coal reserves of all over the world is 990 billion tons ,which is equivalent to 150 years current consumption[1].Among total power consumption over the world coal alone provide 37% and By seeing the rapid power demand ,especially for developing countries it can be said that coal is going to remain main component of fuel for generating power.

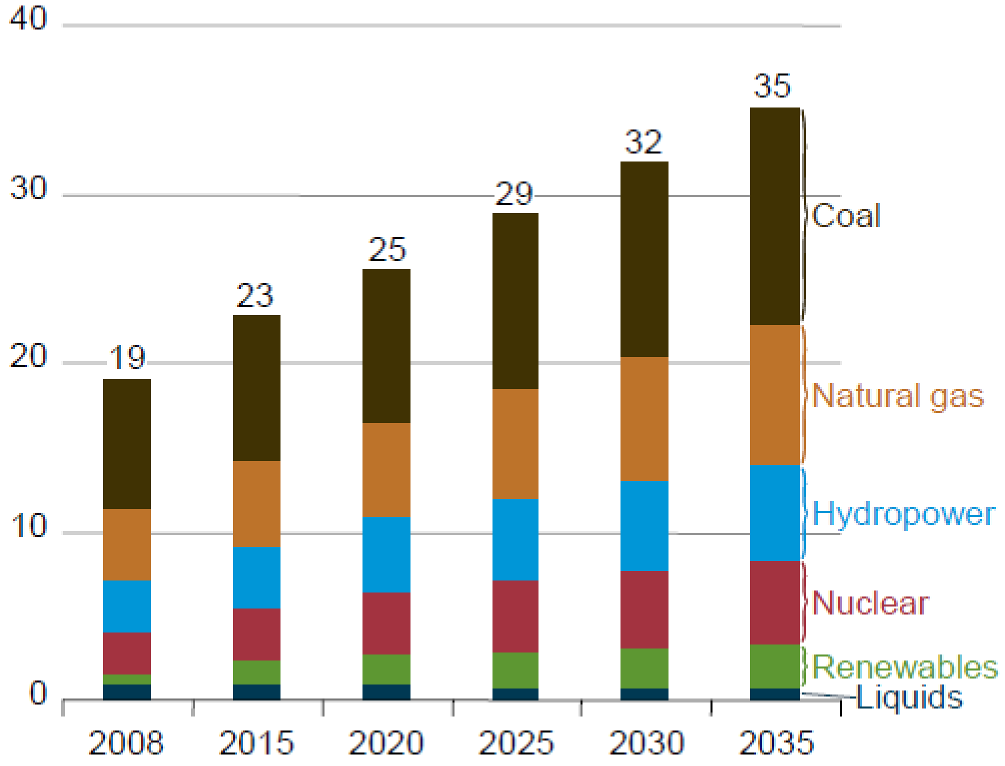


Figure 1.1: Net power generation by different type of fuel in world [2]

Although the initial set up cost of coal based thermal power plant is very high comparative to other fossil fuels, then also it very popular because coal mining technique is simple, easy and cost effective than the mining process of other fossil fuel. For this reason it is easy to run their business for the coal based companies.

Coal based thermal power plant creates some environmental as well as health problem, which are a very big matter of concern to the environmental and engineering researchers. Here some major problems are listed:

- Destruction of land, water sources and job life risk at the time of mining.

- In some of the counties, trains which are using for carrying coal are running on coal, which emits GHG.
- Fresh water is the major requirement for human life. But in thermal power plant there uses a very large amount of fresh water which hampers the ecological balance.
- Burning of coal creates more than 100 pollutants. As coal is burnt in power plant at very high temperature so it releases major amount of CO_2 which is the main GHG, this causes temperature rise of earth, burning coal in power plant also creates sulphur di oxide, which causes acid rain. At high temperature burning of coal creates nitrogen di oxide, which causes smog and asthma. Thermal power plant is the largest source of contributing soot particle in the environment, which causes the heart and lung diseases [3].
- Coal contains mercury. When it is burnt the mercury is released through stakes into air. At the time of rain it mixed with water and goes to pond, river, lakes etc, which hampers the life of water living animal (i,e-fishes), by eating them mercury goes to human body.
- Mining and burning of coal generates large amount of harmful gases.

Keeping in mind this dangerous outcomes, many research work had been done to minimize this pollution. Many power plants have been hybridizing with renewable energy sources to increases the efficiency as well as minimize the pollution effect.

Using of solar energy has been selected in many places out of all other renewable energy sources to reach goal of reducing the pollution level as well as reducing the fossil fuel consumption level in the world. Power generating by using solar energy only is very old technique among the renewable energy sources. In 1870 scientist John Ericsson run an experiment and made a 3.25 m^2 aperture of a Parabolic Trough Collector (PTC) which run a small 373 w engine [4]. Tough in 1974's world's oil crisis indicates to accelerates the uses of alternative energy resources but it was 1984 when independently solar thermal power plant came into picture in functional way in the massive desert. But the economic and thermal advantages given by independently solar thermal power plant was not fulfilling all the demands. As they can’t be operated in night, In cloudy weather they are useless and the investment cost was also very high. So the conventional coal based thermal power plants are found to be more feasible for power making companies, which hampers the proposal of large scale of implementation of solar generation in the world. Besides aiding of solar into conventional thermal power plant provided more economic and thermal advantage over convention coal fired thermal power plant. The aided solar thermal system has been first proposed by Pai in 1991 [5].

Aiding of solar energy in conventional thermal power plant increases heat capacity of the boiler. If we apply this technology we have to supply less energy by burning of coal to boiler for same power output, which means the efficiency of the plant is increased. The solar aided power generation (SPAG) is only adding of some solar heater to already existing thermal power plant and add some simple control panel. So for this we do not need any other space. As solar heaters are added in existing system so capital investment can also be minimized if we run it in power saving mode. It will also help in reducing CO_2 emissions and other harmful gases. As a result the power companies have to give less penalties.

In 1996 the world bank from global environmental facility(GEF) was impressed seeing the SAPG technology and decided to give financial support on the ISCC system in developing nations where sun remains for long time. GEF decided to give financial support where the project was less than 50 million to implement solar in existing coal based power plant. This type of projects running in Greece, India, Morocco, Egypt, Spain, Iran,Mexico. The capacity of the project was 50 to 310 MW and in all this projects parabolic trough systems are installed. In European countries European commission gives financial support in the business which are running with only solar power generation [6].

Although limited agencies are financed in integrated solar thermal plant, but the use of solar in conventional power plant plays a great role in increasing of overall efficiency of plat and controlling the pollution and fuel cost now in which part of the conventional power plant solar can be integrated and how will it operated with the existing system, is the subject of study of this paper.

CHAPTER 2

2. Literature Review

2.1 Integrated Solar Hybrid Systems

Integration of solar energy enjoys the benefits of both energy plants. The solar heat collected from collector array is directed to a heat exchanger using heat carrier fluid such as molten salt solution and in most cases, thermal oil for transporting the captured thermal energy. The overall energy production of a plant is supplemented by the solar field which can either boost the already generation capacity or reduce the fuel consumption rate. The reliability of the generation utility is usually enhanced during summer seasons in which a greater demand for energy arises out of which a significant portion can be met with energy harvested from solar generation system. The simultaneous availability of solar energy on a clear summer sky results in high solar energy to be harvested as it is consumed within the growing power demand side (due to increased ventilation and refrigeration requirement) at the same time.

The following schematic shows an integrated hybrid solar plant with exhaust gas heat recovery steam generator. In such a system, the flue gas of the topping cycle is used to generate live steam for the bottoming steam cycle. The setup incorporates a two stage power cycles with reheat. The steam cycle also integrates regenerative system whereby turbine extracts are used in preheating the feed water from the condenser. The solar heat recovery steam generator (HRSG) generates saturated and superheated steam which later will be further heated to the required live steam conditions at the turbine inlet. In this particular setting, solar steam generator is designed for reheating of the steam between high and low pressure turbines to produce more work in the cycle. [7] In common regenerative cycles the overall plant efficiency of the system is improved while compromising gross energy output from the cycle.

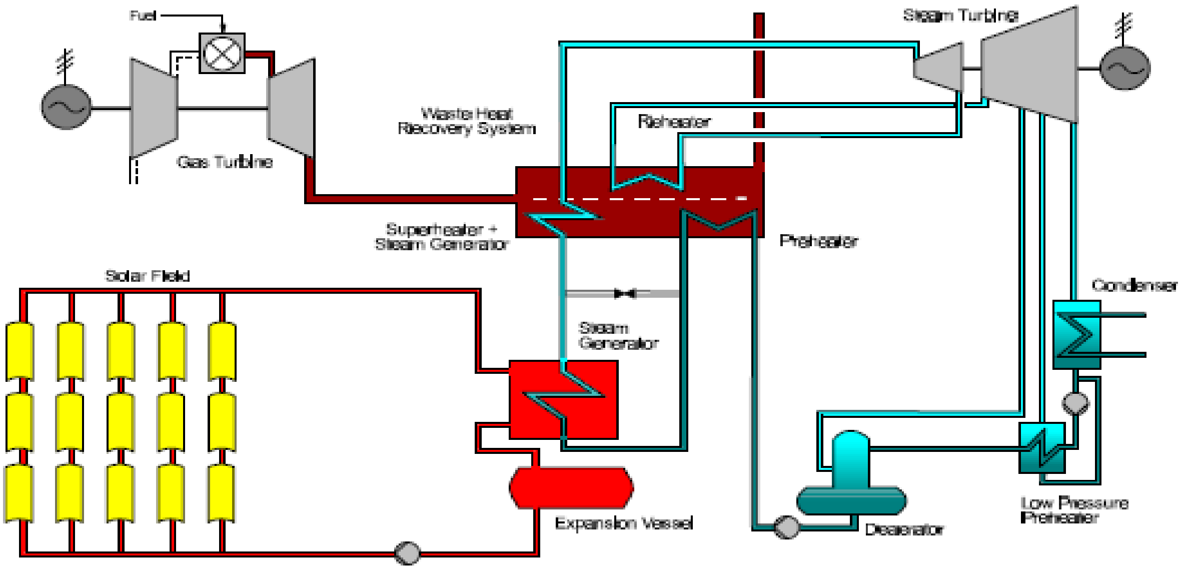


Figure 2.1 Integrated solar plant schematic diagram [8]

Since their introduction by Luz International as a means of integrating a parabolic trough solar plant with modern combined cycle power plants, Hybrid solar power plant have been known to offer distinct thermodynamic and economic advantages over standalone solar systems and fossil-fueled generating schemes. Their advantages can be summarized as:

- Small annual solar thermal contributions to an integrated plant can be converted to electric energy at a higher efficiency than a solar-only SEGS plant, and can also raise the overall thermal-to electric conversion efficiency in the Rankine cycle [8].

- Inefficiency due to daily startups and shut downs that are experienced in steam Rankine plants are avoided.

- Incremental cost in retrofitting existing plants that include building solar field, increasing steam turbine capacity to accommodate the additional solar field generated steam, and controlling equipment cost are comparatively less than the overall unit cost of SEGS plants.

- Reduced operating cost as a result of reduction of fuel consumption in case of fuel saving mode of operation.

- Secure power production independent of solar input variations.

In addition to the stated advantages, in a direct steam generation (DSG) hybrid solar arrangement whereby the feed-water of the Rankine cycle is led into the solar field and is directly heated by the incoming solar irradiation without requiring an intermediate heat exchanger and energy transporter fluid has been shown to provide noticeable financial advantages and efficiency gains due to reduction of thermal and exergy losses within the system. Even though, it is under its developmental stage, the effort to integrate DSG systems with thermal energy storage can possibly be a way forward of the future during non-solar periods of operation [9].

Despite the merits, the introduction of DSG hybrid solar system may introduce complications associated with the simultaneous coexistence of two or more phases within the transmission pipes which in turn reduces the operational life of the plant. A DSG plant also presents difficulties in controllability of operations that makes the control electronics even more expensive and complicated. Currently, using solar towers with central steam generation DSG is becoming the standard process.

As noted earlier, there are two modes of operations through which specific power objective or emission requirement criteria can be achieved. The power boosting mode, as noted by Hu et al. [10] uses the surplus steam in the different pressure steam turbines to generate additional power while maintaining the same level of emissions per unit volume of fuel consumption. The power boosting mode consumes the same fuel mass for higher electrical output. On the contrary, the fuel saving mode, by the virtue of supplementing the heat source, reduces the fuel consumption as well as the in gas emissions associated with fuel usage.

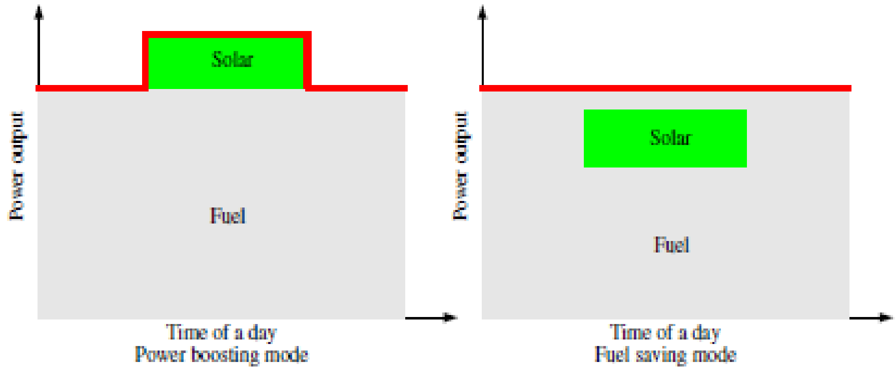


Figure 2.2 Two typical operation modes of solar aided power generation scheme [10]

It has been suggested that the utilization of solar thermal heat in HSPP systems should be done at the highest level of operating pressure and temperature for the saturated steam. But due to the particular solar field of choice (PTC, LF, CRS, and DE), field variables, and geometric field dimensions, the pressure drop within a given system may limit the working pressure of the system. Hosseini et al. assessed the technical potentials of a parabolic trough operated ISCCS plant and found the pressure drop as a controlling variable that limited the separate solar field capacity to 100 MW at the given location ([11]; [12];[13];[14];[15]). The state-of-art maximum attainable working pressure for DSG based technology is around 160 bar

reported by Ivanpah Solar, a central receiver tower plant having maximum capacity of 392MW plant in California, with 35 bar/480 ^{o}C reheat steam cycle. One of the earlier SEGS of Luz international produced steam pressure 35.3 bar for conventional Rankine steam cycle. Comparing to claims by Montes et al., the operating pressures of the present cycle has grown three folds. This is attributed to the usage of non-degrading water/steam heat carrier apart from synthetic thermal oil that loses its key physical properties at elevated temperatures and pressures.

The particular location in thermal system where the solar steam generator can be integrated is very well related to the temperature and pressure of the steam output from the solar collector assembly (SCA). This is in turn related to the particular method of solar steam generation scheme used.

On the integration of solar plant into conventional combined cycle power plant, the solar steam generated from the solar field can be integrated into the Rankine steam cycle at low and high pressures. The difference in the integration varies according to the quality of steam at the input to the steam cycle. In case of provision of the steam at high pressures, steam is supplied at saturated conditions whereby the rest of superheating and reheating takes place in the HRSG in the ISCC plant or by other supplementary fossil fuel in case of a fossil fuel power plant. On the contrary, low pressure superheated steam generated from solar field is led directly to the lower pressure turbine without requiring further heating in lower pressure conditions.

The more efficient way to introduce solar power in a Combined Cycle plant is to preheat the combustion air in the gas turbine, a technology in development. [16]

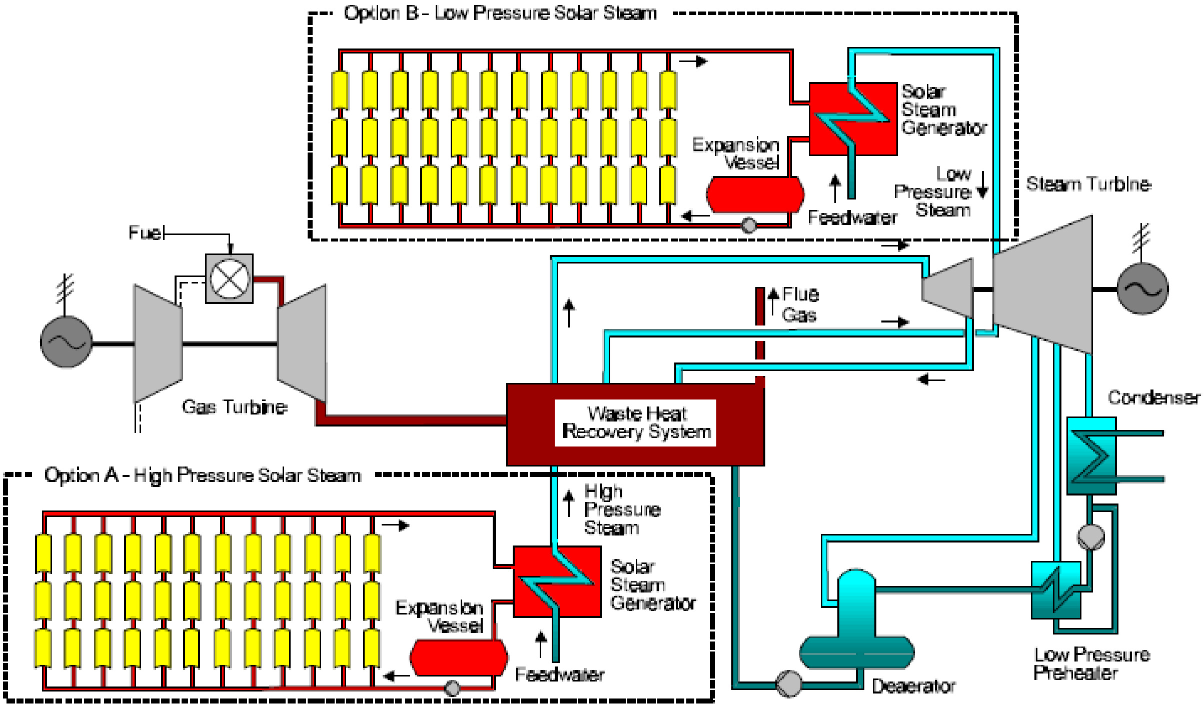


Figure 2.3 Integration of solar steam at different pressures in ISCCS [17]

While thermodynamics commends the efficient use of the working fluid should be at the highest exergetic value, the possibility of low temperature steam into the power plant is also a reality. Demonstration by Hu et al. [10] on base-load coal fired power plant that uses medium or low temperature solar heat source ranging from 100^{o}C to 260^{o}C, DSG system has been proven to possess solar-to-electric efficiency of 36.58 %. The efficiency of integrated heat resource under 100^{o}C is more than 10 % which is much more preponderant than other solar thermal energy power generation types at the same temperature. [17; 18] It is important to mention here that by integrating low or medium temperature heat source, this particular SAPG is not directly producing steam for the steam turbine, and instead, solar thermal energy is used to replace the bled-off steam to pre-heat the feed water at different positions of the circuit. In this way, the bled-off steam is directly led through the turbine to produce additional power. The added benefit from this configuration is the ability to efficiently utilize the solar heat to achieve a multi-level and multi-point solar integration into conventional power plants. [10] As opposed this method, low pressure generated steam from the solar field in figure 1.4 directly participates in generating more power by leading the steam through the lower pressure turbine.

The solar fraction that defines the portion of solar energy contributed towards the net electrical power output of HSPP is limited. This is due to the additional cost incurred in accommodating the additional solar capacity may not be feasible due to the fact that the specific cost of steam turbine size is larger than its technical limit. Bruce et al. [8] after conducting a study on a General Electric Frame 7(FA) gas turbine with a three pressure heat recovery steam generator combined cycle plant integrated with parabolic solar collector system concluded that solar contributions up to 12 percent offer economic advantages over solar only SEGS plants. Among the GEF funded projects, the Egyptian and the Moroccan hybrid solar ISCCS projects have annual solar share close to 4% which is practical indication of how much solar energy can be adapted. Similarly, Solar shares of nearly 10% have been reported to have been achieved on those hybrid solar plants operating in Mojave desert, California.

In the previously mentioned study by Bruce et al , annual solar contributions in the range of 1 to 2% could render highest solar thermal-to-electric efficiency of about 40- 42%. Further augmentation could lead to a decline in solar thermal-to-electric efficiencies by about 5 -10 %. This could be explained from the part load operation of the plant during non-solar operating hours within which the steam turbine efficiency declines by noticeable amount.

At small solar capacities, the HSPP possess high solar efficiencies while the combined solar-fossil conversion efficiency increases till it reaches optimum operating point. Increasing solar share beyond this point, both efficiencies suffer due to efficiency penalties in the part load operation during non-solar operating hours. It is important to note here that optimum operating points for both the Rankine steam cycle run from fossil fuel source and the solar field are not reached at the same time. Higher solar share entails higher capacity steam turbine which is sometimes twice in size compared to fossil only fueled plant. During the non-peak non-solar hours the plant's overall performance steadily deteriorates due to having less live steam pressures. As a result, yearly part load efficiency penalties could reduce the HSPP performance by about 10- 15%. [8].

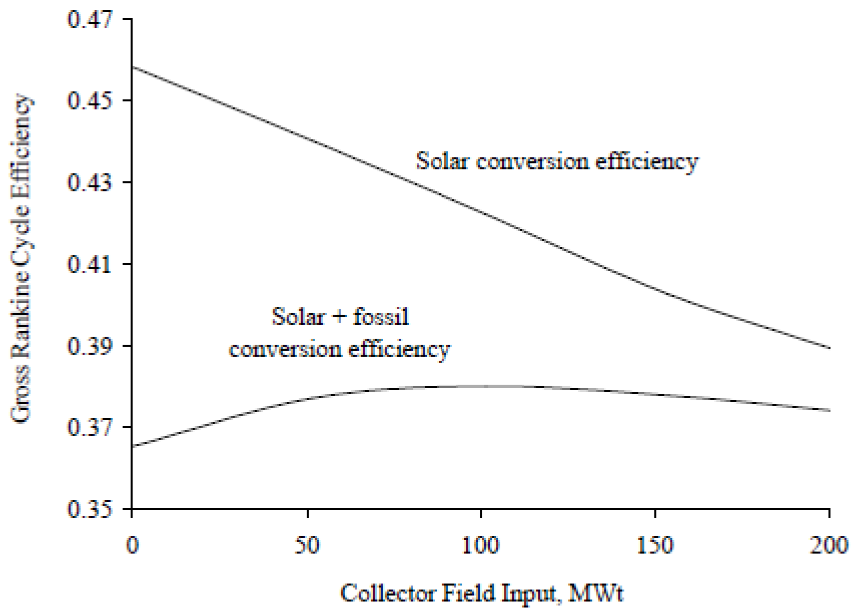
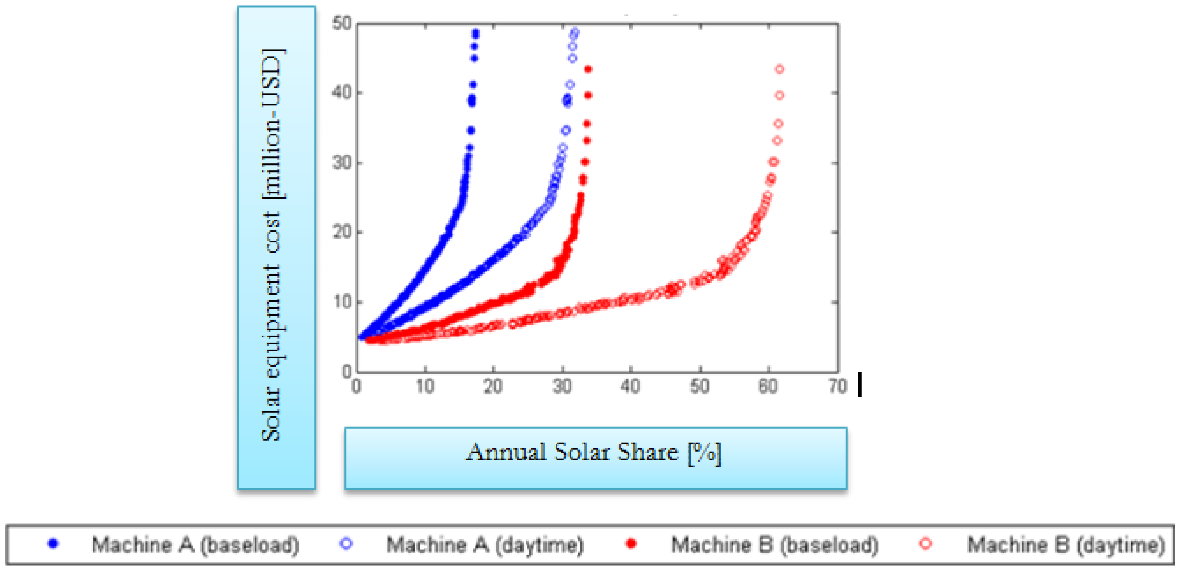


Figure 2.4 Rankine cycle conversion efficiencies for HSPP [8]

Most of the HSPP studies carried out by different authors consider Gas Turbine exhaust recovery systems in which solar energy is used to supplement the heat supply to the lower cycle([8];[19];[20];[21];[9];[11];[12];[22]). Of these systems, the majority of solar energy harvesting systems is mostly based on parabolic trough collector technology due to its well matured technical and technological advantages over other solar steam generation means. Thermo-economical assessment on central receiver system (CRS) integrated with hybrid solar gas turbine power plant (HSGTPP) was performed by James Spelling et al. [19] using a multi-objective evolutionary algorithms between low temperature and high efficiency gas turbines. The result illustrates the unsuitability of very high temperature heat sources in attaining larger solar shares of the cycle. With deployment of low temperature less efficient correctly adapted firing temperatures, even though having constant levelized energy cost over wider range of solar shares, the CO_2 emission penalties associated could very well drive the operating costs to a higher level.



Machine A – High temperature high efficiency gas turbine

Machine B – Low temperature low efficiency gas turbine

Figure 2.5 Annual Solar Share versus Solar equipment investment cost adopted to year 2005 [19]

Similar studies on retrofitting new and existing Coal-fired power plants with solar power have been investigated ([23];[10];[13];[7]) E. M. de Sousa [7], after undertaking steady-state component by component thermal modeling of 500 MW linear Fresnel (LF) modules with coal-fired hybrid plant, concluded that the solar power boosting mode to be the more rentable option compared to fuel saving mode. Based on his computation at current fuel price rates, the fuel saving mode, if it has to offset the economic benefits from solar augmenting mode, there needs to be 55 $ /ton CO_2 feed-in tariff in emission penalty imposed by energy sector regulatory entity.

Oil fired hybrid solar systems were proposed by Dimitry Popov [24] through the replacement of low and high pressure feed water heaters at different load conditions. Alvaro Lentz and Rafael Almanza developed a solar aided geothermal power plant in which the solar energy is used to generate additional steam to the power cycle [10].

Many of the practical experiences related with solar thermal assisted plants are related to gas based generation schemes. Solar hybridization with CSP become the first of its kind in 2009 when Aora solar , Israeli based solar company, built 30m central tower housing a hybrid micro turbine operating with fossil fuels(diesel, biogas, and natural gas) in Kibbutz Samar, Israel . Recently, there has been a growing interest in implementing mainly coal-based solar systems for the purpose of meeting utility emission goals. A practical demonstration plant, Cameo, by Xcel Energy with a combined capacity of 48 MW became operational in 2010 near Grand Junction, Colorado. This facility uses a 4 MW, eight row parallel set of SCA parabolic trough field designed by Abengoa Solar that covers about 6.4 acres of land area to preheat the boiler feed-water before the boiler. The solar energy is used to achieve lower NOx emission targets and at the same time the plant is able to attain savings in coal consumption up to 900 tons per year. Xcel anticipates that the project will increase the power plant's efficiency by up to 5% and will reduce carbon dioxide emissions by 2,000 tons per year. [25]

Figure 2.6 Solar-coal hybrid combined-cycle Cameo plant[25]

Although the Australian government had already introduced a solar boosting project which amounts 0.15% in solar fraction in Liddel power station by 2008, it was not until recently that the government took an ambitious 750 MW coal plant integrated with 44 MW of linear Fresnel solar system which is going to be constructed by AREVA solar to be commissioned in 2013. The project which is under construction in Queensland is said to be the biggest of all solar hybridized coal-based power stations in the world and is also estimated to avoid 35,000 ton of greenhouse gases per year when being operational.

M K Pal, H. Chandra and A Arora[26] had done Thermodynamic Analysis of Existing Coal Fired Conventional Steam Power Plant Using Cycle Tempo on Proceedings of BITCON-2015 Innovations For National Development National Conference on : Innovations In Mechanical Engineering For Sustainable Development.

M. K. Gupta, S. C. Kaushik [28] had done Exergy Analysis And Investigation For Various Feed Water Heaters of Direct Steam Generation Solar-Thermal Power Plant in which they showed that maximum exergy loss occurs in condenser of the plant and minimum exergy loss occurs at feed water heater, so they suggested to introduced more(at least three) feed water heater in the plant.

Narendra Sing, S. C. Kaushik, R.D. Misra[29] had done Exergetic Analysis of a Solar Thermal Power System in which they had done second law analysis of solar aided thermal power plant and they explain the deviation between the actual efficiency and ideal efficiency of a solar thermal power system.

2.2 Objectives of the Study

The main target of this thesis is to investigate the technical and economic feasibility of integrating a parabolic based solar steam generation scheme into a typical coal-fired power plant. The solar share to be achieved that meets economic advantages over SEGS and conventional fossil fuel fired plant will be subject of study. The study strives to reach the following specific objectives with regard to hybrid solar power plants:

- Based on location of case study plant and its system thermodynamic conditions, introduce a solar power generation scheme into a coal-fired power plant.
- By making use of commercial software, analyze the performance of the plant operation with and without solar integration.
- Select decision variables for economic and operational comparison between integrated and fossil only modes.
- Estimate the marketability of the chosen operation mode.

CHAPTER 3

3. Case Study of solar Aided Coal Fired Thermal Power Plant

3.1 Plant Description

Majority of today's thermal power plant is dependent on fossil fuels to fulfill the heavy demand of electricity. Most of this plant uses coal as fuel in the steam generator to produce steam, which are based on rankine cycle. There is a global proposal that when any innovation will done on the present power plants running on fossil fuel utilization, there must be used alternative fuel resources based on economic basis. It was a very good step towards the increase the use of renewable energy sources to fulfill the high energy demands. It is a very intelligent decision taken to integrate renewable energy resources with already existing thermal power plant to minimize the drastic economic uncertainties.

Solar is a good alternative fuel for power generation and it will be very attractive option to integrate with existing coal power plant to achieve this goal. The research is going on to integrating coal and fossil fuel with solar since 20th century and for this result knowledge of integrating power generation growing on increasing.

In order to enhancing the knowledge of pre-existing coal based thermal power plant, a case study of Bakreshwar thermal power plant is taken for knowing the points where the solar power can be implemented.

Bakreshwar thermal power plant is located in West Bengal. It is under the West Bengal Power Development Corporation Limited. Its installed capacity is 1050 MW and there is a proposal of installed another 500 MW unit.

Table 3.1 Bakreshwar thermal power plant site specific geographical and climatic data[30]

Quantity	**Unit**	**Value**
Site Name		Bakreshwar thermal power plant
Latitude	Degree	23º50´N
Longitude	Degree	87°27´E
Installed capacity	MW	1050

The plant is expanded on 4.98 km^2 and there is enough remaining space to installed solar plants. To execute this plan best suited collector is parabolic trough collector. This power plant capacity is 1050 MW and it comes under subcritical thermal power plant. It is one of the highest capacity units in the country. It has 5 separate units and each unit has design capacity 210 MW.

Some information regarding ratings and manufacturer specifications of BkTPS is listed in table3.2

Table 3.2 Plant component specifications for BkTPS[30]

General	
Designed Plant Capacity	1050 MW
Average Out put	980 MW
Rated efficiency of designed turbine output	34.9%
Average availability	90%
Average annually energy production	6 GWh
Working hours per day	24 hrs

Turbine	
Manufacturer	Dongfong electrical limited
Type	Multi cylinder with Impulse reaction

Generator output	440 KVA
Speed	3000 rpm
Design work rate of turbine	223.1 MW
Generator efficiency	98%
HPT isentropic efficiency	90%
IPT isentropic efficiency	85%
LPT isentropic efficiency	80%

Boiler	
Manufacturer	Dongfong electrical limited
Type	Corner fired water tube boiler
Number	5
Mass flow rate of steam	165 kg/s

Generator	
Manufacturer	Shanghai electrical limited
Design capacity	445 KVA
Efficiency	98%

The Bakreshwar Thermal Power Plant consumes sub-bituminous type of coal. Sometime blending between bituminous and lignite type of coal has been done. Information regarding consumption of coal, energy supplied, fuel content of the plant is given by the given table below.

Table 3.3 Fuel content and consumption of a coal burnt in Bakreshwar Thermal Power Plant [30]

Fuel	
Calorific Value	4000 KJ/Kg
Ash content	40%
Sulphur content	0.6%
Coal consumption rate at full load	97.2 Kg/s
Total annual consumption by all units	$6*10^6$MT

The steam set up of Bakreshwar Thermal Power Plant has one HPT, one IPT, one LPT. The evaporator section consists of three super heater called primary super heater, pendent super heater, final super heater, a re-heater, one economizer, boiler drum. Air is first drawn from surrounding with the help of FD fan and preheating in air pre heater (APH) by the exhaust flue gas coming out of the economizer. This air is called secondary air which enters into the combustion chamber. There was another fan called PA fan, which also draws atmospheric air and send it to APH for pre-heating. This hot air is used to blow the pulverized coal from the bowl mill to furnace. The coal is burnt inside the furnace at 900ºc temperature. The air fuel ratio in the furnace was 16. The combustion takes place in the boiler evaporator section where water walls are situated and generates a mixture of saturated steam and water at 160 bar pressure from the riser tubes, then it goes to boiler drum, where the saturated steam and saturated water is separated.

The saturated steam goes to HPT at 540ºc through the super heaters and the saturated steam is send to bottom ring header (BRH) through down comer (D/C). Again from BRH saturated water goes to water wall. In another side the steam coming out from HPT at 40 bar enters into re-heater to regain its thermal energy which it loose in HPT. After coming out the steam from re-heater at again achieves 540ºc. As reheating is constant pressure process so the pressure of coming out of re-heater is 40 bar. A part of HPT extraction is send to high pressure heater for regeneration to increase the plant efficiency. The steam coming out of re-heater is send to IPT then it directly sends to LPT. The extractions are also taken from the IPT and LPT to medium pressure heater and low pressure heater for regeneration purpose and increasing efficiency.

Apart of extraction steam is also used for sealing the turbine. As steam pressure inside the turbine is very high than the atmospheric pressure so there is chance of leaking of steam which decreases the plant efficiency. So sealing is needed.

From the LPT stem and water mixture goes to condenser where steam is condensed with the help of cooling water coming from cooling tower. Condenser is a shell and tube type heat exchanger. Within the condenser below the atmospheric pressure (-ve pressure) is maintained. From the condenser through the condensate extraction pump condensate goes to low pressure heater where it is heated up by the extracted steam from LPT. Then the water goes to deaerator, which is a open type heater. With the help of heated steam the dissolved air in the working fluid is removed in deaerator. Then the water went BFP. Deaerator is situated on 16m height from the ground level to provide NPSH to the BFP. BFP raises the pressure of the working fluid up to the maximum pressure which is the operating pressure of boiler. Then water goes to the high pressure heater where it heats by the steam extracted from HPT. Then water goes to the economizer where it takes heat from the out coming flue gas. Then saturated water and steam mixture goes to the boiler drum. There also separation of saturated water and steam occurs and same cycle repeats as stated above.

The schematic diagram of major component of Bakreshwar Thermal Power Plant is given in figure 3.1.

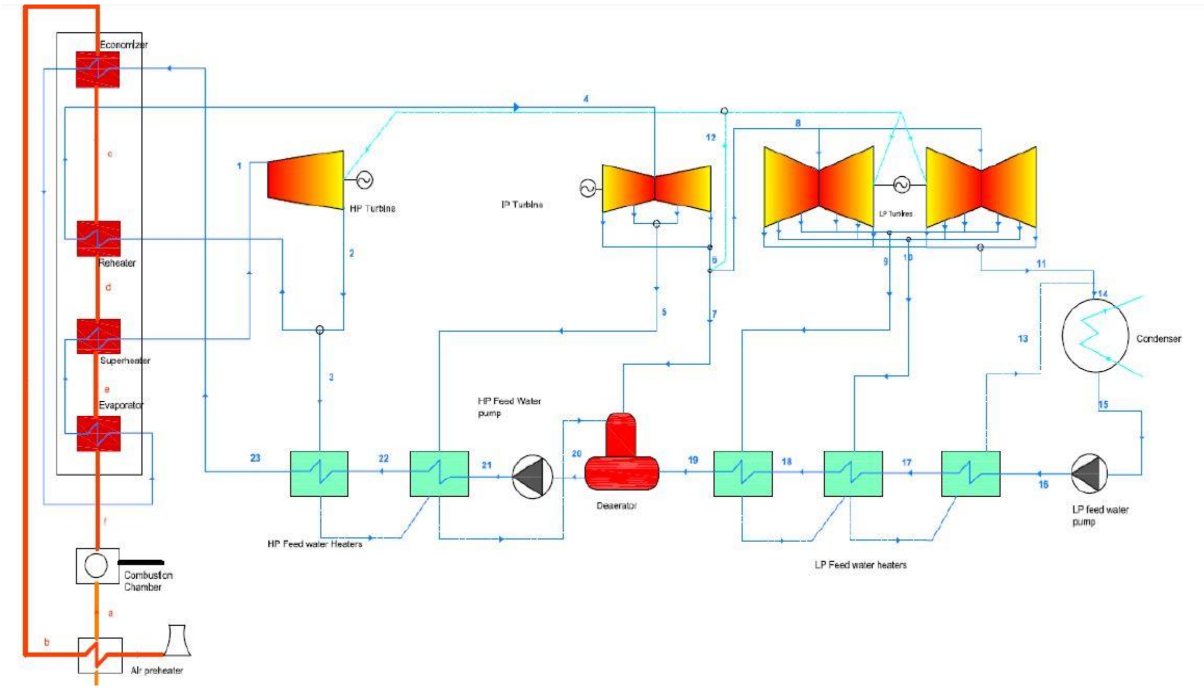


Figure 3.1 Bakreshwar Thermal Power Plant power cycle schematic[30]

3.2 Power Cycle Components

3.2.1 Turbine

Turbine produces work in successive stages. So it is work producing rotating device. The high pressure turbine accepts superheated steam from final super-heater. The mass flow rate of steam enters to the HPT is $\dot{m}_1$. The $\dot{m}_3$ amount steam is extracted from turbine stage which is used to heat the feed water in the open and close heater to increase the cycle efficiency.

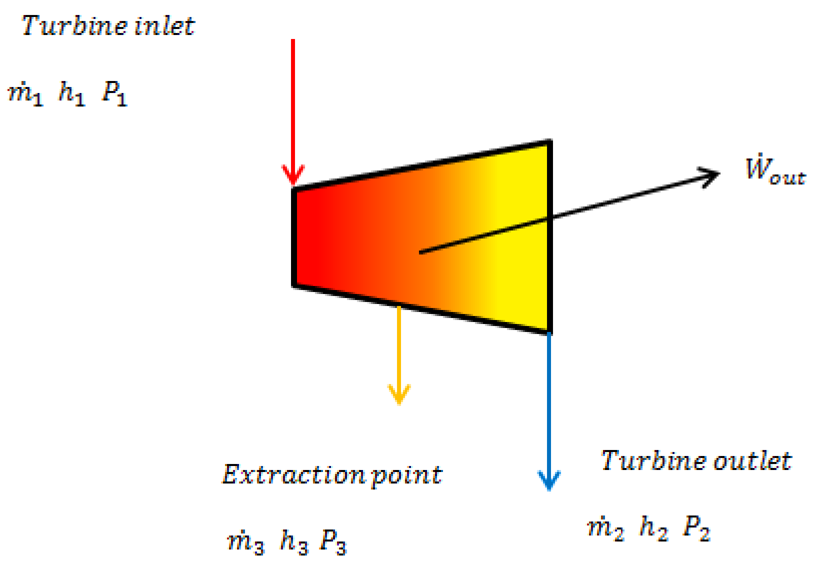


Figure 3.2 Model of turbine showing mass & energy flows

By applying the SFEE for open system for the turbine to calculate work from the turbine is

$$\dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3 = \dot{W}_{out} \quad (1)$$

Isentropic efficiency of turbine is the ratio of actual enthalpy drop to the isentropic enthalpy drop.

$$\eta = \frac{h_1 - h_2}{h_1 - h_{2s}} \quad (2)$$

The stage of each turbine is constructed with respect to its nominal isentropic efficiency which is also derived by its part load performance. The variation of three stage isentropic efficiency can be approximates by a three degree polynomial relation which relates the ratio of actual mass flow rate to the design mass flow rate on which the isentropic efficiency is measured.

3.2.2 Generator

Generator is a electrical device in which mechanical energy is converted to electrical energy. Here also the amount of energy supplied by the turbine will not fully converted into energy. Frictional loss, wind age loss occurs in generator. In generator electricity is generated by the faraday's law of electromagnetic induction which states that 'Whenever a device is rotates in a magnetic field then an electricity is generated in that device'. Here generator shaft is coupled with turbine shaft which is a rotating part and magnetic field is given in rotor part by

DC source. The stator is the static part here which cuts the magnetic field and generates power.

$$\eta_{gen}=0.908+0.258\left(\frac{\dot{w}_{net}}{\dot{w}_d}\right) - 0.3\left(\frac{\dot{w}_{net}}{\dot{w}_d}\right)^2 + 0.12\left(\frac{\dot{w}_{net}}{\dot{w}_d}\right)^3 \quad (3)$$

From this relation generator efficiency can be derived. The graph keeping generator efficiency in y-axis and the ratio of net work out put to the design turbine work is given below.

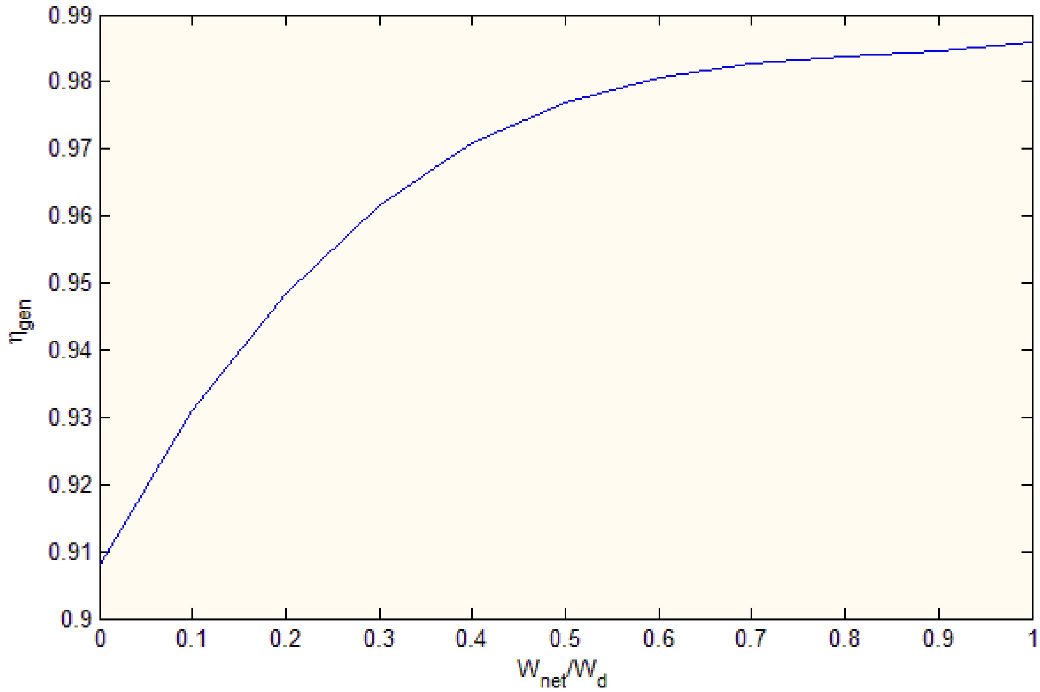


Figure 3.3 Part-load operation of the turbine generator's efficiency with respect to work ratio

Generator efficiency is given by the power coming out of generator to the given energy to the generator by turbine. It should make sure that turbine output should not fall below 35% of rated capacity for the safety purpose of its components. From this concept it can be derived that the generator efficiency can be varies in between 96% to 98.15%.

3.2.3 Boiler

Boiler is a constant pressure vessel. It is four wall room like structure, where water tubes remains. In BkTPS boiler is corner-fired type. Coal burners are situated in six elevations through pulverized coal is injected. There is two header called bottom ring header and top ring header. From boiler drum saturated steam comes to the bottom ring header, from there it goes to the water wall where water takes heat from furnace, generates a mixture of steam and water. From the top riser this mixture is goes to the boiler drum, cyclone separator remains there which separates the steam from saturated water. Then saturated water comes through the down comer to the bottom ring header and the same cycle is repeated. The saturated steam is send to the super heater.

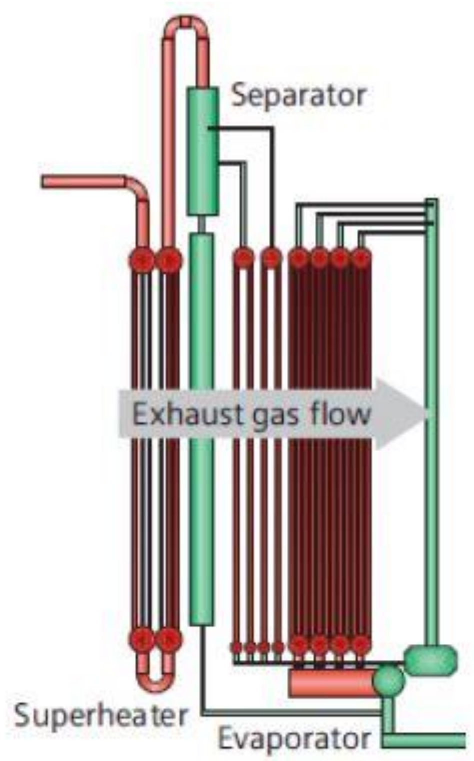


Figure 3.4 A water tube boiler (source SIEMENS)

Boiler is heat exchanger. At the time of design the evaporator section the capacitance of heat transfer wall is taken as negligible. The effectiveness of the heat exchanger is calculated by NTU method.

$\eta_{ev} = 1 - e^{-NTU} = 1 - e^{-(\frac{UA}{C\min})}$ (4)

where $C_{min} = C_{p,gas}.\dot{m}_{gas}$

The amount of heat transferred from the hot flue gas to the water wall can be calculated from energy balance from hot stream

$\dot{Q}_{ev} = \eta_{ev}.\ C_{p,gas}.\dot{m}_{gas}(T_{g,2} - T_3)$ (5)

By the boiler inlet temperature and the maximum temperature difference in the evaporator , the outlet temperature of the hot flue gas is calculated.

$T_{g,3} = T_{g,2} - \eta_{ev}(T_{g,2} - T_3)$ (6)

3.2.4 Superheater and Economizer

After coming out of the boiler drum saturated steam goes to primary super heater, where it takes heat from the coming out flue gas. Then the steam goes to the pendent super heater. Here steam takes heat from the flue gas mainly in radiation heat transfer mode. Then it goes to final super heater. In between attemperator was there to control the inlet temperature of steam to the HPT.

Economizer is last heat-exchanger consists in the feed water circuit. After high pressure heater feed water goes to economizer where it takes heat from flue gas and make the feed water saturated and send it to the boiler drum.

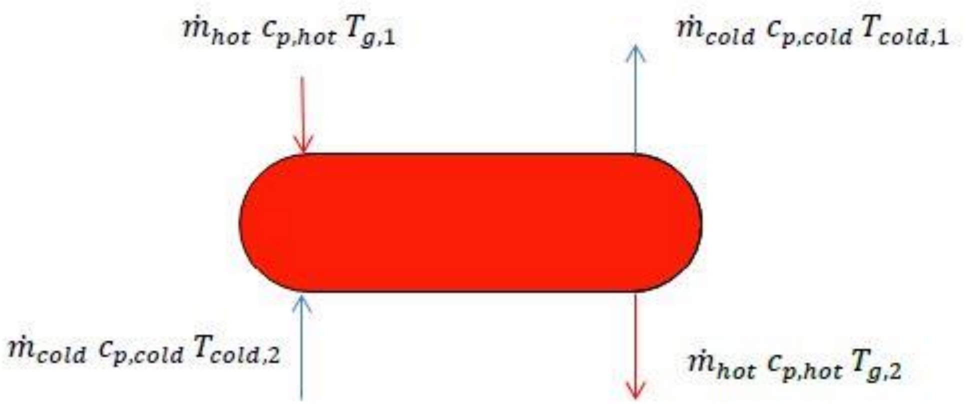


Figure 3.5 schematic figure of economizer

The effectiveness of economizer or super heater is also measured by NTU method and it can be given by

$$\eta_t = \frac{1-e^{-\frac{UA_i}{C_{min}}(1-\frac{C_{min}}{C_{max}})}}{1-\left(\frac{C_{min}}{C_{max}}\right).e^{-\frac{UA}{C_{min}}(1-\frac{C_{min}}{C_{max}})}} \quad (7)$$

Where $UA_t = UA_d(\frac{\dot{m}_{cold}}{\dot{m}_{cold,d}})^{UA_{exp,i}}$ i =Superheater/economiser

$UA_{exp,i}$ = 0.8, According to patnode [31]

The amount of transferred from the hot flue gas is can be calculated by

$$\dot{Q}_t = \eta_t.C_{min}(T_{g,1} - T_2) \quad (8)$$

The exit temperature of the steam and hot can be evaluated by

$$T_1 = T_2 + \eta_{sh}.(\frac{c_{min}}{c_{cold}})(T_{g,1} - T_2) \quad (9)$$

$$T_{g,2} = T_2 + \eta_{sh}.(\frac{C_{min}}{C_{hot}})(T_{g,1} - T_2) \quad (10)$$

Where $T_{g,1}$ and $T_{g,2}$ is the inlet and outlet temperature of the hot gas in the economizer/superheater respectively.

T_1 and T_2 is the inlet and outlet temperature working fluid in the superheater and economizer.

3.2.5 Deaerator (Open Feed Water Heater)

Deaerator is a open type feed water heater. It is in the height 0f 16m from the ground level to provide the net positive suction head (NPSH). It is installed after the LP heaters. Here direct missing of steam and feed water occurs. With the help of hot steam extracted from HPT the dissolved gases in feed water is removed.

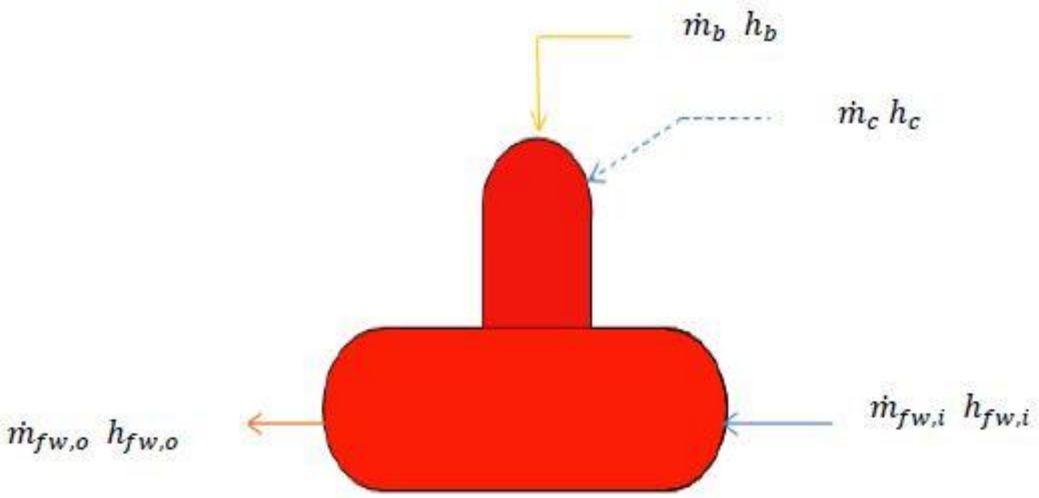


Figure 3.6 Schematic of Open feed water heater (Deaerator)

Deaerator is basically a mixing chamber. The feed water flow rate and its specific enthalpy is derived by the mass and energy flow balance over deaerator.

$$\dot{m}_{fw,o} = \dot{m}_b + \dot{m}_{fw,I} \quad (11)$$

$$h_{fw,o} = \frac{\dot{m}_b h_b + \dot{m}_{fw,i} . h_{fw,i}}{\dot{m}_{fw,o}} \quad (12)$$

where $\dot{m}_{fw,i}$ and $\dot{m}_{fw,o}$ represents the mass flow rate of inlet and outlet feed water and $\dot{m}_b$ mass of extracted steam used for regeneration.

$h_{fw,i}$ and $h_{fw.o}$ represents the specific enthalpy of inlet and outlet feed water heater and h_b is enthalpy carried by extracted steam.

3.2.6 Closed Feed Water Heater

Feed water heater is basically used for regeneration of power plants to increase efficiency. There is two types of feed water heater is used in power plant one is low pressure heater and another is high pressure heater. In LP heater the feed water is heated up by taking the heat from extracted steam of LPT where HP heater is heated up by taking the heat from extracted steam of HPT. These both heater are close type heater means the steam and water is not mixed yet.

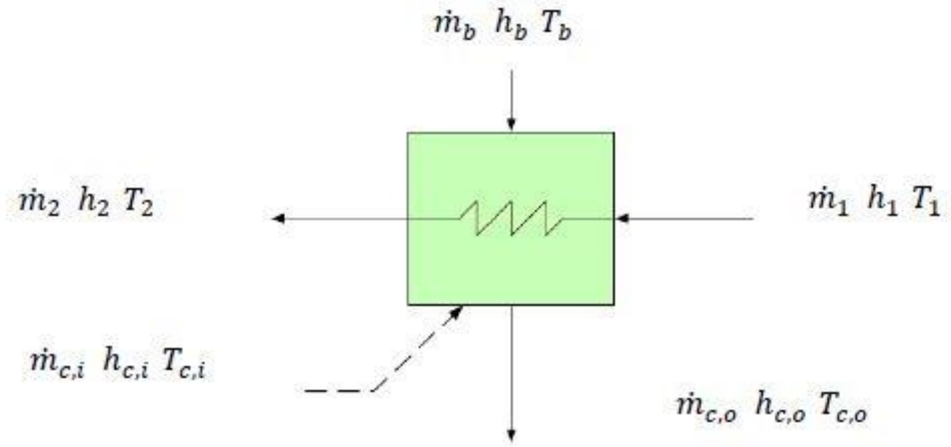


Figure 3.7 Schematic of close feed water heater

The following two equations are established the mass and energy balance of closed feed water heater

$$\dot{m}_b+\dot{m}_{c,i} = \dot{m}_{c,o} \tag{13}$$

$$\dot{m}_{c,o}\left(\frac{\dot{m}_b h_b+\dot{m}_{c,i} h_{c,i}}{\dot{m}_{c,o}} - h_{c,o}\right)=\dot{m}_1(h_2-h_1) \tag{14}$$

3.2.7 Condenser

In condenser there uses highest amount of water for the cooling purpose and the energy losses occurs in condenser is also the highest among all components of the power plant. The main function of condenser is to convert the steam coming from low pressure turbine to saturated liquid so that the water can be reused in power cycle. Condenser rejects only the latent heat of condensation. Beside of this condenser decelerate the speed of the working fluid so that it is going to be easy to handle the liquid for low pressure heaters.

The heat rejected from the condenser is derived by the enthalpy drop occurs between the incoming steam and outgoing water.

$$\dot{Q}_{cond} = \dot{m}_s h_s + \dot{m}_c h_c - (\dot{m}_s + \dot{m}_c) h_{sat} \tag{15}$$

Here it is assumed that the steam entering to the condenser is saturated and the water coming out of condenser is also saturated. So by seeing the amount of temperature rise cooling water the required mass flow rate of cooling water can be calculated.

$$\dot{m}_w = \frac{\dot{Q}_{cond}}{C_{pw}.\Delta T_w} = \frac{\dot{Q}_{cond}}{C_{pw}(T_{w,2} - T_{w,1})} \tag{16}$$

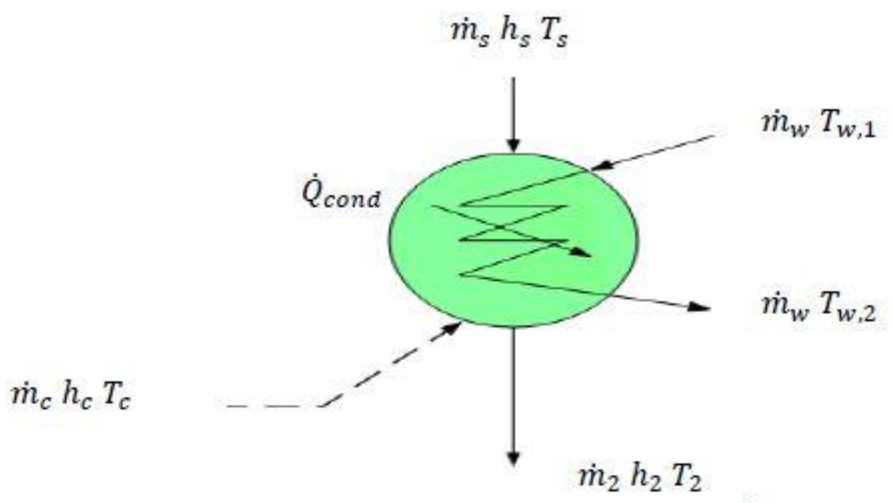


Figure 3.8 Schematic of condenser

The temperature of condenser at which the condensation occurs is determined by the exit pressure of low pressure turbine.

$$P_{cond} = P@T_{sat} = T_{cond} \tag{17}$$

It is recommended that the temperature difference between the temp of cooling water coming of condenser $T_{w,2}$ and the temperature at which condensation occurs T_{cond} is 5-10ºC.

3.2.8 Feed Water Pump

Feed water pump is a work consuming device. It rises the pressure of saturated water to the operating boiler pressure. Before calculating the net work out put or plant work the power consumed by the pump deducted from the total power output of all the turbine. On the basis of this work, plant efficiency is calculated. Between the inlet and outlet of the pump there would not be significant change of state of feed water. So the power given in the pump can be obtained from following equation

$$\dot{W}_{pump} = \frac{\dot{m}_{fw}(p_2 - p_1)}{\eta_{pump}} \quad (18)$$

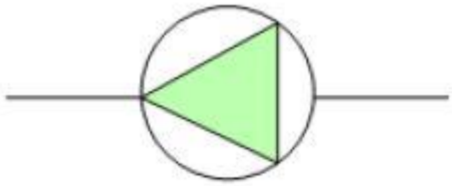


Figure 3.9 Schematic of feed water pump

3.3 Investigating Possibility for Introducing Solar Energy to Coal Plant

There are several places for presenting solar power to increase the energy production of coal-based thermal cycles. All accessible options involve the placement of exact technology based on its variety of thermal applicability, monetary feasibility and cycle presentation at small and fractional load situations. With this respect, five dissimilar options will be inspected under the two main CSP technologies, PTC and CRS. Four of these selections study the making power of solar steam at large solar cycle exit temperatures. As noted in earlier debates, the overview of high temperature hotness has been established to deliver high exergetic efficiencies. In applied demo tests did by the NREL, the increasing damage in energy from the HCE at advanced working temperatures can be balance by the higher cycle exergetic efficiency increase for the power cycle.

3.3.1 The model of the solar aided regenerative Rankine system

Generally the temperatures of the heat transporter achieved by solar collectors are little. The flat-plate collector can send warm fluid at up to 110ºC. The emptied-tube collectors can send fluid (liquid or vapour) at high temperature with comparatively large efficiency. The developments in the action of choosy faces can allow high temperatures (350ºC) to be attained in emptied tubular collectors with a reasonable concentration ratio ([1],[2]). At these temperature stages, it is hard to use the heat to make power economically.

In the orthodox regenerative Rankine cycle, the feedwater is at a small temperature variety and is heated by the steam taken out from the various stages of the expansion process in the turbine. Since in the feed heaters the temperature variations of feedwater and the taken out steam do not match well, there is a huge loss of exergy. The loss of exergy in the heaters can be minimized by growing the number of extraction phases (and feedwater heaters), but it can never be removed ; beside, it is not real to have a huge no. of extraction stages. This is the place where solar energy can fit in. The idea here is to substitute the extraction steam with solar heat to heat the feedwater in some stage(s) in the regenerative Rankine cycle where the temperature profiles of the solar liquid and the feedwater can be match each other and to use the protected steam to produce work.

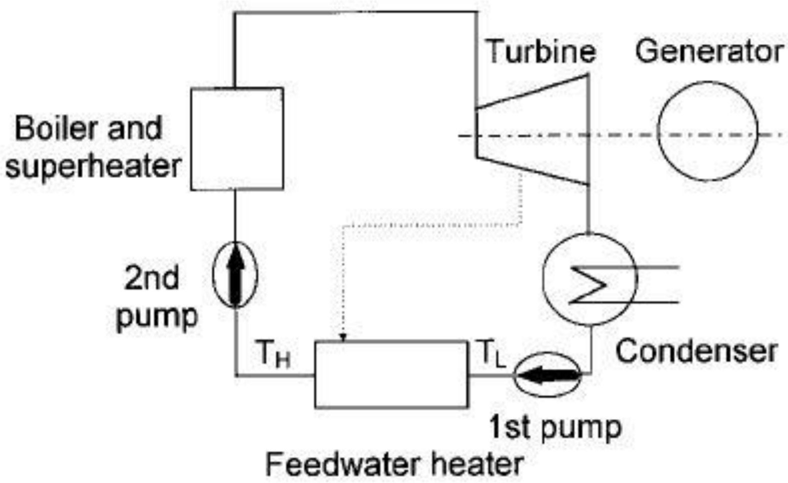


Figure 3.10 Single-stage regenerative Rankine cycle with open feedwater heater.

To demonstrate the benefits of this indication, first, let us inspect a single-stage regenerative Rankine cycle with an open feedwater heater (Fig. 1). In energy scheme studies, not only the number, but also the class of energy should be evaluated. The excellence of an energy stream depends on the work obtainable from the stream. The maximum work $w_{max,}$ the heat q at the temperature T_H can produce, or the accessibility (exergy) of the heat (here given by ``ex") is

$$e_x = w_{max} = q(1 - \frac{T_0}{T_H}) \quad (19)$$

where T_0 is the atmospheric temperature in K.
If the temperature of the source of heat in a heating procedure reductions from T_H to T_L the exergy variation of the heat flow is

$$\Delta e_x = \int_{T_L}^{T_L} \left(1 - \frac{T_H}{T_L}\right) dq = q - T_0 \int_{T_L}^{T_H} \frac{cdT}{T} = q - T_0 c \ ln \frac{T_H}{T_L} = c(T_H - T_L) - T_0 c \ ln \frac{T_H}{T_L} \quad (20)$$

where c is the mean specific capacity of heat of the stream in the temperature variation of T_L -T_H .This change of exergy between the temperature-varying source of heat can also be stated roughly by a simple form

$$\Delta e_x \approx q(1 - \frac{T_0}{\frac{T_L+T_H}{2}}) \tag{21}$$

For simplicity of the design and grip the key points, adopt that the steam extracted in the scheme shown in Fig. 1 is from where the system has extended to a saturation vapour pressure where it is at the state or from a lower pressure state, so the temperature of the extracted steam ruins constant when it transmissions heat to the feedwater although the temperature of the feedwater rises.

Supposing the specific capacity of heat of the feedwater c is constant through the heating process, the ratio of exergy rise E_X to heat Q gained by feedwater (represented by subscript "w") is:

$$(\frac{E_X}{Q})_W = (\frac{e_x}{q})_w = \frac{\int_{T_L}^{T_H}\left(1-\frac{T_0}{T_L}\right)dq}{h_H - h_L} = 1 - \frac{T_0}{T_H - T_L}\ln\frac{T_H}{T_L} \approx 1 - \frac{T_0}{\frac{T_H+T_L}{2}} \tag{22}$$

The ratio of exergy E_x of the steam extracted from the turbine to the heat Q (at the constant temperature T_H, represented by subscript ``v'') is

$$(\frac{E_x}{Q})_w = 1 - \frac{T_0}{T_H} \tag{23}$$

In a well-protected feed heater, the rejected heat by the steam extracted Q_v should be equivalent to the absorbed heat by the feedwater Q_w (i.e. Q_v=Q_w). With adding, the exergy of the extracted steam is very close to the work the steam can do in the steam turbine. So

$$\frac{(\frac{E_x}{Q})_v}{(\frac{E_x}{Q})_w} = \frac{Ex_v}{Ex_w} = \frac{w}{Ex_w} \tag{24}$$

From the above equation it can be said that if we give the feedwater with the equal quantity of heat from nonconventional energy like solar energy another place of the extracted steam, the protected steam can do W amount of work. In the idyllic case, the feedwater obtains the equal quantity of exergy as the solar heat transporter discharges, nearly expresses the ratio of the work we can save to the exergy that has to be funded by solar energy.

3.3.2 An Example of multi-stage regenerative system

Here an example of three stage regenerative system is taken where three solar heaters are installed with three regenerative heaters and one solar heater is installed with the down comer.

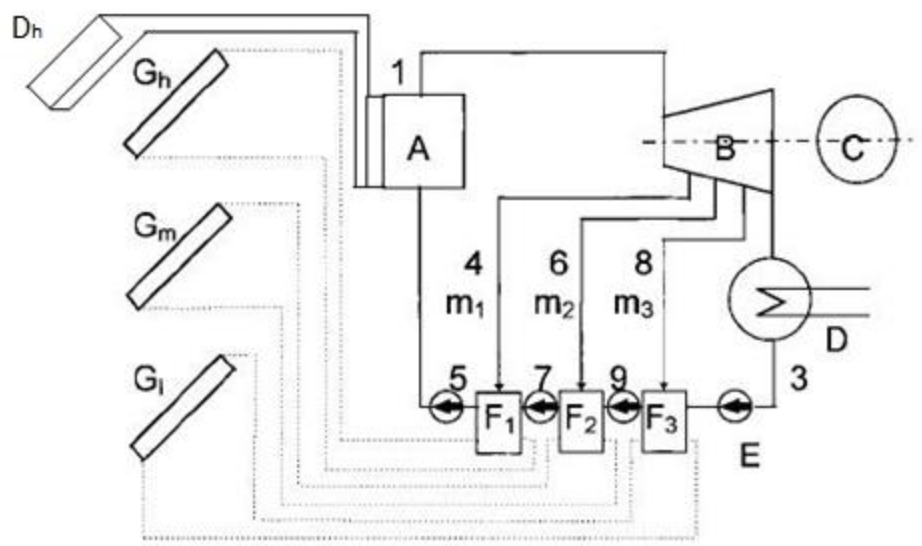


Fig.3.11. A three-stages regenerative steam Rankine cycle. A is Boiler and superheater; B- is turbine; C is generator; D is condenser; E- pump; F1, F2, F3 are feedwater heaters; G_l is low-temperature collector; G_m is medium-temperature collector; G_h is high-temperature collector and D_h is solar heater installed with down comer.

Some input data taken from Bakreshwar thermal power plant for inputs for calculation in EES software

Point	**Pressure(Kpa)**	**Temperature(°c)**
Inlet of high pressure turbine	16000	538
Outlet of low pressure turbine	101.3	40
Extracted steam from high pressure turbine	5500	369
Extracted steam from medium pressure turbine	900	275
Extracted steam from low pressure turbine	800	40

Mass of steam extracted from high pressure turbine ($\dot{m}_1$) = 0.184 kg/s
Mass of steam extracted from medium pressure turbine ($\dot{m}_2$) = 0.115 kg/s
Mass of steam extracted from low pressure turbine ($\dot{m}_3$) = 0.0715 kg/s

3.4 Economic Model

The economic model assists in making easy decisions by weighing the hybrid annualized power plant performance against the incurred costs thus making it simple for the development undertaker if a given method of integrating solar power with conventional coal plant is worth implementing. For simplifying this decision making process, certain economic merit comparison criteria have to be set alongside with the necessary specific emission reduction criteria. Before the economic figures of merit can be calculated, the cost of each component of the power plant has to be estimated for by considering size, performance and time factors.

In the coming sections, cost elements pertaining to general plant components belonging to both parabolic trough and central receiver tower based hybrid coal-fired plants will be presented. Cost estimates and functions belonging to either of the power systems will be discussed under separate sections.

3.4.1 Cost Modeling and Estimation for Parabolic Trough Plant

Currently, there is not much literature available in formulating actual investment costs to help estimate the cost of parabolic trough plants. Alternatively, there are some researches which present the various plant cost element estimates pertaining to construction and deployment of this technology for reference plants.

The parabolic plant cost modeling process, used in this thesis, follows NREL's physical cost model for parabolic trough power plant. Owing to the PTC technological maturity and its modular nature, the cost of parabolic plants can be calculated with small margin of error to the actual field cost. Accordingly, C. Turchi's [32] cost breakdown estimates will be used for wet cooled solar plants. Cost elements that don't relate to hybrid solar-coal plant operation such as the power block cost and the associated engineering costs are excluded towards the final specific cost of energy calculations.

The land cost for parabolic trough field is based on the land requirement set by the design points selected for the solar site. Aperture area required to raise the temperature of the HTF to a field outlet temperature is determined for a given solar loop based on LS-2 collector design. The energy collected from a single solar loop is subsequently used to calculate the total field aperture area for nominal solar power output. The nominal aperture area is multiplied by optimum solar multiple to ensure longer design point power output of the power cycle.

According to Pai [5], the actual land area requirement is three times the total collector aperture area including the area needed to include power house and other supportive buildings. On the other hand, H. Price [32] puts that the land requirement to about 2.33 times its design aperture value. Since the land requirement for power house is avoided by utilizing the power house for the main coal plant, a sensible assumption to this multiplier would be 2 times its design value. Thus, the land cost can be calculated as,

$$C_{land} = 2.SM.C_{land}.A_{collector}.N_{collector}.N_{loop} \quad (25)$$

Where: SM=solar multiple, C_{land} =specific cost of land=2 \$/m2, $A_{collector}$= aperture area of a single collector, $N_{collector}$ = number of collectors in a single loop, N_{loop}= Number of loops in the solar field at design aperture area.

Plant labor cost is estimated from the work force requirement for operational control and field maintenance sections.

The number of field maintenance technicians,$N_{technician}$, required for maintaining PTC field [33] is defined by the total collector aperture area ($A_{PTC.}N_{collector}$)and the required number of personnel needed for a specified area of the aperture, $N_{personnel-tech}$

$$N_{technician} = 1 + N_{personnel-tech}.A_{PTC}.N_{collectors} \tag{26}$$

In the same way, the required number of control operators is obtained as,

$$N_{operator} = 1 + N_{personnel-oprtr}.A_{PTC}.N_{collectors} \tag{27}$$

Where: $N_{personnel-tech} = 2$ persons/100,000 m^2 of aperture area and $N_{personnel-oprtr}$ =3 persons/100,000 m^2 of aperture area.

There is no need to include power block maintenance labour cost as the PTC solar plant is operated in tandem with the main coal plant hence; the maintenance labour cost is calculated as part of the main coal plant cost. In addition, the number of administrative workers is taken to be the same irrespective of the size of solar plant. The average salary of the respective job positions is enumerated in the following table.

Table 3.4 Average salary of respective job positions at the PTC solar plant [34]

Cost Type	Labour Requirement	Salary for single employee [$/year]
Administration	7	62857
Operating cost	$N_{operator}$	57385
Solar field maintenance	$N_{operator}$	55857

Water cost attributed mirror cleaning is determined from specific water consumption of mirror surface multiplied by the total field aperture area. According to Daggett PTC plant in California [35], the amount of water used in cleaning mirror surface is about 0.81 liters/m^2 in every wash with about 63 washes throughout the year. Based on this assumption, the total annual volume consumption of water is estimated to be 51 liters/m^2 .

$C_{water} = 0.051 \frac{m^3}{m^2-yr} . A_{aperarate} . C_{water}$ (28)

Where: $A_{aperarate}$ = total effective mirror aperture area $[m^2]$ C_{water} =cost of water per m^3 of volume=1.1$/m^3$

Table 3.5 Cost elements and assumptions taken for PTC plant

Cost Types	Inclusive costs	Estimate	Reference
Direct Capital Cost			
Site Improvements	land preparation, roads, fences, and site infrastructures ,such as firewater system, warehouse, and control building	15 $/$m^2$	[32]
Solar Field	reflective support structures mirrors, tracking system, receiver module, header piping ,inter collector piping	295 $/$m^2$	[32]
HTF system	HTF pumps, solar heat exchangers, HTF expansion, vessel, piping, valves, and instrumentation	90 $/$m^2$	[32]
Storage	heat exchanger, hot and cold tank mechanical equipment (pumps and heat exchangers),solar, salt, piping, instrumentation and electrical, and civil and structural	30 $/KW thermal	[33]
Balance of Plant	cooling towers, water treatment and storage, electrical, and control systems.	7.5 % of Direct Cost	[33]
Contingency	Unaccounted costs in solar field, Thermal storage, structures and improvements depending on solar construction site.	20 % on site improveme nt, 5% on solar field	[33]

		cost, 10 % on storage tank costs	
Decommissioning	Site restoration, environmental fines	5 % of the Direct cost	[34]
Indirect Capital Costs			
Land	Owner and Environmental protection cost	2 $\$/m^2$	[9]
Sales tax	Tax on non-labor portion of the direct cost upon procurement	No capital acquisition Tax	[36]
Operation and Maintenance Costs			
Equipment and spare parts	Receiver glass replacement, degraded HTF replacement, Heat Exchanger cleaning	0.4 % of Direct Cost/yr.	[36]
Mirror cleaning	Water, cleaning equipment and demineralizing costs	1.1 $\$/m^3$	[36]

3.4.2 Cost modeling for main coal plant

In this subsection, cost modeling for reference coal-fired power plant is presented. The purpose of estimating the total capital, operating and maintenance cost is useful for providing economical comparison benchmark to every solar repower option being investigated. The fossil only plant is compared based on its specific fuel consumption, emission rate and most importantly, based on the cost of energy on a unit of energy output. The underlying assumption for economic calculations on the coal only plant is that the plant operates at nominal conditions throughout the entire year except for the time of yearly overhaul. Costs of plant components are determined by referring to these nominal operating conditions.

Table 3.6 Cost estimates for 210 MW Bakreshwar thermal power plant [30]

Cost type	Cost element	Amount in $
	-Steam Turbine -Cost increase due to reheat for piping, valves and controls	257,541,962
	Heat exchangers	55,632,148
	Piping	110,757,851
	Deaerator	1,081,355
	Condenser	33,943,068
	Cooling tower	9,565,881
	Pump -Condenser feed water pump -Boiler feed pump -Condenser water pump	 65,057 779,356 418,897
	Pipe and auxiliary cost	108,093,478
	Contingencies	18,097,065
	Land	9,960,000
	Civil engineering and construction cost	18,097,065
	Decommissioning	18,097,065
Operating Cost	Coal	104,942,530
	Maintenance and Spare parts	5,868,392
	Labor	8,800,000
	CO2 tax	33,806,186
	Environmental tax	67,067,333

CHAPTER 4

4. Result and Discussion

The objective of this paper is to determine the best possible way to install the solar heaters in the coal based power plant. Here it is shown that where we can install the solar heaters and what will be the effects of that on the performance characteristics of the main parameters of the plant if we installed the solar heater on that position. Here we also analyze how the plant performance effected by effecting the solar radiation intensity.

4.1 Component Analysis

Here the study is based on the input parameters which we taken from table 3.4 and by varying this parameters how the performance of the each component of the plant varies. We are calculating all the parameter taking the mass flow rate of is 1kg/s.

4.1.1 Variation of maximum boiler pressure:

Table 4.1 Variation of solar aided work, rankine cycle work, first law efficiency of solar aided and rankine cycle work with maximum boiler pressure.

Maximum boiler pressure [KPa]	Solar-aided specific work(KJ/Kg)	Rankine cycle specific work(KJ/Kg)	First law efficiency of rankine cycle	First law efficiency of solar aided cycle
10000	865.6	724.7	0.3374	0.4029
11111	887.9	752.9	0.3472	0.4095
12222	907	777.3	0.3558	0.4152
13333	923.7	798.6	0.3634	0.4204
14444	938.3	817.5	0.3703	0.425
15556	951.1	834.3	0.3765	0.4292
16667	962.5	849.4	0.3822	0.4331
17778	972.6	862.9	0.3875	0.4367
18889	981.5	875.1	0.3923	0.4401
20000	989.6	886.1	0.3969	0.4432

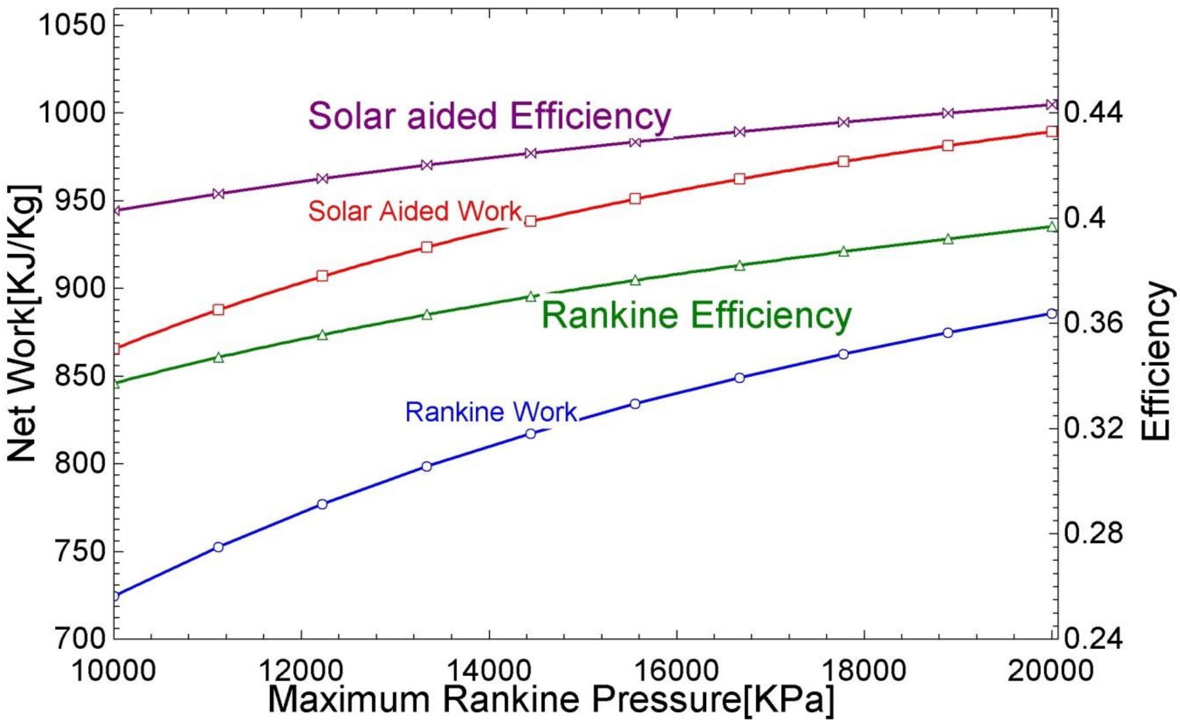


Figure 4.1 efficiency vs net specific work vs maximum rankine pressure

From the table 4.1 and figure 4.1 it is seen that as the maximum rankine pressure is increased the net specific work out for rankine cycle and solar aided rankine cycle is increased. Because as pressure increases the saturation temperature of water is also increased, for that reason the mean temperature of heat addition is also increased, that is why net specific work output and efficiency of both the cycle is increased. But the value of rankine work is less than the value of solar-aided work because in rankine work we use bled steam for heating the feed water heater, thus the mass of steam in the turbine is decreased, so work done by that much of steam we loosed. But in solar aided cycle that much of steam we can use to produce work in the turbine, for reason the value of work done by rankine cycle is less than the solar-aided cycle.

4.1.2 Variation with boiler outlet temperature:

Table 4.2 Variation of solar aided work, rankine cycle work, first law efficiency of solar aided and rankine cycle work with the steam temperature at boiler exit.

Maximum HPT inlet Temperature [K]	Rankine cycle specific work(KJ/Kg)	Solar-aided Specific Work Output	First law efficiency of rankine cycle	First law efficiency of solar aided cycle
550	576.6	620.7	0.2403	0.2586
582.9	611.3	664.7	0.2547	0.277
615.8	645.6	708.1	0.269	0.295
648.6	679.1	750.6	0.2829	0.3128
681.5	712	792.5	0.2967	0.3302
714.4	744.5	833.8	0.3102	0.3474
747.3	776.6	874.8	0.3236	0.3645
780.1	808.5	915.5	0.3369	0.3814
813	840.1	955.9	0.35	0.3983

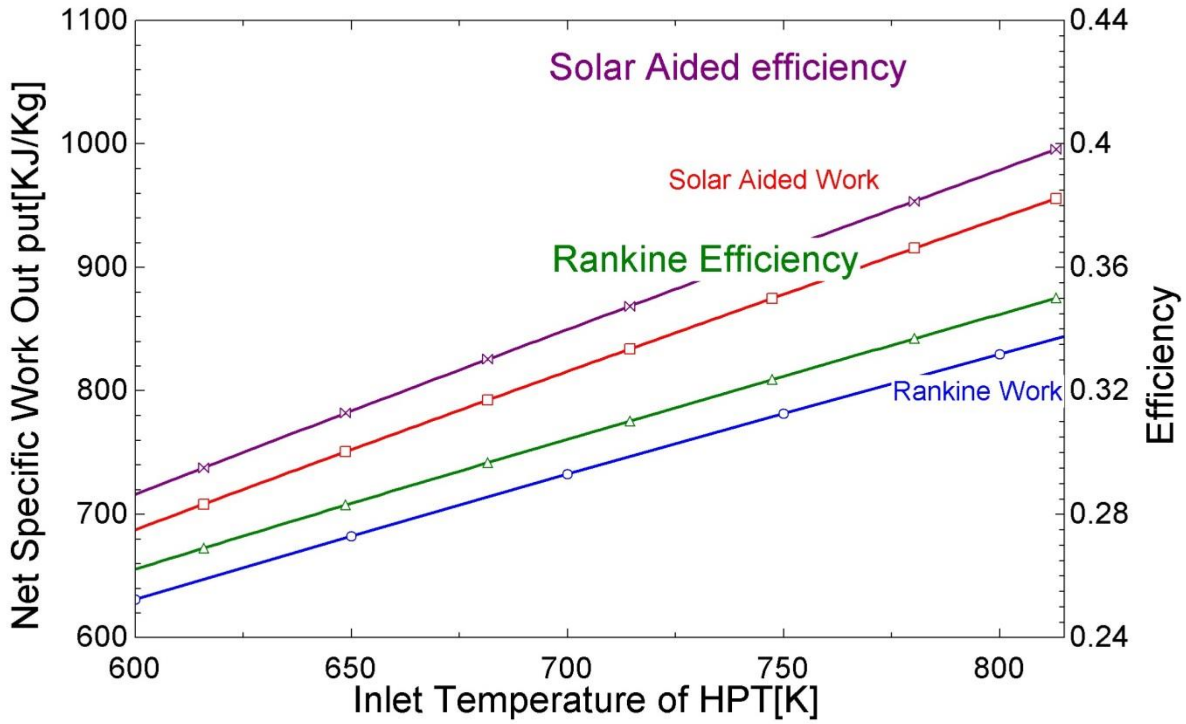


Figure 4.2 Inlet Temperature of high pressure turbine vs net specific work output vs efficiency

From table 4.2 and figure 4.2 it can be said that if inlet temperature to the high pressure turbine is increased then the net specific work for both rankine and solar aided cycle is increased because as we supply more heat so work output is also increased. As the mean temperature of heat addition is increased so the efficiency of both the cycle will increased. As we did not use the extracted turbine steam to the feedwater heater so the steam is used in power generation in the case of solar-aided cycle so the value of net specific work and efficiency both will be more for solar aided cycle than the rankine cycle.

4.1.3 Variation of heat capacity of the solar heater installed in down comer:

Table 4.3 variation of heat needed for combustion with the change of solar heater capacity installed with down comer.

Solar heater capacity installed in down comer[KJ]	Variation of combustion heat[J]
10	2390
11	2389
12	2388
13	2387
14	2386
15	2385
16	2384
17	2383
18	2382
19	2381
20	2380

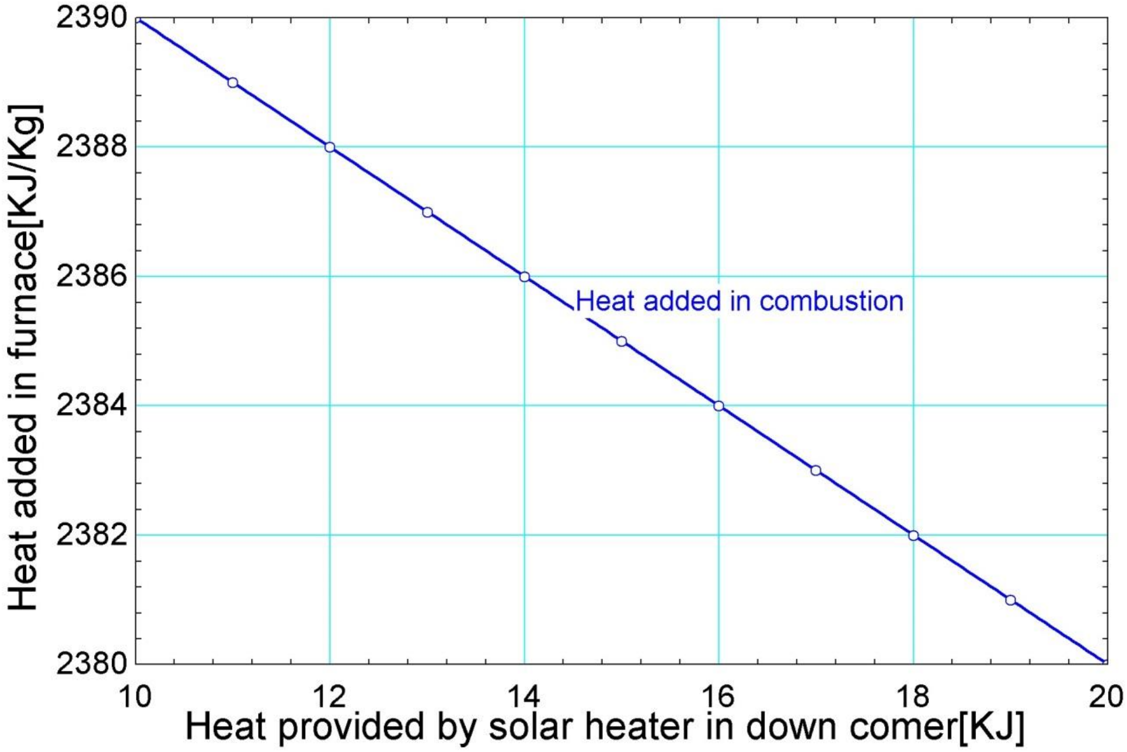


Figure 4.3 variation of heat added in furnace with heat capacity of solar heater installed in down comer.

From the table 4.3 and figure 4.3 it is clear that if the capacity of solar heater installed near down comer is increased then heat required in furnace is decreased. The temperature of at boiler outlet should be constant as of metallurgical limits of turbine blades. As the heat added by solar heater installed in down comer is increased the temperature of water coming from boiler drum so we need less heat at furnace for sensible heating of water.

4.1.4 Variation of required area of solar heater with solar intensity:

Table 4.4 Variation of required solar heater area with the variation of solar intensity

Solar radiation intensity(kw/m^2)	Solar heater area installed with medium pressure heater(m^2)	Solar heater area installed with low pressure heater (m^2)	Solar heater area installed with downcomer pressure heater (m^2)	Solar heater area installed with high pressure heater (m^2)
0.5	20384	4278	305313	31200
0.5333	19110	4011	286231	29250
0.5667	17986	3775	269394	27529
0.6	16987	3565	254427	26000
0.6333	16093	3377	241037	24632
0.6667	15288	3209	228985	23400
0.7	14560	3056	218081	22286
0.7333	13898	2917	208168	21273
0.7667	13294	2790	199117	20348
0.8	12740	2674	190821	19500

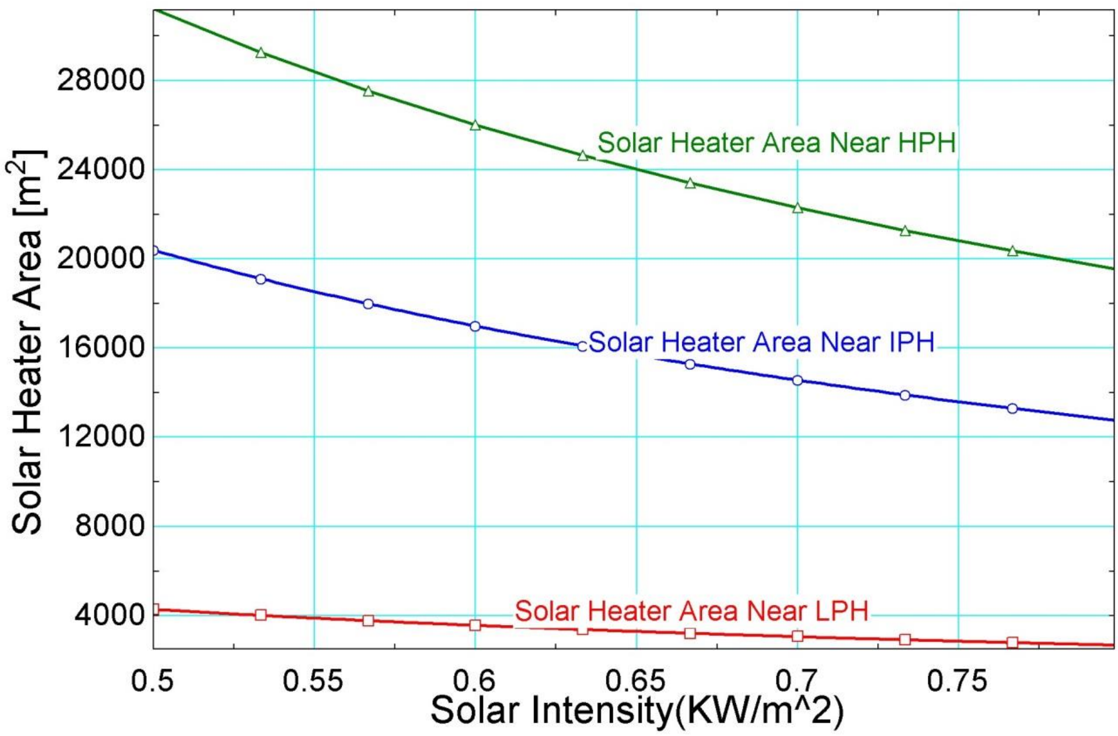


Figure 4.4 Solar intensity vs Solar heater area

The area of each solar heater is 235 m^2 and in each row we take 16 solar heaters and there is 12 rows. As the solar intensity increases we need less amount of energy for the same power out put. That time the required heater of heater should calculated and rest amount of heaters should be covered. So in the time of winter season when the solar radiation is less that time we should uncover all the rows and in full shine days or summer days we should cover some no of rows according to the solar intensity.

4.1.5 Variation of coal consumption rate with solar intensity:

Table 4.5 Variation of coal consumption rate with solar intensity

Solar intensity(w/m^2)	Coal consumption rate of normal thermal power plant(g/Kwh)	Coal consumption rate of solar-aided thermal power plant(g/Kwh)
0.5	704.4	619.2
0.5333	704.4	580.5
0.5667	704.4	546.4
0.6	704.4	516
0.6333	704.4	488.9
0.6667	704.4	464.4
0.7	704.4	442.3
0.7333	704.4	422.2
0.7667	704.4	403.9
0.8	704.4	387

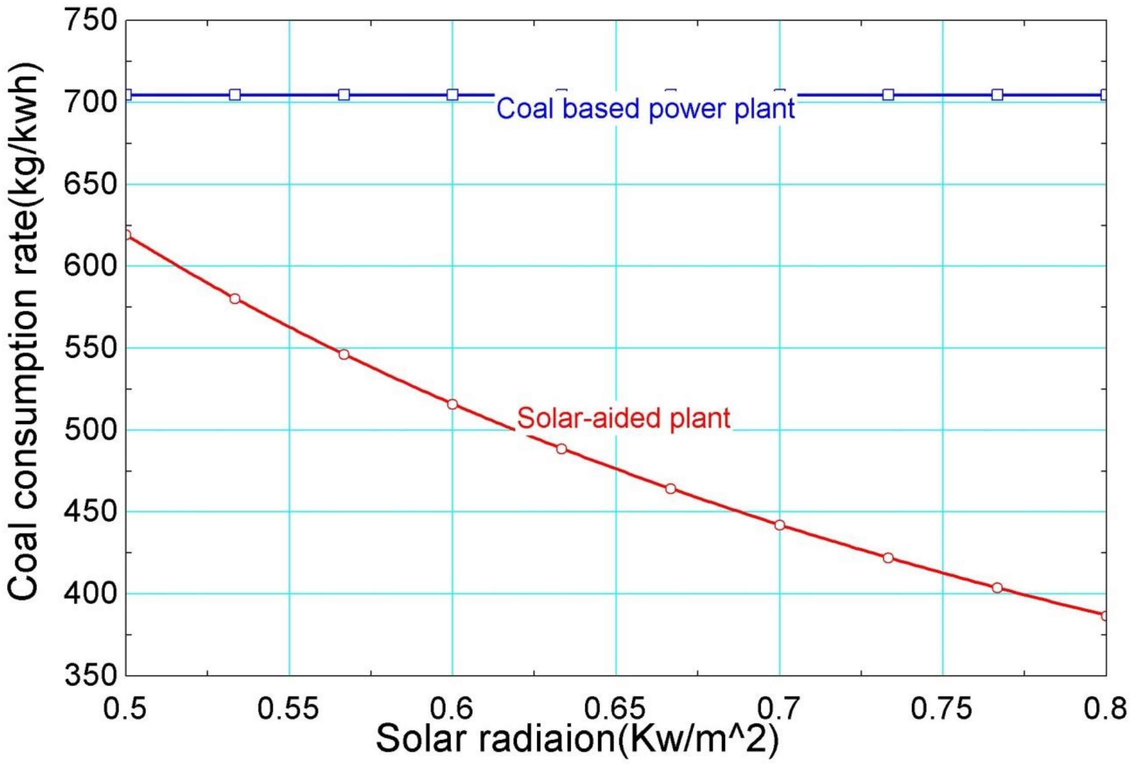


Figure 4.5 Coal consumption rate vs solar radiation intensity

As coal based power plant did not depend on the solar intensity so coal consumption rate for coal based power plant will be constant with solar radiation. But as the radiation is increased, solar heater will start to supply heat to the feed water as well as water comes from boiler. As a result the coal consumption rate will decreased for same power output because we have to supply less amount of heat through fossil fuels

4.1.6 Variation of coal consumption rate with heat added by solar heater:

Table 4.6 Variation of coal consumption rate with heat added by solar heater

Coal consumption rate(Kg/kw-h)	Heat supplied by solar heater installed with high pressure heater(KJ)	Heat supplied by solar heater installed with medium pressure heater(KJ)	Heat supplied by solar heater installed with low pressure heater(KJ)
600	2428	2403	2341
650	2419	2402	2319
700	2410	2401	2296
750	2401	2399	2274
800	2392	2398	2252
850	2383	2397	2230
900	2373	2395	2208
950	2364	2394	2186
1000	2354	2393	2164

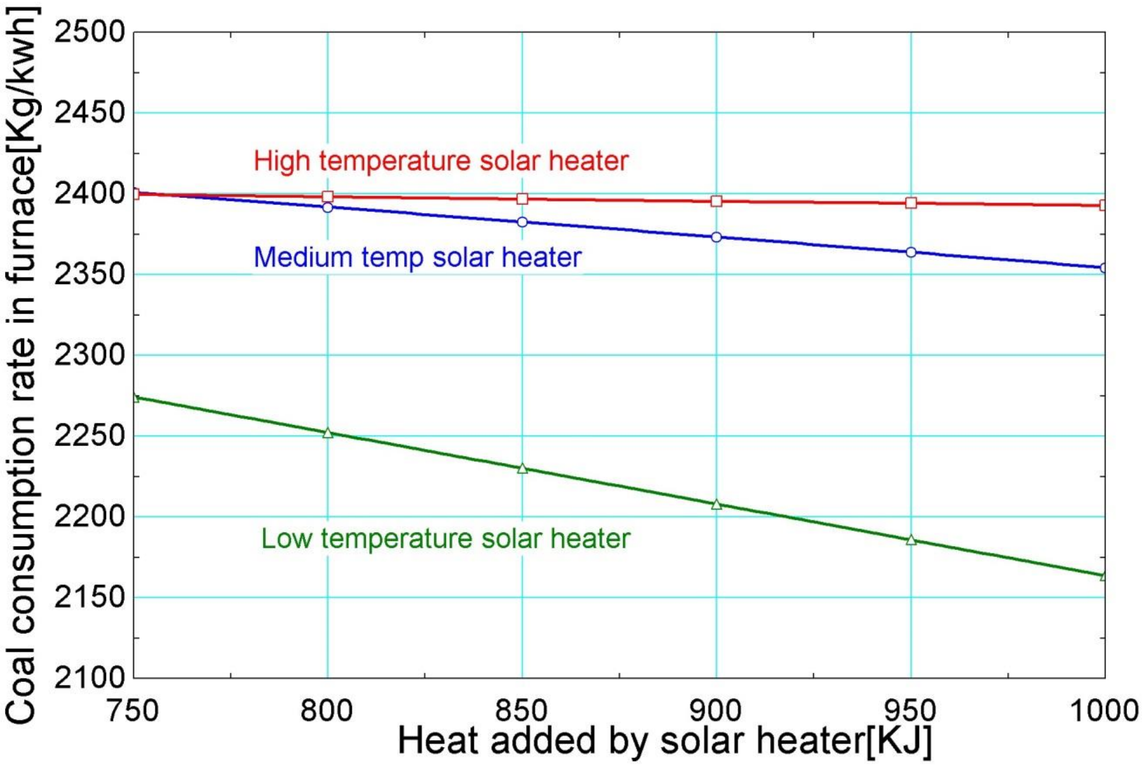


Figure 4.6 Coal consumption rate in furnace vs heat added by solar heater installed with feed water heater.

From the table 4.6 and figure 4.6 it is seen that as heat added by solar heater is increased the coal consumption rate is decreased because the as the heated supplied by the solar heater is increased the feed water consumes more heat from solar heater. As a result we have to supply less heat in the boiler so, coal consumption rate will also decreased. Thus the use of fossil fuel is deceased.

4.1.7 Variation of mass flow rate of steam extracted from HPT:

Table 4.7 Variation of mass flow rate of steam extracted from HPT with heat supplied by solar heaters and net specific work done and first law efficiency of both the cycle.

Mass flow rate of steam extracted from HPT(kg/s)	Heat supplied by solar heater installed with high pressure heater(KJ)	Heat supplied by solar heater installed with medium pressure heater(KJ)	Heat supplied by solar heater installed with low pressure heater(KJ)	Heat supplied by solar heater installed with downcomer (KJ)	Rankine cycle specific work(KJ/Kg)	Solar-aided specific work(KJ/Kg)	First law efficiency of rankine cycle	First law efficiency of solar aided cycle
0.05	10.53	1.879	50.92	15.6	913.2	976.7	0.3805	0.4069
0.1	10.41	1.968	93.96	15.6	886.1	971.5	0.3692	0.4048
0.15	10.28	2.066	131	15.6	859	963.2	0.3579	0.4013
0.2	10.15	2.175	161.9	15.6	831.9	951.9	0.3466	0.3966
0.25	10	2.297	186.8	15.6	804.8	937.5	0.3353	0.3906

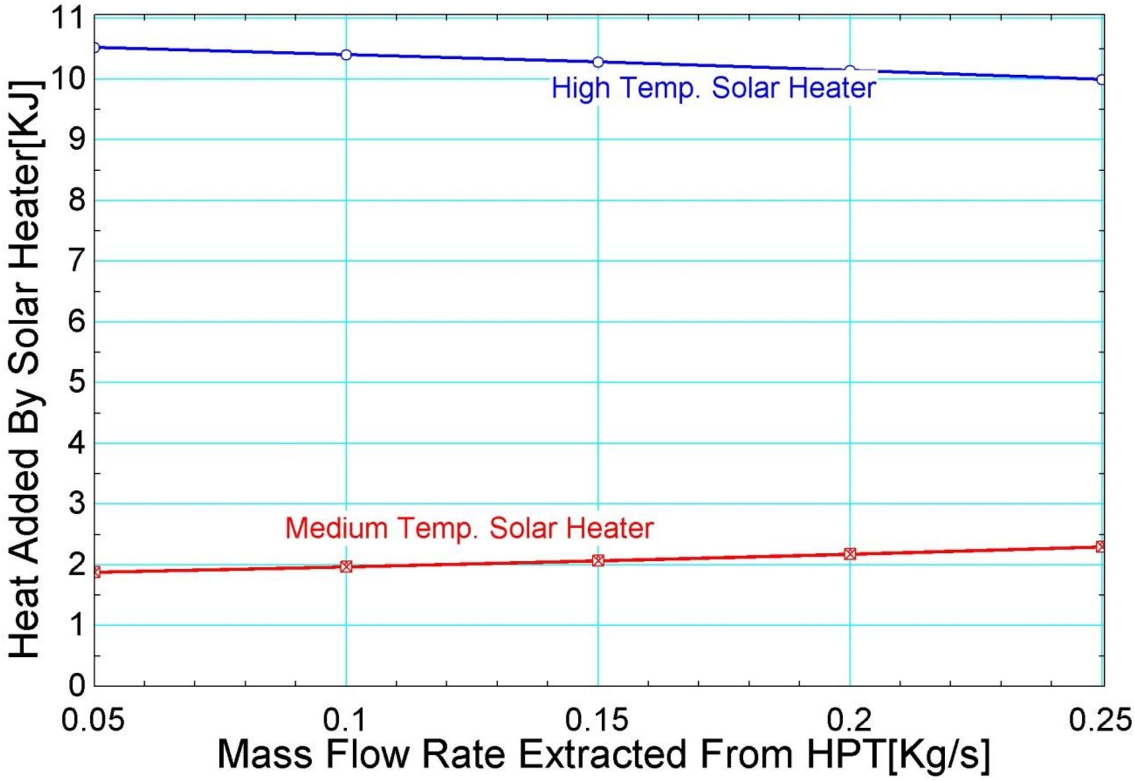


Figure 4.7 mass flow rate vs heat added by solar heater

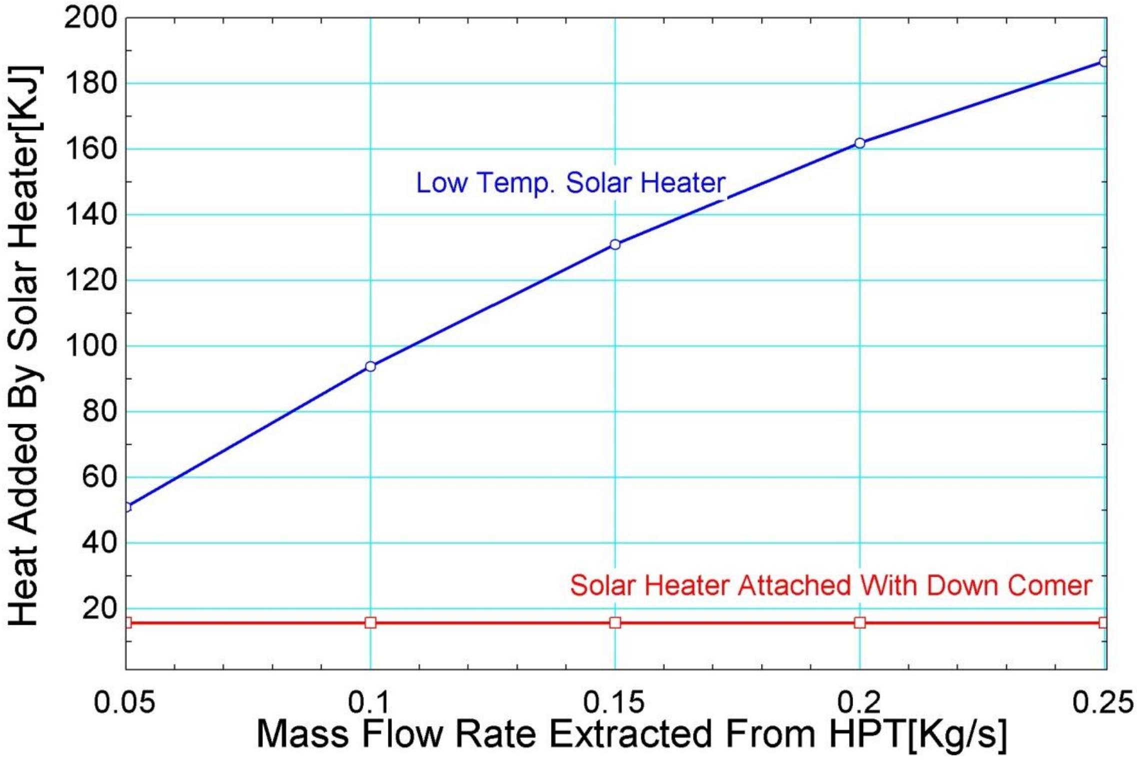


Figure 4.8 Heat added by solar heater vs mass flow rate extracted from HPT

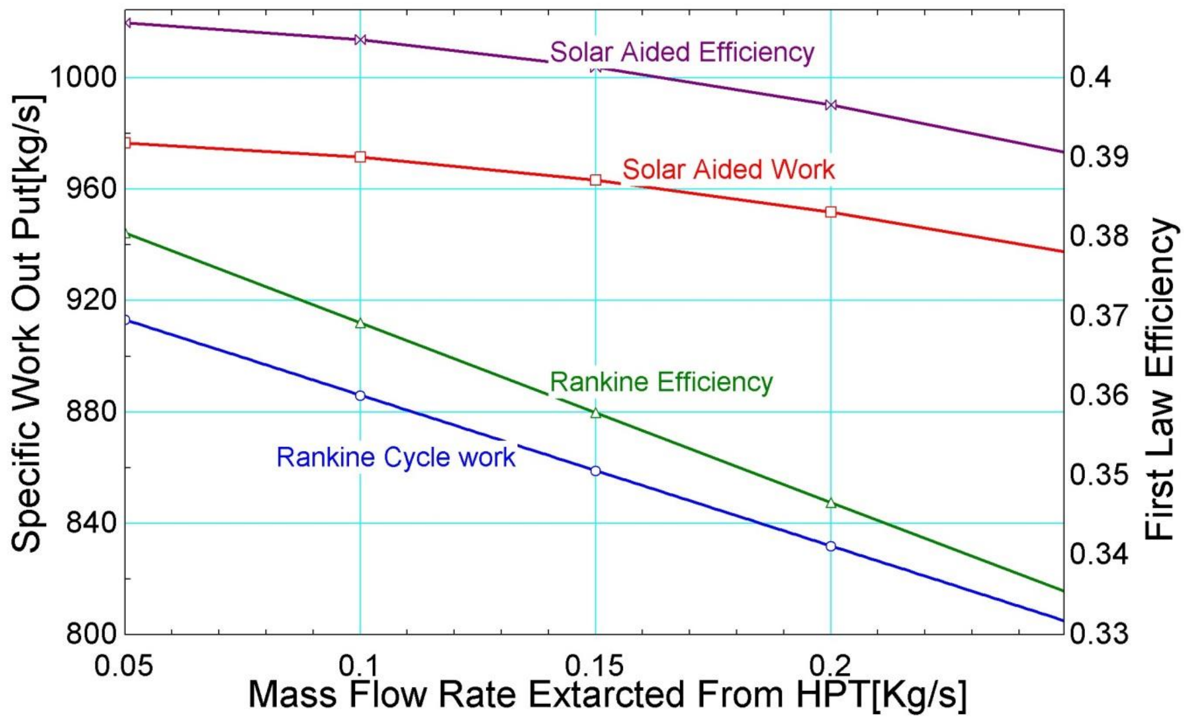


Figure 4.9 Specific work out put vs first law efficiency vs mass flow rate extracted from HPT

We have to face some condition when we after uncovering all the heaters have to extract some amount of steam from the turbines. In that case performance characteristics obtained is shown in figure 4.8 and 4.9. As mass flow rate extracted from the HPT is increased so HPH get more heat from the steam carrying heat. As a result we need to supply less temperature from the high temperature solar heater as a result as we increased the mass flow rate extraction, the heat added by high temperature heater is get decreased. But the opposite result will occur in the case of low temperature heater and medium temperature heater. As mass extracted from HPT increased so that heat carried by steam coming from HPT increased so that mass flow rate extracted from IPT and LPT is decreased so that carried by steam coming from IPT and LPT is very less so heat carried by steam coming from IPT and LPT is very less so solar has to supply more heat. That is why as mass flow rate extracted from HPT increased heat added by medium pressure heater and low pressure heater is increased.

In figure 4.9 it is seen that slope of work in solar aided rankine cycle is less means if we used solar heater then loss of work in solar aided rankine cycle was less than conventional rankine cycle.

4.2 Variation of cost of solar heater with variation in solar radiation intensity:

Table 4.8 Variations of cost of solar heater with variation in solar radiation intensity.

Solar Radiation Intensity(KW/m^2)	Cost of Solar Heater(USD)
0.5	16479
0.5333	15449
0.5667	14541
0.6	13733
0.6333	13010
0.6667	12359
0.7	11771
0.7333	11236
0.7667	10747
0.8	10300

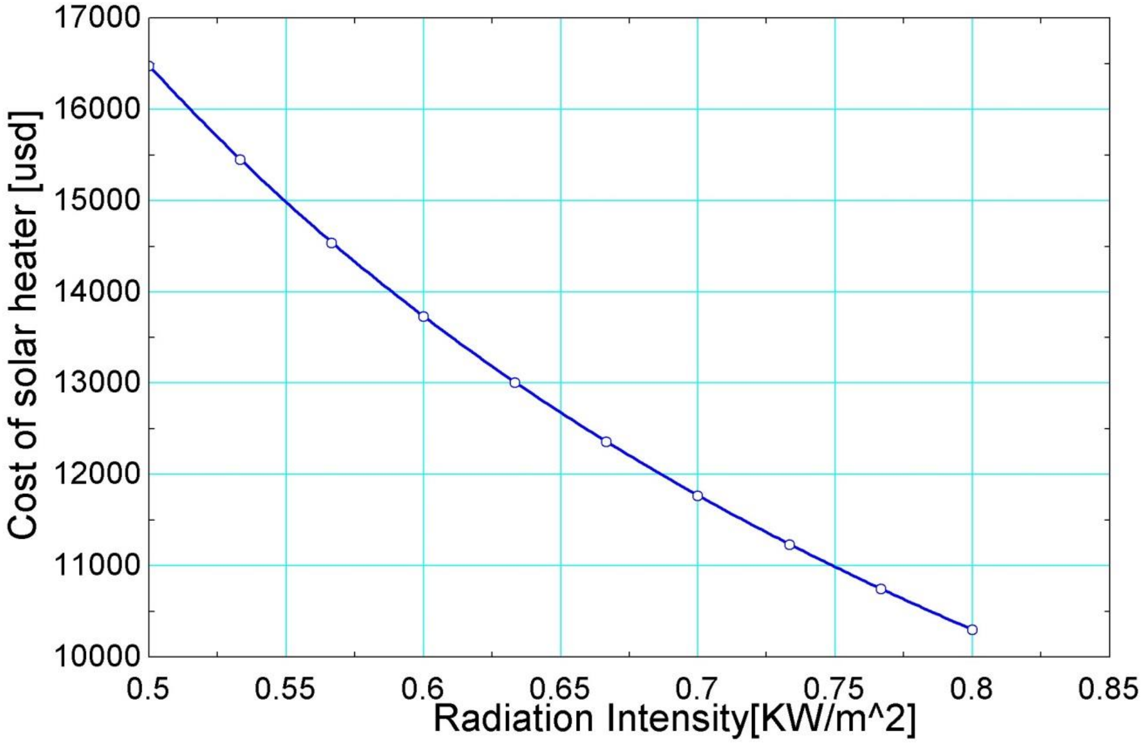


Figure 4.10 Solar radiation intensity vs cost of solar heater

From the table 4.8 and figure 4.10 we can conclude that as the radiation intensity increase we need less no of solar heater in use. Maximum solar heater we have keep in cover as solar radiation increases. So maintenance cost of the solar heater will decreases. As a result overall cost of solar heater is decreased.

4.3 Discussion on compensating the extra cost invest to introduced solar in conventional thermal power plant

The area required for each solar heater is 235 m^2 and we need about 42971 m^2 area to install solar heater .So the total cost for land and heater will be 644565 USD and 1276445 USD. To compensate this huge amount we need payback period of 10 years according to present days rate of electricity. These figures are calculated by EES software. But after 10 yrs plant can be run at same profit as it is running in present days and the pollution and saving of fossil fuels also be achieved by this technology.

CHAPTER 5

5.1 CONCLUSIONS:

This thesis demonstrated the technical and economic potentials of integrating solar thermal power with the widely utilized coal thermal plants. Four different integrations points using one principal technology were initially compared on thermodynamic sense whereby to show that superior annual performance were given further economic analysis.

In this work, the First and Second Laws of Thermodynamics have been used to study the performance of a solar-aided power plant. The main parameters such as the solar irradiation and power output have also been varied to analyse their effects to the plants. The main conclusions are as follows:

(1) When the solar irradiation is 650 W/m^2 and load ratio of the unit is 100%, the first law efficiency of the base plant is 34.9%, lower than the first law efficiency of the solar-aided thermal power plant 39.7% and the new plant saves about 85.2 g/kWh of coal.

(2) When the unit is operated in full load, (*i.e.*, with output of 210 MW), the solar irradiation differs from 500 W/m^2 to 800 W/m^2, the coal consumption rate reduces from 619.8 g/kWh to 387.6 g/kWh, saving coal from 85.2 g/kWh to 317.4 g/kWh.

(3) When the capacity of Solar heater installed in down comer varies from 10KJ to 20KJ the heat supplied by furnace is also decreases from 2390 J to 2380 J. If we do not install heater then energy supplied by furnace is 3000 J. This indicates that we are saving 20 J of energy supplied from the furnace by installing a solar heater in down comer.

(4) About 10 years of payback period is needed to compensate the extra investment cost to install solar heater. Besides saving of fossil fuel and pollution control also achieved by this process with lower environmental issues.

CHAPTER 6

6.1 FUTURE SCOPE:

Based on the above analysis, in order to improve the efficiencies of a solar-aided coal-fired power plant, extra modifications can be made to the boiler, condenser and solar-aided water heat exchanger. In the design of a new plant; in the plant operation, solar irradiation and load demand from the customers should be considered. The effective predictions of solar irradiation and load demands can be make the unit work in a high efficiency range, which can be useful to improve the performance of the plant. There are two parameters (i,e solar irradiation and load variation) in the parametric study, discussions can be made based on keeping one parameter unchanged while the other changed. So in future research, the analysis based on the change of both the solar irradiation and load demand can be made to obtain an optimal operating strategy for the system.

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