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Exergy and Thermoeconomic Analyses of Solar Aided Thermal Power Plants with Storage-A Review

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Exergy and Thermoeconomic Analyses of Solar Aided Thermal Power Plants with Storage-A Review

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Abstract- Ever increasing energy demand, spiralling fuel prices dwindling resources and emissions foot print of fossil fuel based power generation has forced the world to increase the share of renewable energy based power generation. Out of all renewable energy sources (RES), solar has emerged as a viable option for addressing several challenges being faced by the power generation industry currently. Solar PV and solar thermal are the two options for solar based power generation. Although Solar PV provides excellent energy solutions for small scale grid and off grid power generation, it is not suitable for large scale power generation. Now within solar thermal, solar alone power generation has not gained popularity due to high capital costs and poor thermal efficiency but solar integration with existing/new power plants (both coal fired thermal and gas fired combined cycle) popularly known as solar aided power generation has been widely accepted by several researchers. This technology has been widely in use in countries like U.S, Spain, Egypt, Morocco, Algeria, Iran and Mexico (NREL web site).Due to lower capital costs and better solar to electricity conversion efficiencies, solar aided thermal power plants have surpassed solar alone power plants. Solar energy can be integrated in a coal fired plant either for steam generation or for feed water heating. In a combined cycle plant, solar energy can be integrated either in the Brayton cycle or in the bottoming Rankine cycle. This chapter discusses in detail, various integration techniques, the application of exergy, economic and thermo economic principles in analysing these integrated cycles. Detailed economic analysis has been reviewed on the integrated cycles to ascertain their techno economic viability. The levelised cost of electricity generation (LCoE) and simple payback period have also been predicted for both the reference and integrated plants.

Keywords: solar aided feed water heating, integrated solar combined cycle, exergy analysis, levelised cost of electricity generation, simple payback period.

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I. Introduction

The usage of solar energy for power generation has been widely considered as a promising solution for reducing fossil fuel dependency, emissions footprint. For sustainable development and as a move towards greener power generation, the usage of solar energy has gained prominence across the countries having good solar potential. The solar energy can be converted in to electricity either through solar photovoltaic (PV) technology or through solar thermal power generation technology. However, solar PV is mostly suitable for distributed power generation due to its small scale generation capacities. For large scale power generation, solar thermal is better suitable than solar PV. Within the solar thermal power generation, one may go for either solar alone or solar aided power generation. In the solar alone thermal power generation, the concentrated solar energy is imparted either directly to working fluid or via a heat transfer fluid for steam generation. The generated steam can be used in the power block for power generation. However, the solar alone power generation suffers from higher capital costs, lower solar to electric conversion efficiencies and lower plant utilisation factors. The low solar to electric conversion efficiencies of solar alone systems can be attributed to low source temperature of cycle heat addition. The lower plant utilisation factors can be attributed to daily plant start-up and shut downs. Solar aided power generation in coal fired steam power plants involve adding the solar heat in an existing Rankine cycle either for additional steam generation or feed water preheating. The latter is popularly known as solar aided feed water heating (SAFWH). In a combined cycle power plant, solar energy can be added either in the topping Brayton cycle or bottoming Rankine (steam) cycle. These cycles are referred to as integrated solar combined cycle (ISCC) power plants. In the Brayton cycle, solar energy can be used to lower the gas turbine compressor inlet air temperature or it can be used for heating the compressor discharge air. In the bottoming cycle, the solar energy can be used for generating additional steam which increases the steam turbine power output. This chapter presents a bird's eye view of solar aided thermal power generation.

II. Review On Solar Collector And Storage Technologies

Concentrating solar power (CSP) technologies can be classified based on focus geometry as either line-focus concentrators (parabolic trough collectors and linear Fresnel collectors) or as point-focus concentrators (central receiver systems, parabolic dishes and Scheffler systems)[1].The line focus is less expensive and technically less difficult, but not as efficient as point focus. The other classification methodology on the basis of receiver type consists of fixed receivers which are stationary devices that remain independent of the plant's focusing device (linear Fresnel collectors and central receiver systems) and Mobile receivers move together with the focusing device thus collecting more energy (parabolic-troughs and parabolic dishes)[1].

The parabolic trough collector (PTC) system is one of the proven CSP technologies in the medium temperature range (100-400^{0}C) due to its good optical efficiency and low initial cost. Parabolic trough collector system consists of parallel rows of large reflective parabolic troughs which focus solar energy on to a central receiver pipe (also called absorber pipe or heat collector element) placed at the focal line of the parabolic surface. The receiver is designed to absorb the solar energy concentrated on it. The receiver is made up of high conductivity steel tubing with a black coating surrounded with a protective glass cover, the space in between the protective glass cover and the steel tube is evacuated to reduce convection and radiation losses. The solar energy concentrated at the receiver tube is absorbed by a circulating heat transfer fluid (HTF).The HTF exchanges this heat to feed water in a heat exchanger (HE). After exchanging heat with the feed water, the HTF returns back to the solar field for heat pick up. The parabolic trough collector is usually aligned to north-south axis and tracks the sun to focus the solar radiation on to the receiver tube placed at the focal point of the trough system. The parabolic trough collector system can focus the solar radiation at 30 to 100 times of its normal intensity on to the central receiver tube located at the focal plane[2].The parabolic trough collector system for commercial power generation has been used initially in the Solar Electric Generating System (SEGS) plants I to IX. These plants have been brought to commercial operation between 1985 and 1991 and are located in the Mojave Desert of California in US[3].The LS-3 solar collector assembly used at Kramer Junction in Mojave Desert of California (Source: Sandia National Laboratory, U.S) has been shown as Fig.1 In the year 2007, Nevada Solar One plant has deployed the parabolic trough collector system for its 64 MWe capacity plant[3].The first commercial parabolic trough power plant of Europe, And asol I has been generating electricity since December, 2008 followed by And asol II in the middle of 2009[3]. These plants are located in the southern part of the Spain. The US and Spain have most of these projects followed by several countries in the Sun Belt. Several projects are under planning/construction stage in countries like China, Egypt, Algeria, Morocco, Australia etc. As on 15/05/2016, a total of 5464.67 MW of net power has been under various stages of planning, development, construction and operation based on parabolic trough collector technology [3].

The Fresnel mirror type of CSP system is very much similar to parabolic trough systems but as a replacement of using trough shaped mirrors that track the sun, long flat mirrors at various angles are used that have the effect of focusing sunlight on one or more pipes containing HTF which are mounted above the mirrors. The comparative plainness of this type of system makes this relatively cheap to manufacture but suffers from lower energy conversion efficiency relative to high optical efficiency of dish and trough systems. In a Fresnel solar collector, a number of discreet mirrors approximate a large parabolic trough collector. These mirrors (reflectors) are capable of concentrating the solar radiation on the receiver approximately 30 to 60 times its normal intensity[2]. The receiver is placed at the focal line of the collector system to absorb maximum amount of solar radiation. The receiver is usually a bank of black coated parallel tubes placed inside an insulated inverted trapezoidal stainless steel cavity. These tubes are capable enough to withstand high pressures and kept close to each other to absorb maximum concentrated solar radiation. The cavity aperture is covered with glass shield to allow concentrated solar radiation. This also minimises the heat losses due to convection and radiation. The cavity is insulated with thermal insulation and is encased in a metallic envelope to minimise the heat losses. The concentrated solar energy is further transferred to the heat transfer fluid like thermic fluid or water. This solar energy can be used either for feed water heating or steam generation as per the need[2]. The first prototype of Fresnel collector was developed by Solarmun do from Belgium. In 2004, an Australian company named Ausra (the then solar heat and power) built Fresnel collector system for Liddell power plant in Australia. In 2008, Ausra built the first Fresnel solar only power plant in Bakersfield/California.

This has been shown as Fig.2.The first commercial Fresnel power plant in Europe, PE 1, was built by Novatech Solar AG (the then Novatech Biosol). This plant commissioned in 2009 is situated in Spain and has a capacity of 1.4 MWe [3].

A solar power tower system, also known as a central receiver system generates high temperature heat from incident solar radiation by focussing concentrated solar energy on to a central receiver. The system uses large number of flat, trackable mirrors called heliostats to concentrate the solar radiation on to a tall tower located in the middle of heliostat field. The energy can be concentrated up to 1500 times the incident solar radiation [2]. The concentrated heat energy absorbed by the receiver is transmitted to a circulating HTF. The HTF can be liquid sodium, molten salts, air or water. The HTF heated in the receiver is used to generate steam, which can be sent in to a steam turbine for power generation. Usually molten salt is used as working fluid in solar power tower. The liquid molten salt is circulated through the receiver from cold tank and gets heated up in the receiver then passed to the hot storage tank. The hot molten salt is circulated through a heat exchanger to generate steam [2].The Crescent Dunes solar power tower plant at Nevada in US is operating since 2015. This plant uses molten salt as heat transfer fluid and has 10 hours of thermal storage [3].

A parabolic dish collector uses an array of parabolic dish shaped mirrors to focus solar energy on to a receiver located at the focal point of the dish. The two axis tracking system of the concentrator tracks the sun. HTF is circulated in the receiver to absorb the heat from the receiver. The concentration ratio of the parabolic dish collector is varies between 300 and 3000 [4].

A thermal energy system (TES) basically stores the solar energy collected during peak sunny hours for later use during non-solar hours. TES decouples solar energy availability with electricity generation. There are numerous criteria to evaluate TES systems and applications such as technical, environmental, economic, energetic, sizing, feasibility, integration and storage duration. Each of these criteria should be considered carefully to ensure successful implementation [5]. A TES designer should possess or obtain technical information on TES such as types of storage appropriate for the application, the amount of storage required, the effect of storage on system performance, reliability and cost and the storage systems or designs available [5].The technical properties of the storage materials are a very important aspect of technical design of any TES system. The material used for storage should have an excellent thermal energy storage capacity. This greatly reduces the system volume and foot print and improves system efficiency. A good rate of heat transfer between the TES material and heat transfer fluid (HTF) is highly essential to achieve shorter charge and discharge cycles. The storage material should have excellent chemical and mechanical stability for a large number of charge and discharge cycles [6].The environmental criteria should ensure that the basic design and operational practices that are used for the TES should not impair the public health or natural ecology and environment. Materials used should not be toxic or dangerous if released, could adversely affect the environment during the manufacture, distribution, installation or operation of the storage system [5].The cost of TES mainly consists of the cost of storage material, heat exchanger and land cost [6]. The economic justification for storage system normally requires that the annualised capital and operating costs for TES be less than those required for primary generating equipment supplying the same service loads and periods [5]. The evaluation of cost effectiveness of TES include hourly thermal loads for the peak day, the electrical load profile of the base case system against which TES is being compared and the size of the storage system and the control methods used [5]. Economic information that is needed includes electricity demand charges and time of use costs, the costs of the storage and financial incentives available [5]. Economic evaluation and comparison parameters often determined include the simple payback period [5]. Other methods are also used to compare the annualised investment cost of a TES with annual electricity cost savings [5].

Based on the energy storage mechanism, Thermal energy storage systems can be classified as sensible heat, latent heat and chemical storage systems. The energy storage density (kWh/m^3) increases from sensible heat storage to chemical storage with latent heat storage in between these two. As far as the development is concerned, the sensible heat storage systems are highly developed followed by latent heat storage systems. The chemical storage systems are yet to be developed. The popularity of sensible heat storage systems can be attributed to their low cost of large number of available storage materials. However, they suffer from lower energy storage density and hence occupy large space. The latent heat storage systems have relatively larger storage densities than the sensible heat storage systems with charging and discharging taking place at nearly isothermal conditions.

However, latent heat storage systems have poor heat transfer thereby increasing the charge/discharge cycle time. To tackle this issue, heat transfer enhancement techniques have to be incorporated so that the rate of heat transfer between the heat transfer fluid and storage material is maximised. Chemical storage systems have the highest energy storage capacities among all. Their energy densities are of the order of GJ compared to latent heat storage systems which are of the order of MJ per m^3. But the chemical storage systems suffer from poor long term reversibility of chemical reactions, complicated reactor vessel design and poor chemical stability[6].The state of the art storage materials for sensible heat storage are molten salts particularly Hitec/Hitec XL and solar salt [6][7]. These molten salts are widely used in several parabolic trough systems [6] [7]. Before the use of molten salts, Therminol VP-1,which is syntheticoil, has been used in several parabolic trough power plants. The maximum operating temperature for use of Therminol VP-1 is limited to 400^0 C. The solar salt is relatively cheaper and has a maximum operating temperature of 585^0 C but its high melting temperature of 220^0 C necessitates the use of costly anti freezing agents **[6] [7].**For latent heat storage, inorganic salts/salt eutectics and metals/metal alloys are the potential materials in view of higher operating temperatures required for CSP plants. The main disadvantage with most of the phase change materials (PCM) is their low thermal conductivity, which makes it necessary to adopt heat transfer enhancement techniques **[6] [7].**The insertion of high conductivity materials like carbon cloth, brush etc. into the PCM, improves the heat transfer rate of composite material significantly **[6].**

III. Solar Aided Coal Fired Power Plants

The solar energy can be successfully used in an existing/new coal fired thermal power plants for generation of steam or for preheating the feed water. This helps in increasing the cycle efficiency as this will either increase the steam turbine output (power boosting) for same fuel consumption or reduce the coal consumption (fuel saving) for the same turbine output depending on the plant operating mode. Either way, this is environment friendly as this will reduce the emission foot print. Coal fired power plants utilising solar energy in this way either for additional steam generation or feed water preheating are popularly known as solar aided coal fired power plants. Both cases will be discussed in detail in subsequent sections.

a) Additional Steam Generation

Using solar energy, steam can be generated in a solar boiler by either direct steam generation (DSG) or through a heat transfer fluid. The generated steam is used in an existing coal fired power plant for either additional power generation or for reducing fuel consumption. Usually, when solar energy is integrated with an existing coal fired power plant, the fuel saving mode of operation is preferred as it does not need resizing of turbo generator.

The Liddell power station at New South Wales, Australia uses a 9 MW_{th} solar boiler which feeds steaminto an existing 2000 MW coal fired power station. The solar field uses linear Fresnel reflector technology for solar energy capture. NREL has reported that the replacement of coal by the solar boiler will cut greenhouse gas emissions by approximately 5,000 tonnes per annum**[3].**Kogan Creek Solar Boost project at Queensland region of Australia is set to become the largest solar integration with a coal-fired power station in the world. The project consists of a compact linear Fresnel reflector solar thermal augmentation of the existing Kogan Creek Power Station, increasing the power station's electrical output and fuel efficiency. The solar addition of 44 MW will enable the 750 MWcoal fired power station, already one of Australia's most efficient coal-fired power stations and Australia's largest single unit, to produce more electricity with the same amount of coal. The project will help avoid 35,600 tonnes of carbon dioxide per year annually**[3].**The Colorado Integrated Solar Project (Cameo) was a hybrid CSP/coal plant approach using parabolic-trough solar technology.

A parabolic trough solar field provided thermal energy to produce supplemental steam for power generation at Xcel Energy's Cameo Station's Unit 2 (approximately 2 MWe equivalent) in order to decrease the overall consumption of coal, reduce emissions from the plant, improve plant efficiency, and test the commercial viability of concentrating solar integration. The plant was used for testing purposes until the coal plant was retired and the CSP plant was decommissioned**[3].**

b) Solar Aided Feed Water Heating (Safwh)

Integration of solar energy with conventional power plants can be explored as a viable option for achieving cleaner and cheaper power generation. The steam generator (Boiler) of a conventional power plant generates steam at a high pressure and temperature. This high pressure steam is allowed to expand in high, intermediate and low pressure sections of a condensing steam turbine to generate power. The condensate collected in the hot well is pumped through various low and high pressure feed water heaters before it reaches the economiser. Bleed steam taken from various stages of different turbine sections are used for preheating of the feed water. The final feed water temperature is increased to match with boiler design steam parameters and economic cycle design considerations. This regenerative feed water heating is basically aimed at increasing the cycle efficiency. The number of feed water heaters and their steam extraction points depends upon the techno-economic considerations of cycle design optimisation.

The feed water cycle consists of two series of heaters. They are Low Pressure (LP) heaters and High Pressure (HP) heaters. The LP heater series consists of up to four low pressure heaters supplied with bled steam from low and intermediate pressure turbine sections. After passing through this heater group, condensate enters an open feed water heater known as deaerator where deaeration of feed water occurs. The deaerated feed water then enters the next series of HP heater group consisting of up to three feed water heaters taking bleed steam from high and intermediate pressure turbine sections.

In SAFWH, solar thermal energy at various temperature ranges is used to replace the bleed steam coming from various turbine extractions either partially or fully to preheat the condensate/feed water in the feed water heaters (FWH). This solar energy used for feed water preheating can be used either for saving the bled steam or for minimising the fuel consumption. When the saved bled steam is allowed to expand in the turbine, extra power can be generated, and is known as power boost mode. If the turbine power output is maintained constant, the fuel consumption reduces with solar aided feed water heating and this is known as fuel saving mode.

The thermodynamic advantages of using solar energy in the regenerative Rankine based power plant cycle have been found to be better than the solar stand-alone power generation **[8].**The Exergy Merit Index (EMI) (the ratio of the work generated by the saved steam to the exergy supplied by the solar heat) of solar aided systems can be greater than 100% while maximum efficiency of stand-alone solar thermal power plants never reach 100% **[8].**It has been observed that by the substitution of turbine bleed stream to high pressure feed water heaters alone with SAFWH results in about 5–6% instantaneous improvement in coal consumption and additional power generation for the fuel conservation and power boosting modes in comparison to reference power plants **[9].** Hu et al [2010] have demonstrated energy and exergy advantages of solar aided power generation by carrying a case study on a 500 MW power plant of Loy Yang power station located in Latrobe valley, Victoria, Australia using THERMOSOLV software. With 100% replacement of bleed steam for all closed feed water heaters, the power output was 572.5 MW in power boosting mode and with cycle efficiency increase by 6.65%**[10].** Yang et al. [2011] have demonstrated through a case study that solar aided power generation (SAPG) is an efficient way to utilise solar energy in the low and medium temperature range for power generation by replacing bleed steam with solar energy in feed water heaters. Four schemes were suggested to replace bleed steam of the feed water heaters. In the first scheme, bleed steam of first HP feed water heater was replaced with solar energy at 260^{0}C. In the second scheme, bleed steam of second HP feed water heater was replaced with solar energy at 200^{0}C. In the third scheme, bleed steam of all LP feed water heater was replaced with solar energy at 160^{0}C. In the fourth scheme, bleed steam of last LP heater with solar energy up to 100^{0}C**[11].** Dimityr Popov [2011] has modelled Rankine regenerative steam cycled power plant with Thermo flow software. The plant model incorporated a field with solar Fresnel collectors that directly heats boiler's feed water. The proposed plant modification was yielded substantial fossil fuel input reduction. The best results were obtained when the group of high pressure heaters is replaced and feed water temperature exceeds its original design case, having efficiency higher than 39%for the best solar hour of the year**[12].** Yan et al. [2011] analysed the performance of fossil fuel fired power plants with different MW outputs, subcritical, supercritical and ultra-supercritical plants with integration of solar energy at different temperature levels. They observed that at high temperature integration levels, better benefits could be obtained in terms of solar to power efficiency, fuel savings. They found that subcritical and supercritical plants are better options in comparison to ultra-supercritical plants with solar integration**[13].** Zekiyilmazoglu et al. [2012] carried out a case study on solar repowering of Soma thermal power plant of 22MWe located in Turkey**[14].** Bakos et al. [2013] have simulated the operation of the 300 MW lignite fired power plant of Ptolemaist integrated with a solar field of parabolic trough collectors using TRNSYS software in both power boosting and fuel saving modes. The power plant performance, power output variation, fuel consumption and CO_2 emissions were calculated. Furthermore, an economic analysis was carried out for both power boosting and fuel saving modes of operation and optimum solar contribution was estimated**[15]**. Warrick et al. [2013] have compared solar aided power generation (SAPG) and stand-alone concentrating solar power (CSP) for a South African Plant. They found that the annual electricity generated from solar thermal at the SAPG plant is more than 25%greater than the stand-alone CSP plant. They have observed that if the cost of SAPG is taken as 72% of the cost of a stand-alone CSP, SAPG is 1.8 times more cost effective than the stand-alone CSP option**[16].** Jamel et al. [2013] have presented a review paper on advances in the integration of solar thermal energy with conventional and non-conventional power plants**[17].** Peng et al. [2014] have investigated solar aided feed water heating in a 330 MWe coal-fired power plant in Sinkiang Province of China. They have demonstrated the advantages of the solar aided coal-fired power plant under off-design conditions**[18].** Peng et al [2014] have also performed Exergy evaluation of a typical 330 MW solar-hybrid coal-fired power plant in China**[19].**

Boukelia et al. (2015) have performed 4E comparative study of 8 different configurations of parabolic trough solar thermal power plants with two different working fluids (Therminol VP-1 -oil and molten solar salt), with and without integrated thermal energy storage or/and backup fuel system. Their results have indicated that the configurations based on molten salt are better in terms of environmental and economic parameters**[20].** Hou et al. (2015) have done performance analysis of a solar aided plant in fuel saving mode**[21].**

c) Exergy Analysis Of Integrated Solar Aided Coal Fired Power Plant

The first step before performing an exergy analysis on an integrated solar aided coal fired plant is developing a conceptual integrated cycle. This involves

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collection of reference power plant heat and mass balance data, choice of feed water heater (s) for solar aided feed water heating, choice of direct/indirect heat transfer method for transferring the solar heat to feed water, arrangement of feed water heat exchanger and choice of heat transfer fluid (in case of indirect heat transfer), choice of solar collector system (depends on temperature of feed water entering and leaving the feed water heat exchanger) etc. among several others. Then site specific hourly average values of direct normal irradiance (DNI) have to be collected for simulation of solar field. Simulation of solar field yields the heat loss coefficient, receiver temperature and pressure loss in the HTF circuit etc. Then the thermodynamic properties of integrated cycle at salient points can be found by applying mass and energy balance to all cycle components.

The exergy rate balance for a steady flow process of an open system is given by

$$\sum_j \left(1-\frac{T_0}{T_j}\right)\dot{Q}_j + \sum_i \left(\dot{m}_i \psi_i\right) = \dot{W}_{CV} + \sum_e \left(\dot{m}_e \psi_e\right) + \dot{E}_D \qquad (1)$$

In this equation, $\dot{E}_D$ is the rate of exergy destruction (Irreversibility) associated with the process. The irreversibility (Van Wylen et al, 1994) is also given by Gouy-Stodola theorem as

$$\dot{E}_D = T_0 \dot{S}_{gen} \qquad (2)$$

Here $\dot{S}_{gen}$ is the entropy generation associated with the process.

For a complex thermal system, the exergy analysis can be performed by analysing the components of the system individually.

The exergetic efficiency of a thermal system or system component is defined as the ratio of exergetic output to exergetic input.

$$\eta_{II} = \frac{\psi_O}{\psi_i} \qquad (3)$$

Here the subscripts i and o refer to input and output respectively.

The exergy analysis of important components of the integrated solar aided coal fired power plant is given below:

Boiler

For the sake of exergy analysis, the boiler has been divided into combustion and heat transfer zones. The exergy balance for the combustion zone is given as:

$$\dot{m}_f \psi_f + \dot{m}_{air} \psi_b = \dot{m}_p \psi_p + \dot{E}_{D,C} \qquad (4)$$

Where $\dot{m}_f$, $\dot{m}_{air}$ and $\dot{m}_p$ are the mass flow rates of fuel, air and the products of combustion respectively. $\dot{E}_{D,C}$, is the rate of exergy destruction in the combustion zone of the boiler.

Exergy of the coal and flue gasses have been calculated as explained in Kotas (1984) **[22].**

The Exergetic efficiency of combustion zone is defined as:

$$\eta_{II,C} = 1 - \frac{\dot{E}_{D,C}}{\dot{m}_f \psi_f + \dot{m}_{air} \psi_b} = 1 - \frac{T_0 \dot{S}_{gen,C}}{\dot{m}_f \psi_f + \dot{m}_{air} \psi_b} \qquad (5)$$

Where, $\dot{S}_{gen,C}$ is the associated entropy generation in the combustion zone of the boiler.

The exergy flow equation for the high temperature heat transfer zone becomes:

$$0 = \dot{m}_p\left(\psi_p - \psi_0\right) - \dot{m}_{fwi}\left(\psi_1 - \psi_{fwi}\right) - \dot{m}_{crh}\left(\psi_{hrh} - \psi_{crh}\right) - \dot{m}_{air}\left(\psi_b - \psi_0\right) - T_0 \dot{S}_{gen,HT} \qquad (6)$$

Where, the subscripts 1, fwi, crh and hrh refer to final super heater outlet, feed water inlet, cold reheat and hot reheat respectively.

The Exergetic efficiency of heat transfer zone is defined as:

$$\eta_{II,HT} = 1 - \frac{\dot{E}_{D,HT}}{\dot{m}_p(\psi_p - \psi_0)} = 1 - \frac{T_0 \dot{S}_{gen,HT}}{\dot{m}_p(\psi_p - \psi_0)} \qquad (7)$$

Where, $\dot{E}_{D,HT}$ and $\dot{S}_{gen,HT}$ are rate of exergy destruction and entropy generation in the heat transfer zone of the boiler respectively.

Steam Turbine

The exergy balance for a simple steam turbine section is given as

$$\dot{m}_1\psi_1 = \dot{m}_2\psi_2 + \dot{W}_T + \dot{E}_{D,T} \tag{8}$$

Where, the subscripts 1 and 2 refer to turbine inlet and exit conditions respectively. $\dot{W}_T$ and $\dot{E}_{D,T}$ are Turbine work output and rate of exergy destruction in the turbine section respectively.

The mass balance is given by

$$\dot{m}_1 = \dot{m}_2 \tag{9}$$

The exergetic (second law) efficiency of steam turbine is given by

$$\eta_{II,T} = \frac{\dot{W}_T}{\dot{m}_1(\psi_1 - \psi_2)} \tag{10}$$

Condenser

The exergy balance for condenser section is given as

$$\dot{m}_1\psi_1 = \dot{m}_2\psi_2 + \left(1 - \frac{T_0}{T_k}\right)\dot{Q}_{cond} + \dot{E}_{D,cond} \tag{11}$$

Where, subscripts 0,1, and 2 refer to ambient, condenser inlet and exit conditions respectively. T_k and $\dot{Q}_{cond}$ are temperature of heat rejection and rate of heat transfer from condenser respectively.
The mass balance is given by

$$\dot{m}_1 = \dot{m}_2 \tag{12}$$

The exergetic (second law) efficiency of the condenser is given by

$$\eta_{II,cond} = 1 - \frac{\dot{E}_{D,cond}}{\dot{m}_1(\psi_1 - \psi_2)} \tag{13}$$

Pump

The exergy balance for a pump can be expressed as

$$\dot{m}_1\psi_1 + \dot{W}_{pump} = \dot{m}_2\psi_2 + \dot{E}_{D,pump} \tag{14}$$

Where, the subscripts 1 and 2 refer to pump inlet and exit conditions respectively. $\dot{W}_{pump}$ and $\dot{E}_{D,pump}$ are pump work input and rate of exergy destruction in the pump respectively.
The mass balance is given by

$$\dot{m}_1 = \dot{m}_2 \tag{15}$$

The exergetic (second law) efficiency of the pump is given by

$$\eta_{II,pump} = \frac{\dot{m}_2(\psi_2 - \psi_1)}{\dot{W}_{pump}} \tag{16}$$

Feed Water Heater

The exergy balance for a feed water heater can be expressed as

$$\dot{m}_1\psi_1 + \dot{m}_s\psi_s = \dot{m}_2\psi_2 + \dot{m}_d\psi_d + \dot{E}_{D,fwh} \tag{17}$$

Where, the subscripts 1, 2, s and d refer to heater feed water inlet, exit, steam and drip respectively.
The mass balance is given by

$$\dot{m}_1 = \dot{m}_2 \tag{18}$$

$$\dot{m}_s = \dot{m}_d \tag{19}$$

The exergetic efficiency of feed water heater can be expressed as

$$\eta_{II,fwh} = 1 - \frac{\dot{E}_{D,fwh}}{\dot{m}_s(\psi_s - \psi_d)} \tag{20}$$

Solar Field

The exergetic solar power input to parabolic trough is given by

$$Ex_I = \dot{Q}_I * \left[1 - \frac{4}{3}\left(\frac{T_0}{T_s}\right) + \frac{1}{3}\left(\frac{T_0}{T_s}\right)^4\right] \tag{21}$$

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Where $T_s =$ 5600 K is apparent black body temperature of sun and T_0 is the ambient temperature and $\dot{Q}_I$ is the solar power incident on the mirror surface.

The exergetic solar power absorbed by the receiver is given as

$$Ex_a = \dot{Q}_a\left[1 - \frac{T_0}{T_r}\right] \quad (22)$$

Where T_r is the receiver temperature (K)

The useful exergetic gain by the heat transfer fluid for a segmental length is given as

$$Ex_u = \dot{m}_f(\psi_e - \psi_i) = \dot{m}_f[(h_e - h_i) - T_0(s_e - s_i)] \quad [23]$$

d) Economic Analysis Of Integrated Solar Aided Coal Fired Power Plant

Economic analysis is a very important step in the feasibility study of any power project. This involves finding the levelised cost of electricity (LCoE) for the life of the project, net present value and payback periods.

The economic analysis is very useful in managerial decision making and in the determination of worthiness of the chosen project.

The economic analysis involves estimation of capital costs, fuel costs, operation and maintenance (O&M) costs and other expenses. The first step in performing the economic analysis is estimation of capital costs of different plant equipment. The best way is to consider the actual cost data if it is available. In the absence of actual capital cost data, capital cost functions for different plant equipment may be considered for estimation of capital costs. The obtained costs have to be brought to the same reference year for which economic analysis has to be done by multiplying with cost index (CI). This capital cost for the reference year can be obtained by Cost for reference year = Purchase cost*CI for reference year/CI for year of purchase [24] Another method is to assume a certain percentage of escalation in costs every year from the year of purchase to reference year. The total capital costs are further classified as direct capital costs (DCC) and indirect capital costs (IDCC). Direct capital costs comprise the capital costs of power block, solar field including thermal storage system, land and site preparation. The cost of power block includes the costs of all equipments in the conventional plant, installation, piping, instrumentation and controls and electricals. The cost of solar field includes the cost of mirrors, support structure, foundation, absorber tubes, swivel joints, hydraulic and electrical drives, heat transfer fluid (HTF), HTF system, Electronic controls and electricals (ECE) and thermal storage system. The indirect capital costs comprise the Engineering, procurement and construction (EPC) costs, pre-operative expenses and interest during construction. Once total capital costs are known fixed capital costs per unit (USD/kWh) can be calculated by dividing the total capital costs (USD per kW) by net annual energy generated(kWh/kW). Fixed O&M cost per unit (USD/kWh) has been obtained by dividing the fixed O&M cost per kW by net annual energy generated. Total variable cost per unit (USD/kWh) can be obtained by adding variable O&M cost per unit and fuel cost per unit.

Three economic indicators are annualised cost of electricity generation (ACoE), Levelised cost of electricity generation (LCoE) and simple payback period. The ACoE can be obtained by adding fixed capital cost per unit; fixed O&M cost per unit and total variable cost per unit. To levelise the fuel, O&M-fixed and variable cost for the life of the plant, a levelizing factor has to be taken into account. LCoE can be obtained by adding fixed capital cost per unit and levelised fuel, fixed and variable O&M costs per unit. Simple payback period can be calculated by dividing total capital cost by net annual benefit. Sensitivity analysis can be performed to find out the variation of LCoE with discount rate, plant capacity factor and fuel cost.

e) Thermo economic Analysis Of Integrated Solar Aided Coal Fired Power Plant

Thermo economic analysis is a very useful technique, which is finding increasing application in the area of thermal systems design. This involves the integration of the exergy and economic principles in achieving the objective(s) of cost calculation of products generated by different devices in a large thermal system and/or optimising specific decision variables in minimising the cost **[23].**

Zhang et al. (2006) have done exergy cost analysis on a 300 MW pulverised coal fired power plant based on structural theory of thermo economics. Based on Fuel-Product concept, they have developed a productive structure of the reference power plant for carrying out thermo economic analysis**[24].** M. Ameri et al. (2008) have carried out energy, exergy and exergo economic analysis on a 250 MW gas fired steam power plant in Iran. They have calculated the exergy destruction in all major components of the power plant and concluded that the rate of exergy destruction in the boiler is higher than the rate of exergy destruction of other components. They have carried out exergo economic analysis and found that the boiler has the highest cost of exergy destruction. They have developed a thermo economic optimisation model and found that the cost of exergy destruction and purchase cost can be considerably reduced by adjusting the extraction steam mass flow rate of and pressure of feed water heaters **[25].** L. Wang et al. (2012) have performed an exergoeconomic analysis on a coal fired ultra-super critical thermal power plant existing in China with an objective to understand the cost formation process and to evaluate economic performance of all components of the plant using SPECO (specific exergy costing) method **[26].** J.Uche et al. (2000) have carried out thermo economic optimisation of a steam power plant coupled with a multi stage flash desalination unit. They have developed a physical and a thermo economic model of the plant in carrying out the thermo economic optimisation **[27].** Amin M Elsafi (2015) has performed an exergy and exergoeconomic analysis on sustainable direct steam generation solar power plants. For each component of the plant, exergy and exergy-costing balance equations have been formulated based on fuel-product concept **[28].** Zhai et al. (2016) have analysed a solar-aided coal-fired power generation system based on thermo-economic structural theory.

They have applied thermo economic structural theory on a solar aided thermal power plant compared the performance in both fuel saving and power boosting mode. They have observed that the coal consumption rate has reduced by 15.04 g/kWh in fuel-saving mode.

The power output is 57.2 MW higher in power-boosting mode. They have found that thermo economic cost of electricity has been increased due to large investment in solar field**[29].**

For carrying out the thermo economic analysis on an integrated solar aided thermal power plant, a physical model has been developed by aggregating and disaggregating certain plant equipment. The turbine sections can be disaggregated into several units. The solar field and thermal energy storage system can be aggregated into one single unit.

Further, a productive structure has to be developed based on the fuel-product approach. In the fuel product approach, fuel to specific plant equipment means the different resources consumed by that plant equipment in delivering a product. The resources (fuel) for different plant equipment are exergetic flow (FB), Negentropic flow (FN)/electricity (FW) and capital cost of the component (FZ). In a productive structure diagram, plant equipment is represented by a rectangle. Incoming arrows to particular equipment represent the resources consumed and outgoing arrows from that equipment represent the product generated by it. Bifurcations are represented by circles and junctions are represented by rhombuses. Junctions are basically distributing resources among different plant equipment.

The capital cost of kth plant component per unit time can be obtained from the equation

$$F\dot{Z}_k = FZ_k * CRF * \varphi / N \qquad [25]$$

Where, FZ_k is the capital cost of the plant equipment, CRF is the capital recovery factor, φ is the maintenance factor (≈ 1.06) and N is the plant annual operating hours respectively.

Capital recovery factor (CRF) can be calculated from the equation

$$CRF = \frac{i * (1+i)^n}{\left((1+i)^n - 1\right)} \qquad [26]$$

Where, "I" is the interest rate (taken as 10%) and "n" is the plant life in years (≈25) respectively. A set of linear equations can be formulated based on the productive structure for each plant equipment, junctions and bifurcations. These linear equations can be solved to find out the costs of all major flow streams in the reference plant.

IV. Solar Aided Combined Cycle Power Plants

The integration of solar thermal energy with conventional gas fired combined cycle power plants (CCPP) has gained wide acceptance among the countries with high solar potential like U.S, Spain, Egypt, Morocco, Algeria, Iran and Mexico**[3].** These plants are popularly known as integrated solar combined cycle (ISCC) power plants. The Kuraymat ISCC plant, 100 kM south of Cairo in Egypt comprises two gas turbines of 40 MWe each and one steam turbine of 70 MWe with a parabolic trough solar field capable to generate 200 GWh per annum**[30].**The plant generates 20 MWe of solar based electrical power**[3] [30].** The plant integrates solar energy in the bottoming (steam) cycle by heating the feed water leaving the preheater and sending it to super heater located in the heat recovery steam generator. The plant uses Therminol VP-1 as the heat transfer fluid. The solar field was provided by Flag sol GmbH. The Archimede concentrating solar power project operating in Sicily, Italy is a parabolic trough plant which produces steam (4.72 MWe equivalent) sent to a combined-cycle steam turbine rated at 130 MW. The parabolic trough system of this plant is the first one to use the molten salt as heat transfer fluid. This plant has 08 hours of thermal storage **[3].** The Agua Prieta II ISCC in Mexico is an under construction (Status date: 30 October, 2013) ISCC plant where, the fossil fuel is partially replaced with solar energy [3]. This plant has an overall capacity of 478 MW comprising of 464 MW with combined cycle and 14 MW with solar. Duct burners are provided to produce 14 MWe output when solar is not in operation. ISCC Ain Beni Mathar is an ISCC plant operating in Morocco which has a combined plant output of 470 MWe which includes a solar based power output of 20 MWe **[3].**This plant like Agua Prieta II

ISCC, partially replaces fossil fuel with solar energy. However, this plant has no thermal storage system unlike Agua Prieta II ISCC [3].ISCC Duba I is an under construction project having a capacity of 43 MWe expected to start production in year 2017 [3].

a) Solar Integration In The Brayton Cycle

The solar thermal energy can be integrated with a conventional CCPP either in topping cycle or bottoming cycle or both. The topping cycle solar integration can be gas turbine (GT) inlet air cooling using solar operated vapor absorption chiller [31] or heating the gas turbine compressor discharge air [32].

i. Solar Operated Vapour Absorption Chiller For Gas Turbine (Gt) Inlet Air Cooling

Dimityr Popov, 2014[31] has proved that inlet air cooling of gas turbines in a combined cycle configuration using solar assisted vapour absorption chiller has lower specific incremental capital costs and requires smaller land area than other options. The options considered in his study are an integrated solar combined cycle with medium temperature integration, inlet air cooling using a mechanical chiller and a vapour absorption chiller. He has clearly indicated that ISCCPP's suffer from following drawbacks:

- Low power output during cloud cover and night time. This causes part load operation of steam turbine and hence higher heat rate and higher cost of electricity generation. This needs thermal energy storage.
- Cycle efficiency suffers as the solar heat is added at low temperatures. Hence the solar contribution of ISCCPP's even at best locations where solar conditions are excellent is between 2 to 6%.
- Not possible for existing combined cycle plants as the steam turbine size needs to be increased.

On contrary to the ISCCPP's, the solar operated vapour absorption chiller offers several advantages. The power output of a gas turbine reduces drastically with increase in ambient temperature. Usually hot summer season is the peak demand season so power output reduction from gas turbines is definitely a cause of concern for power generators. So in general, most of the gas turbine plants resort to various methods of inlet air cooling. Inlet air cooling can be achieved by either evaporative cooling techniques or by using chillers. The evaporative cooling methods involve using wetted media, Inlet fogging etc. [31].The temperature of cooled air in evaporative cooling is always higher than the ambient wet bulb temperature. However, with chilling techniques, temperatures well below the wet bulb temperatures can be achieved. These chillers can be mechanical chillers or vapour absorption chillers (VAC). Mechanical chillers consume electrical energy for running of refrigerant compressor. However vapour absorption chiller systems do not use significant electrical energy as they need a low temperature heat source (hot water or steam) for their operation [31]. As mentioned by Dimityr Popov, (2014)[31], there is an excellent match between gas turbine power output reduction during hot weather conditions and abundance of solar energy for steam generation during the same period for gas turbine inlet air cooling. Said et al. (2015) [33] have performed design and analysis of a solar powered absorption refrigeration system modified to increase its COP using refrigerant storage. They have observed an increase of 8% in COP over the conventional design by using the refrigerant storage. Kaynakli et al. (2015)[34] have performed energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources. Bakos et al, (2013) [35]have carried out techno economic assessment of an integrated solar combined cycle power plant in Greece using line-focus parabolic trough collectors using Transys software. Baghernejad et al. (2010) have carried an exergy analysis of an integrated solar combined cycle system and found that maximum exergy destruction (29.62%) occurs in the combustor of gas turbine [36].

ii. Solar Heating Of Gt Compressor Discharge Air

Another application of solar energy in Brayton cycle is heating the compressor discharge air. The compressed air from the gas turbine compressor is allowed in to a pressurised receiver placed on a central tower. Heliostat field is made to focus on to the pressurised receiver for heating the air. This will increase the temperature of air entering the gas turbine combustor [32]. This kind of solar integration in the topping (Brayton) cycle of a combined cycle power plant greatly reduces the fuel consumption without affecting the gas turbine output.

b) Solar Integration In The Bottoming Cycle

In the bottoming cycle solar integration, there are three integration levels based on the fluid temperature capability [37]. They are referred to as high/medium and low temperature integration technologies. In high temperature integration, solar tower systems can be used to generate super-heated steam at temperatures up to 545^0 C [37]. This steam is allowed to mix with the superheated steam generated in heat recovery steam generator (HRSG) before admitting to high pressure turbine. Reheating is also possible in the solar [37]. Ugolini et al, (2009) have mentioned that for the medium temperature solar integration technologies generating steam up to around 395^0 C, it is best to generate dry saturated steam at high pressure and mix with the steam coming from HRSG HP drum. In the low temperature integration, low pressure steam is generated using linear Fresnel collectors which can be sent to cold reheat line or in to the LP admission line [37].

i. *Low Temperature Solar Integration*

Low temperature solar integration technologies involve fluid temperatures between 250^0 C and 300^0 C. However, the actual temperature of integration depends on the type of gas turbine plant with which integration has to be done. Linear Fresnel reflectors are most widely used for low temperature solar integration with combined cycle power plants. As described by Ugolini et al. (2009) **[37]**, it is possible to generate steam at two different pressure levels for integration with steam cycle.

The first one is to generate dry saturated steam at 30 bar pressure and admit it to cold reheat line. The second one is to generate dry saturated steam at 5 bar pressure and admit it to low pressure (LP) steam admission line **[37]**.The feed water take off temperature must be below the saturation temperature corresponding to the pressure of the steam generated **[37]**.

ii. *Medium Temperature Solar Integration*

Medium temperature solar integration with combined cycle power plants has been considered as a proven technology. The best practice for medium temperature solar integration is to maximise feed water heating (sensible heat addition) in the heat recovery steam generator (HRSG) so that only latent heat is added in the solar field **[37]**.This will not only reduces the solar field size but also maximises the heat recovery from exhaust gasses in the HRSG. In view of this, it is always better to take the feed water for solar heating from the last high pressure economiser exit. This maximises the solar conversion efficiency **[37]**.Once sensible heat addition to HP feed water is finished in the HRSG, latent heat addition should be accomplished in the solar field. A separator vessel is usually provided at the outlet of solar field, so that any moisture can be removed before it is mixed with the steam leaving the HP drum. Alternatively, the wet steam leaving the solar field may be sent to HP drum for water separation from steam. The parabolic trough collectors are widely used for medium temperature solar integration. The trough collectors have the advantage of maturity in technological development and higher optical efficiencies in comparison to linear Fresnel collectors.

iii. *High Temperature Solar Integration*

The high temperature solar integration involves generating superheated steam at temperatures up to 545^0 C and allowing this steam to mix with the superheated steam leaving the HP super heater before it is allowed to expand in the high pressure turbine **[37]**.A heliostat field focuses the collected solar radiation on to a receiver placed on top of a tower. A heat transfer fluid collects the solar energy from the receiver and exchanges in turn with the feed water in a solar boiler.

The cold reheat steam leaving the high pressure turbine can be sent back to solar boiler for reheating **[37]**.The high temperature solar technology has got minimum integration issues as this has minimum impact on the HRSG **[37]**.

c) *Exergy Analysis Of Direct Steam Generation Solar Aided Cc Plant*

The exergy analysis of direct steam generation solar aided CC plant involves the exergy analysis of individual plant equipment of the integrated plant.

Compressor (C)

The mass and energy balance for air compressor is

$$\dot{m}_1 = \dot{m}_2 \qquad [27]$$

The exergy balance for the compressor is

$$\dot{m}_1\psi_1 + \dot{m}_1\psi_{1v} + \dot{W}_c = \dot{m}_2\psi_2 + \dot{m}_2\psi_{2v} + T_0\dot{s}_{gen,C} \qquad [28]$$

Here, $\dot{m}$, h, w and $\dot{W}_c$ are mass flow rate of air, enthalpy of the air, the specific humidity of air and rate of compressor respectively. The subscripts 1, 2 and v refer to compressor inlet, discharge and water vapour respectively. $\dot{S}_{gen,C}$, is the rate of entropy generation in the compressor.

Exergy destruction (irreversibility) in the compressor is

$$\dot{E}_{D,C} = T_0\dot{s}_{gen,C} = \dot{m}_1 T_0(s_2 - s_1) \qquad [29]$$

The exergy efficiency of the compressor is

$$\eta_{II,c} = \frac{\dot{m}_1(\psi_2 - \psi_1)}{\dot{W}_c} \qquad [30]$$

Combustion Chamber (CC)

The mass and energy balance for the combustion chamber is

$$\dot{m}_2(1+w) + \dot{m}_f = \dot{m}_3 \qquad [31]$$

Where $\dot{m}_2$, $\dot{m}_f$ and $\dot{m}_3$ are the mass flow rates of air, fuel and combustion products and h_2, h_{2v} and h_3 are the enthalpy of air entering the combustion chamber, enthalpy of water vapour entering the combustion chamber and enthalpy of combustion products leaving the combustion chamber respectively.

The subscripts 2 and 3 refer to the inlet and exit of the combustion chamber.

The energy efficiency of the combustion chamber is The exergy balance for the combustion chamber is

$$\dot{m}_2\psi_2 + w^*\dot{m}_2\psi_{2v} + \dot{m}_f\psi_f = \dot{m}_3\psi_3 + \dot{E}_{D,CC} \qquad [32]$$

Where, ψ_f and $\dot{E}_{D,CC}$ are standard chemical exergy of the fuel and rate of exergy destruction in the combustion chamber respectively.

The exergy efficiency of the combustion chamber is

$$\eta_{II,cc} = \frac{(\dot{m}\psi)_g}{(\dot{m}\psi)_{a+f}} = \frac{\dot{m}_3\psi_3}{\dot{m}_2\psi_2 + w^*\dot{m}_2\psi_{2v} + \dot{m}_f\psi_f} \qquad [33]$$

GAS Turbin (GT)

The mass and energy balance for gas turbine is

$$\dot{m}_3 = \dot{m}_4 \qquad [34]$$

Here, $\dot{m}$, h and $\dot{W}_{gt}$ are mass flow rate of combustion gasses, enthalpy of the combustion gasses, rate of gas turbine work and energy loss in the gas turbine respectively. The subscripts 3 and 4 refer to the gas turbine inlet and outlet.

The exergy balance for the gas turbine is

$$\dot{m}_3\psi_3 = \dot{m}_4\psi_4 + \dot{W}_{gt} + T_0\dot{s}_{gen,GT} \qquad [35]$$

Exergy destruction (irreversibility) in the gas turbine is

$$\dot{E}_{D,GT} = T_0\dot{s}_{gen,GT} = \dot{m}_3T_0(s_4 - s_3) \qquad [36]$$

The exergy efficiency of the gas turbine is

$$\eta_{II,gt} = \frac{\dot{W}_{gt}}{\dot{m}_3(\psi_3 - \psi_4)} \qquad [37]$$

Heat ReCovery Steam Generator (HRSG)

The mass balance for the heat recovery steam generator (HRSG) is

$$\dot{m}_{fgi} = \dot{m}_{fgo} \qquad [38]$$

$$0 = \dot{m}_{fgi}(\psi_{fgi} - \psi_{fgo}) - 2^*\dot{m}_{hpei}(\psi_{hpeo} - \psi_{hpei}) - 2^*\dot{m}_{hpevo}(\psi_{hpsho} - \psi_{hpevo}) - 2^*\dot{m}_{lpei}\left(\psi_{lpeo} - \psi_{lpei}\right) - \dot{m}_{cphi}(\psi_{cpho} - \psi_{cphi}) - T_0 s_{gen}, hrsg \qquad [39]$$

The exergy efficiency of HRSG is

$$\eta_{II,hrsg} = \frac{2^*\dot{m}_{hpei}(\psi_{hpeo} - \psi_{hpei}) + 2^*\dot{m}_{hpevo}(\psi_{hpsho} - \psi_{hpevo}) + 2^*\dot{m}_{lpei}\left(\psi_{lpeo} - \psi_{lpei}\right) + \dot{m}_{cphi}(\psi_{cpho} - \psi_{cphi})}{\dot{m}_{fgi}(\psi_{fgi} - \psi_{fgo})} \qquad [40]$$

Where, the subscripts fg, hpe, hpev, lpe and cph refer to flue gas, high pressure evaporator, high pressure evaporator, low pressure economiser and condenser preheater respectively. I and o refer to inlet and outlet respectively.

The exergy analysis of Steam plant equipment has been already explained in section 3.3.

d) Economic Analysis Of Combined Cycle Power Plant

The base cost estimate fora conventional natural gas fired combined cycle facility can be obtained from either actual plant capital cost data or Updated Capital Cost Estimates for Utility Scale Electricity Generating Plants obtained from Independent Statistics & Analysis, U. S. Energy Administration & Information**[38].**This includes the cost of civil structural material and installation, mechanical equipment supply and installation, electrical/instrumentation & control, project indirects which include engineering, distributable costs, scaffolding, construction management & start up, EPC cost, fee & contingency and owner costs. Land cost can be obtained by multiplying the cost of land per acre with the land area of CCPP. The fixed operation and maintenance (O&M) cost (FOM), Plant capacity factor and auxiliary power consumption can be obtained from tariff norms of local electricity tariff regulator. A suitable escalation rate in fuel/O&M-fixed & variable cost and prevailing fuel cost in USDollar per million BTUneeds to be considered in the economic analysis. The detailed procedure as explained in section 3.4 has to be followed for carrying the economic analysis.

e) Economic Analysis Of Direct Steam Generation Solar Aided Cc Plant

The additional capital cost of the integrated plant (due to increased size of turbo generator) can be calculated on pro rata basis from the Updated Capital Cost Estimates for Utility Scale Electricity Generating Plants obtained from Independent Statistics & Analysis, U. S. Energy Administration & Information. Additional land requirement for the installation of solar field (Parabolic trough collector system), site improvement costs and cost of the solar field including storage can be obtained from NREL's (SAM 2015.1.30)**[39]**.The detailed procedure as explained in section 3.4 has to be followed for carrying the economic analysis.

f) Economic Analysis Of Integrated Solar Ccpp With Solar Operated Vapour Absorption Chiller For Gas Turbine (Gt) Inlet Air Cooling

The incremental capital cost of the vapour absorption chiller system for power enhancement in gas turbines over and above the rated load can be obtained from literature available online. Punwani (2004)**[40]** has shown that the incremental capital cost of the vapour absorption chiller system for power enhancement in gas turbines over and above the rated load is 427.328 USD/KW. The detailed procedure as explained in section 3.4 has to be followed for carrying the economic analysis.

V. Conclusions

It has been realised world over that for sustainable development; the percentage share of power generation through renewable energy sources needs to be increased significantly. This assumes a greater importance in the wake of continuously dwindling fossil fuel resources, associated pollution and greenhouse gas emissions etc. The use of solar energy for power generation is gaining importance day by day due to these reasons. In countries like India and China, where major power generation is coal based, the concept of solar aided feed water heating (by either complete/partial substitution of turbine bleed steam) can be successfully employed by retrofitting the existing units under renovation & modernisation (R&M) programmes of power stations. The same is true with integrated solar combined cycle power plants. The payback periods are good and bound to reduce further due to continuous on going improvements in the design and manufacturing of solar collector and receiver systems. Moreover, integration of thermal energy storage (TES) with the solar field will further improve the system reliability.

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Source: Sandia National Laboratory, U.S

Fig.1: LS-3 Solar collector assembly at Kramer Junction (SEGS, Mojave Desert, California, U.S)

Source: NREL website

Fig.2: Kimberlina solar thermal power plant, (A linear Fresnel reflector based plant near Bakersfield, California.)

Source: http://www.solarreserve.com/en/global-projects/csp/crescent-dunes

Fig.3: Crescent Dunes solar power tower plant at Nevada, U.S