A Study on Thermodynamic Performance of a Coal based Thermal Power Plant considering Energy and Exergy Analysis

Thesis submitted by

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Abstract

This research conducted based on energy and exergy analysis to evaluate the performance of a 500 MW coal-based thermal power plant located at Mejia in India under the Damodar Valley Corporation. The primary objective of this research is to find out energy and exergy efficiencies of the turbine cycle, and irreversibility of different individual components of the plant at different unit load conditions (50%, 60%, 80% and 100%). Then the work extended to measure turbine heat rate at different unit loads to find out a condition at which reduction occurs in coal consumption rate as well as in harmful emissions like CO_2, NO_x, SO_2 and in ash generation.

Due to presence of a large number of concluding factors, a thermal power plant makes it challenging to accomplish the most efficient possible conversion of energy. The analysing of energy and exergy is a strong method to determine the quantity as well as quality of any energy system. It is found that the heat rate is lower at full load (100%) condition. On the basis of heat rate improvement, it is calculated that coal consumption as well as ash generation, sulphur dioxide and carbon dioxide emission are minimum at the full load condition.

In this work, an effort is made to improve performance and efficiency of the plant, and to reduce the harmful emissions such as CO_2, NO_x, SO_2, and to reduce ash generation. A suitable option to achieve this is integration solar energy with existing coal based conventional thermal power plant using parabolic trough collectors in place of low pressure heaters (LPHs) for heating the feed water. This eliminates bleeding of steam from the turbines. As a result, the steam flow rate increases through turbines, accordingly turbine power output increases and improves the turbine heat rate. This integration of solar energy leads to a significant reduction in coal consumption as well as reduction in the harmful emissions.

In brief, implementation of present model is essential in order to perform effectively by an existing coal based thermal power plant, which reduces coal consumption, reduces emissions, and moreover reduces the global warming.

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Statement of Originality

I, **Malay Kanti Naskar** registered on 15th October 2015, do hereby declare that this thesis entitled **"A Study on Thermodynamic Performance of a Coal based Thermal Power Plant Considering Energy and Exergy Analysis"** contains literature survey, and original research work done by the undersigned candidate as part of Doctoral studies.

All information in this thesis have been obtained and presented in accordance with existing academic rules and ethical conduct. I declare that, as required by these rules and conduct, I have fully cited and referred all materials and results that are not original to this work.

I also declare that I have checked this thesis as per the "Policy on Anti Plagiarism, Jadavpur University, 2019", and the level of similarity as checked by iThenticate software is 6%.

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Certificate from the Supervisors

This is to certify that the thesis entitled "**A Study on Thermodynamic Performance of a Coal based Thermal Power Plant Considering Energy and Exergy Analysis**" submitted by **Mr. Malay Kanti Naskar**, who got his name registered on 15th October, 2015, for the award of Ph.D. (Engineering) degree of Jadavpur University is absolutely based upon his work under the supervision of Dr. Sudip Simlandi and Dr. Nilkanta Barman, and that neither his thesis nor any part of the thesis has been submitted for any degree or any other academic award anywhere before.

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Nomenclature

Symbol	Description
G	*Generator*
HPT	*High Pressure Turbine*
IPT	*Intermediate Pressure Turbine*
LPT	*Low Pressure Turbine*
LPH	*Low Pressure Heater*
HPH	*High Pressure Heater*
DVC	*Damodar Valley Corporation*
GVC	*Gross calorific value in kJ*
HR	*Heat rate in kJ/KWh*
$\dot{m}$	*Mass flow rate in Kg/sec*
$\dot{E}$	*Rate of energy flow in MW*
h	*Specific enthalpy in kJ/kg*
$\dot{I}$	*Rate of Irreversibility in MW*
$\dot{Q}$	*Rate of heat transfer in MW*
$\dot{W}$	*Rate of work transfer in MW*
T	*Temperature in K*
$\dot{X}_{heat}$	*Net exergy transfer due to heat in MW*
ε	*Specific exergy in kJ/Kg*
η	*Efficiency*
HTF	*Heat transfer fluid*
SACFPG	*Solar Aided Coal Fired Power Generation*
MW	*Megawatt*

Subscripts

Subscripts	
0	*Dead state*
cv	*Control volume*
in	*Inlet*
out	*Outlet*
ai	*After solar integration*
bi	*Before solar integration*

Chapter - I

Introduction and Literature Review

Chapter I

Introduction and Literature Review

1.1 Background

One of the most important indicators of a country or a community level is its development and quality of the living, those indirectly depict by the amount of energy that it consumes as a whole because the energy consumption is proportional to its economic growth including academics as well as the industrial development. The increase in energy consumption is directly caused by a number of factors those include population growth, spreading of urbanization and industry, development of new technologies, and many others. This quick spreading of demand or related development has a lot of repercussions in our environment and its living beings like huge pollution and the greenhouse effect. Now, it is noted here that about eighty percent (80%) of the energy demand in this world is generated by the thermal power plants those are powered by fossil fuels such as coal, petroleum, fuel oil, natural gas, and others. The direct consequence of this is air pollution on global level, contributes a huge in the global warming – a real threat to the living beings at present and even in the near future. Unfortunately, after knowing these detrimental effects, installation capacity of these power plants continues to rise, India is also a witness to the installation of such thermal power plants those are the major sources of air contamination. The area around such thermal power plants is affected most. The harmful emissions into the atmosphere, out of generating electricity from the thermal power plants, is a serious issue, now-a-days, and holds a potential to threaten a large percentage of human lives in addition to alarming our biodiversity and environment. It is a prediction [1, 2] that indicates the coal basis production capacity will reach to a high of about 930 MW by the year of 2040 in India, i.e., an increase of about 300% of the present production. India ranks second in the world on coal basis production, behind China [3] where it is projected that overall coal utilization will rise from 660 million tonnes per year to 1800 million tonnes per year [4]. As a consequence, it is also forecasted that the CO_2 emissions will rise from 1590 million tonnes per year to 4320 million tonnes per year. The emissions of particulate matters (PM), SO_2, and NO_x are expected to at least double throughout this time period. It is anticipated that an overall number of premature deaths caused by pollution from the coal−fired thermal power plants will increase by a factor of two to three, reaching a range of 186500 to 229500 yearly in the year of 2030. Also by the year of 2030, a number of asthma cases linked to pollutants out of the coal−fired thermal power plants, will increase to 42.7 million [5]. All of these factors lead us towards a challenge, one of the most challenges of today's world, to reduce use of coal in order to reduce the ash generation as well as harmful emissions such as CO_2, SO_2, and NO_x among others, to create a pollution−free environment and reduce the greenhouse gas effects.

Since last few decades, many researchers reported installation of the solar photovoltaic (PV) power plants and solar thermal power plants [46−68] in order to reduce generation of electricity out of coal−fired based power plants. However, installation of any solar energy based power plant is very much expensive, even too expensive for a low capacity plant, and needs a wide area of lands for its installation that is also depend on the geometric location as well as weather condition of that area. Thus, instead of installing complete phase of a solar power plant and towards fulfilling present demand for time being, a modification of existing thermal power plant is suitable and one of the best solutions. As outcome of the literature review [1−68], this modification indicates mainly identification of losses, especially irreversibility in each component of a thermal power plant for its overhauling or repairing, possibly redesigning and/or their replacement. Therefore, this work is a consideration of an energy and exergy analysis in order to identify the energy losses or the irreversibility of each component in an India based thermal power plant to extend attention to its operation, and secondly possible inclusion of solar energy as a part of conventional plant in order to increase its thermal efficiency. This increase in thermal efficiency that reduces use of coal, as a whole, emissions out of the thermal power plant will also reduce − a remedial of the present challenge.

1.2 Basics of Energy and Exergy Analysis

Usually performance of any thermodynamic cycle is being evaluated on the basis of energy conversion following the first law of thermodynamics. It represents a measure of quantity of energy whereas knowing quality of energy is essential in order to understand extend of its conversion and a part as destruction of energy during its conversion by evaluating exergy. Exergy analysis which is assessed using the second law of thermodynamics, has been developed over the decades as one of the valuable tools with respect to design, assessment, optimization, and improvement of thermal power plants [6]. However, this exergy analysis accounts both the first and second laws of thermodynamics, in contrast to the energy analysis which is a consideration of the first law of thermodynamics only. Therefore, the exergy analysis is being utilized as a contemporary approach for process analysis because it is possible to provide an accurate description by exergy balance sheet and in a better way of representation by using the Grassman diagram of related energy and its conversion involved in the process. Accordingly, necessary analysis of a power producing unit using energy and exergy is the most accurate approach and it is able to present insight of energy conversion, all losses or destruction of energy occur in a number of components built that unit.

The first law of thermodynamics states conservation of energy while exergy able to present a destruction of energy known as irreversibility of related process. Both internal and external frictions involved on process, heat transfer to process at a finite temperature difference, mixing of two or more fluids, and chemical reactions are major sources of the irreversibility, which is possible to determine using exergy analysis only. Accordingly, present thesis considers exergy analysis using the second law of thermodynamics in order to calculate irreversibility along with maximum amount of work available or possible to extract from an energy source and quality of that source. Even in this decade, most of the power plants are built on the basis of first law of thermodynamics i.e. energetic performance criterion basis. However, such energy balance of system may not be enough to find problems arises in operation of the system. Using exergy analysis, it is possible to identify these degrading issues of any plant in operation easily, where system or component losses or destruction of energy results a less value in thermal efficiency. So, exergy evaluation is a powerful tool for measuring quality of the energy, which helps a complex thermodynamic system to work better. This work considers, accordingly a thermodynamic analysis on the basis of energy and exergy evaluation for an Indian coal based thermal power plant to identify energy losses or irreversibility in each component of the plant in order to extend attention in their operation.

1.3 Literature Review

This is an observation out of available literature that design and its optimization for a thermal power plant are so complicated in nature. Researchers have given a lot of endeavour over the years in order to understand the behaviour of energy flow and its conversion while it is flowing through the components of a plant. For basic understanding of the thermodynamics related to flow and its nature in a component, a through literature review is a prerequisite in conclusive of themes for the present thesis. In this point of view, a detailed literature review has been conducted related to energy and exergy–based analysis, and their applications in coal, gas, combined cycle, and cogeneration systems of thermal power plants as presented in subsection 1.3.1.

The emission out of thermal power plants is very harmful to our environment and its living beings. This thesis considered analysis of emissions accordingly out of the thermal power plants. Related review of literature has been then presented in subsection 1.3.2.

An interest in integration of solar energy as a part of the existing thermal power plant as well as a part of this thesis is considered in order to increase thermal efficiency of the thermal plants. Accordingly, related literature on utilization of solar energy for possible application in a conventional thermal power plant have also been reviewed and presented in subsection 1.3.3.

1.3.1 Literature related to energy and exergy analysis, reported on coal based thermal power plants

A fundamental understanding on the energy and exergy evaluation in a coal–fired basis thermal power plant using the First and Second Laws of Thermodynamics is performed by a thorough review of the related literature. The respective review of literature is presented below.

Aljundi [7] conducted an energy and exergy-based analysis of a 396 MW thermal power plant in Jordan to identify and quantify energy and exergy losses in all system components. They reported that the condenser lost 134 MW of the total energy whereas the boiler lost only 13 MW. According to their findings, the boiler destroyed the most exergy (77%), followed by the turbine (13%), and then forced draught fan condenser

(9%). They also stated that the plant's thermal efficiency and exergy efficiency were approximately 26% and 25%, respectively.

Sachdeva and Karun [8] performed an energy and exergy-based analysis to determine the magnitude, location, and source of thermodynamic inefficiencies in thermal power plants. They reported that the overall loss was 68% of total plant energy. They claimed that the boiler had the maximum exergy destruction. When comparing the three turbine stages, they observed that the HP and IP turbines produced more exergy destruction than the LP turbine, and based on feed water heater analysis, they determined that the LP feed water heater produced the most exergy destruction.

Mali and Mehta [9] performed the energy and exergy based analysis of a 125 MW coal-fired thermal power station. According to their findings, the significant losses of available energy, occur at the combustor, super-heater, economizer, and air-pre heater (APH). They observed that exergy efficiency is lower than energy efficiency, and the APH, Super heater, and economizer are the main components that lead to exergy loss. They reported a 47.43% exergy loss in the combustor. They further stated that while HP and IP turbines have the maximum performance, LP turbines require maintenance.

Alvaro *et al*. [10] conducted an exergetic and environmental assessment of a typical pulverized coal-fired power station in Brazil in order to measure the environmental impact and energy destruction by this thermal power plant. They conducted an energetic study using the second law of thermodynamics, and an environmental analysis based on a life cycle assessment (LCA) that took into account the effects of acidification and climate change. According to their report, this power plant's combustion process released 1300 kg of CO_2 per MW and total amount of exergy destruction was 122.3 MW.

Mohammad and Enadi [11] carried out a thorough thermodynamic modelling of a gas turbine power plant in Iran to evaluate the exergy efficiency, exergy destruction of each component, and cost of each flow line of the system. According to reports, the exergy destruction in combustion chamber (CC) is maximum compared to other cycle components and the exergy destruction of the combustion chamber can be decreased by raising the gas turbine inlet temperature (TIT). Additionally, they stated that a 350 K rise in TIT can reduce exergy destruction by around 22%.

Regulagadda *et al.* [12] conducted a thermodynamic analysis of a subcritical boiler, turbine, and generator of a 32 MW coal-fired power plant under various operating conditions, including various operating pressures, temperatures, and flow rates in order to identify the parameters that maximise plant performance. The energy efficiency and exergy efficiency of the plant were reported to be 30.12% and 25.38% respectively for the gross generator output. They also stated that the maximum exergy destruction occurred in the boiler.

Ganapathy *et al.* [13] conducted an exergy analysis on a 50 MW unit of lignite-fired steam power plant in Tamil Nadu, India in order to determine irreversibility and exergy losses in various plant components and to improve the performance of existing systems, processes and components. According to their findings, the condenser experiences the greatest energy losses of 39%, whereas the combustion chamber experiences the greatest exergy losses of 42.73%. Additionally, they mentioned that the exergy efficiency of the plant is 27%.

Mborah and Gbadam [14] analysed a thermodynamic system of a 500 KW Steam Power Plant at Benso Oil Palm Plantation (BOPP) to identify the locations and magnitudes of losses in various components in order to maximize the performance. They reported that almost 50% of the heat energy produced in the combustor is lost. They also added that energy and exergy analysis of individual components and combustor improvement will assist in maximizing plant performance.

Shamet *et al.* [15] analysed the four power plants at Garri in Sudan to identify irreversibility, heat loss, and energy destruction of specific components. According to their assessment, the heat losses in the boiler and in the condenser are 29.1% and 67% respectively whereas thermal efficiency of the plant is 21.12%. Additionally, they estimated that the rate of exergy destruction in the boiler and condenser are 82.72% and 4.21%, respectively.

Kami *et al.* [16] evaluated the energy efficiency, exergy efficiency, and irreversibility of all cycle components of a 200-MW Shahid Montazeri Power Plant located at Isfahan in Iran. They reported that the condenser losses account for 70% of total energy, while the boiler losses account for 10% of total energy. They also reported that 85.66% of the entire exergy entering the cycle is lost, whereas condenser exergy loss is only 1.53%. They also evaluated that the efficiency of energy and exergy is 32% and

35.2%, respectively. According to their findings, each 0.01 bar rise in condenser pressure reduces power generation by around 0.7 MW.

Rahman [17] analysed a 28 MW thermal power station of IOCL, Guwahati, using energy and exergy based analysis of its individual components. According to their findings, the boiler had the most exergy destruction at approximately 72.40%, followed by the deaerator at around 16.74%, and finally the turbine at approximately 5.47%. They also calculated that the thermal efficiency of the plant was 26%, while its exergy efficiency was only 24.27%.

Kamate and Gangavati [18] conducted an exergy-based analysis of a bagasse-based cogeneration plant with a capacity of 2500 Tons of cane per day (tcd) sugar refinery to evaluate overall and component efficiencies, as well as to identify and assess thermodynamic losses for a wide range of steam inlet conditions. They reported that under optimum steam input parameters of 61 bar and 475 °C, the backpressure steam turbine cogeneration plant has energy and exergy efficiency of 0.863 and 0.307, respectively, while the condensing steam turbine plant has energy and exergy efficiency of 0.682 and 0.260 respectively. They also mentioned that the boiler is the least efficient component of the plant, while the turbine is the most efficient.

Datta *et al*. [19] estimated energy and exergy efficiency of a 210 MW coal-fired thermal power station in India using plant operating data at different unit loads such as 100%, 75%, 60%, and 40% of full load, different condenser pressures, with and without regenerative heaters, and different turbine controlling settings. For the analysis, they divided the entire power plant into three zones: (1) the turbo-generator and its inlets and outlets, (2) the turbo-generator, condenser, feed pumps, and regenerative heaters, and (3) the entire cycle with boiler, turbo-generator, condenser, feed pumps, regenerative heaters, and plant auxiliaries. They reported that the boiler is a major source of irreversibility in the power cycle, with 60% exergy destruction occurring. They claimed that part-load operation increases irreversibility in the cycle while increasing condenser back pressure decreases exergy efficiency. They also mentioned that gradually withdrawing the high pressure heaters assisted in a steady increase in exergy efficiency.

Dai et al. [20] performed an energy and exergy analysis for various cogeneration power plants used in a cement plant to recover waste heat from pre-heater exhaust and clinker cooler exhaust gases, a single flash steam cycle, dual-pressure steam cycle, organic Rankine cycle (ORC), and the Kalina cycle. They mentioned that the exergy losses in the turbine, condenser, and heat recovery vapour generator are relatively large, and lowering these exergy losses could improve by using the cogeneration system. In addition, they reported that, when compared to other systems, the Kalina cycle could achieve the best performance in a cement factory.

Rosen [21] conducted energy and exergy-based comparison analysis of coal-fired and nuclear power plants. They claimed that the overall energy and exergy efficiencies for the coal-fired power station are 37% and 36%, respectively, whereas for the nuclear power station are 30% and 30%, respectively. They stated that the highest energy losses associated with emissions while emission related exergy losses is only 10% of total exergy losses. They also observed that the remaining 90% of exergy losses are due to internal consumptions, mainly in components that generate heat through combustion or nuclear reactions, as well as components that transmit heat across large temperature variations.

Aliyu *et al*. [22] performed energy and exergy analysis of a power plant based on the design data to calculate the exergy efficiency and exergy destruction of each component. As reported, the turbine had an exergetic efficiency of over 92%, making it the most efficient part of the steam turbine cycle, whereas the condenser had an exergetic efficiency of less than 63%. They also stated that the output and efficiency of the low-pressure steam turbine are strongly influenced by the super heater pressure, reheat pressure, and steam quality at the exit of the turbine.

Rajper *et al*. [23] performed energy and exergy analysis of 210 MW dual-fire, subcritical, reheat steam power plant located near Jamshoro in Pakistan. They reported that the net power output, energy efficiency and exergy efficiency are 186.5 MW, 31.37% and 30.41% respectively under design operating conditions. They stated that maximum energy loss is observed in the condenser i.e. 280 MW (68.7%) followed by boiler with 89 MW (21.8%). They also stated that the maximum exergy destruction is found in the boiler with 350 MW i.e. 82.11% of the total exergy destruction, followed by turbine with 43.1 MW (10.12%) and condenser 12 MW (5.74 %).

Memon, *et al*. [24] performed an energy and exergy based analysis of an open cycle gas turbine power plant. They found out impacts on overall cycle efficiency and CO_2 emission due to variation of some operating parameters like inlet temperatures of compressor, turbine and pressure ratio. They reported that the largest exergy destruction take places in the combustion chamber and exhaust stack whereas the exergy destruction in the regenerative cycle is comparatively lower. They also stated that the CO_2 emission and costs involvement will be minimum at maximum performance of the plant.

Ehyaei *et al*. [25] performed energy and exergy based analysis of the Shahid Rajaee thermal power plant located at Ghazvin in Iran in order to determine the effects of the inlet fogging system on the first and second law efficiencies. They introduced a new function for system optimization that was social cost of air pollution from the power plant. They claimed that adopting an inlet fogging system improved the average output power production, the first and second law efficiency over the period of three months of a year i.e. June, July, and August by 7%, 5.5%, and 6%, respectively.

Gulhane and Thakur [26] performed an energy and exergy based analysis of a 6 MW gas and coal power plant. They have taken numerous efforts to minimise heat loss through proper boiler insulation. They stated that the boiler having maximum exergy destruction at home load 1.1 MW is around 83.35%. They observed that when the plant is operating at its maximum capacity i.e. 5.6 MW, exergy destruction of the boiler drops to 76.33%. In addition, they claimed that under peak load conditions, the first and second law efficiencies become maximum whereas irreversibility becomes minimum.

Hussain *et al.* [27] conducted an exergy based analysis of a 120 MW steam power plant in Malasiya and compared the three turbine stages of this plant. They observed that the boiler has the maximum exergy destruction of 54 MW and the high pressure (HP) and intermediate pressure (IP) turbines destroy more energy than the low pressure (LP) turbine. They also reported that the feed water heater creates a considerable amount of exergy destruction.

Tsatsaronis and Moung-Ho [28] analysed the thermodynamic performance and cost effectiveness of a thermal systems and suggested the possibility for its advancement. They reported that improvement efforts should be given only on these avoidable parts where exergy destruction and investment cost are associated. They also evaluated exergy losses and investment costs for compressors, turbines, heat exchangers, and combustion chambers.

Adibhatla and Kaushik [29] performed an energetic and exergetic analysis of a 660 MW coal fired supercritical thermal power plant in order to evaluate energy as well as exergy losses of various components, including the boiler, high pressure turbine, intermediate pressure turbine, low pressure turbine, condenser, gland steam coolers, condensate extraction pumps, low pressure heaters and drip heaters. They reported that boiler has the maximum rate of exergy destruction than any other component in the power plant, followed by turbine. They added that the rate of exergy destruction for the turbine operating under sliding pressure has been significantly reduced under part-load conditions. They calculated the rate of exergy destruction in the turbine as 49.16 MW, 43.22 MW, and 43.92 MW, respectively, for constant pressure operation at 100%, 80%, and 60% of part load conditions.

Abuelnuor *et al.* [30] analysed to find out the exergy efficiency and exergy destruction of each component of the 180 MW Garri "2" combined cycle power plant. They reported that gas turbine accounted for 63% of all exergy destruction, followed by steam turbine (6.4%), heat recovery steam generator (6.3%), exhaust stack (4.7%), compressors (3.8%), and cooling systems (2.3%). Additionally, they calculated that the energy and exergy efficiencies of the plant are 38% and 49%, respectively.

Hasti *et al.* [31] performed an exergy analysis for an ultra-super-critical power plant using a process simulation and a computer model in Microsoft Excel. They stated that 615 MW of exergy is destroyed in the furnace, which accounts for approximately 86% of total energy losses; however, only 15 MW of energy is destroyed in the condenser, and 45% of energy is destroyed in the turbine which has an average exergetic efficiency of 82%. Additionally, they determined that the exergy efficiency of the plant is 70%.

In order to assess the impact of heat recovery on the system performance, **Vandani *et al*. [32]** performed energy and exergy based analyses of boiler blow-down heat recovery of a steam power plant in Iran. According to their analysis, the net generated power increases by 0.72% when blow-down recovery is used. They also reported that the energy and exergy efficiency of the system increased by 0.23 and 0.22 respectively due to use of boiler blow-down heat recovery.

Olaleye *et al*. [33] performed an energy and exergy based analysis of a 550 MW supercritical power plant integrated with post-combustion CO_2 capture. They observed that the once-through boiler had the largest exergy destruction but had little effect on the system's potential for fuel savings, whereas the turbine subsystems had lower exergy destruction than the boiler subsystem. Additionally, they reported that increasing turbine performance and lowering the forces responsible for the CO_2 capture process assist to improve the integrated system's rational efficiency.

During continuous operation, the performance of energy systems gradually deteriorates away from the ideal work conditions.

Wang *et al*. [34] developed an energy and exergy based model to successfully detect the components with performance deterioration by calculating the endogenous and exogenous exergy destructions of various components in a coal-based thermal power plant. According to their observation, endogenous exergy destruction arises from a component's inherent irreversibility, but exogenous exergy destructions is brought on by the inefficiency of the remaining components. They observed that the boiler and condenser are responsible for a significant amount of exergy losses. Additionally, they stated that exogenous exergy destructions is about 41% of total exergy destruction.

Zhao *et al*. [35] performed an energy and exergy based analysis for the turbine system of a 1000 MW double reheat ultra-supercritical power plant. They reported that the turbine, particularly the high pressure and low pressure cylinders, experienced the biggest exergy loss due to irreversibility. They claimed that for the regenerative system, exergy loss is lower whereas exergy efficiency is higher in the double reheat unit than in the single reheat unit. They also observed that exergy loss in the condenser of the double reheat unit is relatively smaller than that in the condenser of the single reheat unit.

Oko *et al*. [36] performed an energy and exergy based analysis of an integrated (IPP) gas, steam, and organic Rankine cycle (ORC) thermal power plant. They reported that the IPP system has increased its exergy efficiency by 1.93% and its energy efficiency by 1.95%. They also stated that the combustion chamber of the combined cycle had the highest rate of exergy destruction, which is 59%. The evaporator in the ORC had a rate of exergy degradation that is 62% higher than any other component.

Rozpondek and Siudek [37] reported in their paper that new and improved technologies can greatly reduce the emissions produced per ton of burning coal. They stated that the wet methods of desulfurization at present are the widest applied technology in professional energetics.

Zdemir *et al*. [38] performed exergoeconomic analysis on individual sub-systems like a ventilation fan (VF), a flue gas clean-up chamber (FBCC), a heat recovery steam generator (HRSG), a cyclone (CY), an economizer (ECO), an aspiration fan (AF), a pump (P), and a chimney (CH) of an FBCC steam power plant located in the city of Izmir in Turkey. They reported that at a steam mass flow rate of 1.861 kg/s, exergetic efficiency of plant is 20.28%.They proved that the FBCC has the highest exergy destruction rate with an irreversibility rate of 89.2%, followed by the HRSG, VF, ECO, AF, CH, and P. They also stated that the FBCC steam plant has a unit exergy cost of 17.88 US dollars per gigajoule and an exergy cost of 93.57 US dollars per gigajoule and 93.57 US dollars per hour for the steam it produces.

Adibhatla [39] performed energy and exergy analysis of a 660MW supercritical thermal power plant under various load situations like 100%, 80%, and 60% and under various continuous and pure sliding pressure. According to their observation, the boiler in the plant has the greatest rate of energy destruction. They stated that compared to the constant pressure operation, the sliding pressure operation significantly reduces the rate of exergy destruction at part load conditions for the turbine. Hence, the sliding pressure operation is preferable for running under part load conditions.

Vuckovic *et al*. [40] performed advance exergy analysis and exergoeconomic performance evaluation of thermal processes in an existing industrial plant to determine the performance of individual components and its exergy efficiency. They reported that with increase of exergy efficiency, economy of the plant would improve.

From the above literature, it is observed that irreversibility ascends from the power plants is a form of environmental damage as it creates more emission of pollutants. The environmental damage is certainly minimised by preserving exergy through improved efficiency of the plants. Therefore, it is essential for monitoring performance of an operative power station that includes exergy or 2nd law analysis besides conventional energy or 1st law analysis. It is also observed that work on boiler emission control to minimise the impact on environment is rarely available in the literature. In addition, the crisis exists in reduction of fossil fuels whereas the electrical demand increases continuously, leads to operate existing power plants in a better mode for effective performance. Hence, few literature based on emission control of conventional coal fired power plants are reviewed as follows:

1.3.2 Literature related to emissions and its control from coal based thermal power plants

The emissions of conventional thermal power plants has a life threatening effect and thus this section states a thorough review of related literature in order to obtain necessary information and basics of the emissions and its control. The related review is presented below.

Narula *et al*. [41] reported in their study that the addition of CO_2 amine scrubbers at the back end of the power plant can reduce CO_2emissions. They also stated that it leads to an increase of heat rate from 9,800 kJ/kWh to 13,250 kJ/kWh of the plant but on the other hand it increases capital cost of the plant approximately 77%.

Nihalani et al. [42] demonstrated various technological methods such as the Flue Gas Desulfurization (FGD) System, Spray Dryer Absorber (SDA), Circulating Dry Scrubber (CDS), Limestone-based Wet FGD, Low NO_X burners, Selective Non Catalytic Reduction, Electrostatic Precipitator, and Bag House Dust Collector to reduce CO_2, SO_2, NO_x, particulate matter, and other emissions from power plant.They also stated that every control technology has benefits and drawbacks. Major characteristics, potential operating and maintenance cost influence the choice of one technology over another.

Rana and Mehta [43] evaluated energy and exergy efficiencies, exergy destruction and turbine heat rate (HR) at maximum continuous rating (MCR) of 70% and 85% for a steam turbine.

Parker *et al*. [44] reported that approximately 46% of the world's power generation is estimated to be from the combustion of coal. They stated that 50% of the electricity generated in the USA, 89% of the electricity generated in China, and 81% of the electricity generated in India respectively. They also stated that by the year 2025, the combustion of coal for the purpose of producing electricity will account for approximately 41% of the world's total CO_2 emissions.

Barzegar *et al*. [45] performed an exergy based analysis on a gas turbine power plant to find out the best operating condition for reducing the environmental effect due to emissions. According to their observation, increase in exergy efficiency leads to reduce in CO_2 emission.

Ahmad *et al*. [46] reported the disposal problem of coal ash with the development of leachate. They stated that this leachate contains heavy metals which create an enormous hazard for the environment. They also stated that coal-based power generation produce number of unfavourable emissions which create adverse effects on environment including the acceleration of global warming and the creation of a difficult management challenge involving fly ash.

It is noticed from the above literature that the reduction of environmental damage by analysing emission of pollutants is less cited. Hence, reduction analysis of conventional power plants in consumption of coal, generation of ash, emissions of SO_2 and CO_2 carried out in the present work. It is also notice that an integration of solar energy with the conventional existing power plants may reduce the harmful emissions. Hence, some literature related to solar integration with coal fired power plants are studied as follows:

1.3.3 Literature related to possible utilization of solar energy in an existing coal based thermal power plant

In order to reduce emissions of harmful gases and to increase thermal efficiency of existing thermal power plants, an integration of solar energy is considered in this work. A thorough review of the related literature has also been conducted, thus, for understanding basics of solar energy and its applications. The review of related literature is presented below.

Ghorpade and Goswami [47] performed exergy based analysis of an existing coal-based thermal power plants integrating with solar energy to optimize solar-aided coal-fired thermal power plant. They reported that thermal efficiency of the solar aid thermal power plant increases and greenhouse gas emissions decreases. They also stated that the availability of solar energy at different times of the day and year, the initial cost of the investment, land acquirement, and the need for battery storage were also stated by them as disadvantages.

Zhen *et al*. [48] integrated solar energy with coal-fired power plants where feed water heating by solar energy is considered. They investigated the plant performance under various unit load circumstances. They reported that in a year with average radiation, the optimal aperture area of the solar field and the lowest Levelized Electricity Costs are 115395 m^2 and Rs 0.28/kWh respectively, in a year with exceptionally low radiation, 138945 m^2 and Rs 0.61/kWh and in a year exceptionally high radiation, 91845 m^2 and Rs. 0.26 /kWh.

Ashouri *et al*. [49] analysed a low temperature organic Rankine cycle (ORC) based on flat plate solar collectors with a storage tank in order to evaluate the system's performance under various parameters and to identify the main causes of energy destruction as well as the possibility of reducing them. They considered four different working fluids i.e. R245fa, R134a, pentane, and toluene to evaluate the system. They stated that the primary sources of energy destruction are, in order, the solar collector, thermal storage tank, and vapour generator. They also stated that, at the same load, pentane performs best, followed by R245fa, toluene, and R134a with exergy efficiencies are 24.08%, 22.53%, 22.09%, and 21.76%, respectively.

Mills [50] analysed a coal based thermal power plant by combining solar energy to increase the efficiency and reducing environmental impacts. He stated that his technique can improve operational flexibility, moderate plant costs and reduce emissions. He also stated that use in solar energy in coal based thermal power plant can significantly reduce the coal consumption.

Jaiganesh [51] developed the Glass to Glass Photovoltaic Thermal System (G2G-PVTS), which combines a photovoltaic (PV) with a flat plate solar water heating system (FPSWHS). They stated that wateris used as a coolant inside the copper fins of FPSWHS technology, to absorb heat from the PV panel and to store it in the insulated storage tank. They reported that the electrical efficiency of the G2GPVT panel was 0.7% higher than that of the standard G2TPV panel. They also stated that thermal efficiency of G2GPVT panel was 44.37%.

Ikhlef and Larbi [52] analysed Hassi R'mel solar power project located in southern Algeria using multi-objective optimization to find out the ideal circumstances and parameters for maximization of power generation from future power plants. The project has 25 MW parabolic troughs coupled with 130 MW cycle gas turbines. They reported that the lowest levelized cost of electricity (LCOE) is about Rs. 504/kWh for a solar multiple of 1.6. They stated that the optimum value of fossil fill fraction is 0.2, with a capacity factor of 60%. They also stated that the best optimization of the storage system is 4 h when power plant produced 118.26 GW per year.

Mihoub *et al*. [53] analysed a 50 MW Concentrated solar power (CSP) plant to develop a method for identifying the best configuration and design for future solar thermal power plants. He implemented this configuration and design in Hassi R'mel solar power project located in southern Algeria with maximum of annual power generation and minimum of Levelized Cost of Energy (LCOE).

Boukelia *et al*. [54] performed an analysis to optimize 50 MW thermal power plant combined with thermal energy storage, and a fuel backup system which is integrated with two parabolic solar troughs. They reported that the arrangement improvesperformance, power output and reduces the environmental effectsof the power plant. They also stated that further improvement is possible if solar radiation, cooling system technology, and critical plant design parameters such as energy, environment, exergy, and economy are considered for the analysis.

Bishoyi and Sudhakar [55] performed an energy and exergy based analysis of a 100 MW parabolic trough solar thermal power plant located at Rajasthan in India using System Advisor Model (SAM) to evaluate its optimum design and thermal performance. They stated thatthe thermal energy storage has 6 hours back up. They reported that the efficiency of the plant is 21% and an annual production of power is about 285,288,352 kWh.

Guzman *et al* [56] conducted an analysis using System Advisor Model ofa 50 MW parabolic trough solar power system with thermal energy storage and natural gas as backup located at Barranquilla in Colombia. They reported that the minimum LCOE for a typical solar field is Rs. 40.50/kWh, which is costly compared to traditional systems but has a favourable influence on carbon footprint. They also stated that using natural gas, this value is lowered to Rs. 15.81/kWh.

Rohani *et al*. **[57]** conducted an analysis based on operating data of Andasol parabolic trough power plant located at Guadix in Spainusing the Fraunhofer in-house simulation software ColSimCSP. They compared the Simulation data and actual data for plant performance characteristics like solar field thermal power and net electrical energy yield on a component level. They observed that solar field thermal power has the lowest mean deviation with 0.59% accuracy and reliability. They also stated that the net electrical energy yield's 2.3% mean deviation reveals an excellent match between simulation and operating data.

Hong *et al*. **[58]** conducted an analysis of a 330 MW coal-fired power station integrated with solar power to evaluate solar efficiency, exergy efficiency and exergy destruction. They used ParabolicTrough collectors to heat feed water at 300°C using solar energy. Thus they eliminated steam bleeding from high-pressure turbine. They found that integrated power plants produce more work than coal-fired plants. ButSolar irradiation and incidence angle affect solar-to-power performance for a given turbine load. They stated that as solar irradiation rises, Parabolic Trough collectors can substitute bleed steam for feed water preheating and can improve solar-to-electricity efficiency. They also stated that at high sun irradiation, the heat gathered by Parabolic Trough collectors exceeds the need to warm the feed-water, reducing the solar collector field's efficiency.

Zhang *et al*. **[59]** conducted an analysis of two coal fired thermal power plant integrated with solar energy to heat superheated steam and sub-cooled feed water. They used a molten salt as heat transfer fluid. They reported that when 6.1% solar input is used to heat superheated steam, the integrated CSP coal plant was 6.1% more efficient than conventional power plant. They also repored thatwhen 4.9% solar input is used to heat sub-cooled feed water, CSP-coal plant efficiency was 3.6% greater than conventional power plant.

Gupta and Kaushik [60] performed an energy and exergy analysis for the various components of a thermal power plant integrated with solar energy. They reported that the condenser is responsible for the greatest amount of energy loss, while the solar field is the location that suffers the greatest amount of exergy loss.

Dabwan *et al.* [61] presented a new hybrid solar preheating intercooled gas turbine (SP-IcGT) cycle. They used parabolic solar trough to preheat the compressed air before entering the combustor. They considered several performance indicators under Guangzhou (China) weather data. They reported that the SP-IcGT is better than the conventional hybrid solar preheating gas turbine (SP-GT) system because it can enhance the fuel-based efficiency by 19.35%. They stated that the SP-IcGT has lower specific fuel consumption i.e. 7017 kJ/kWh compared with the 10362 kJ/kWh for SP-GT. They also stated that the SP-IcGT has the highest fuel-based efficiency i.e. 51.4% and a levelized electricity cost is Rs. 377.86/kWh.

Zhao *et al.* [62] modelled a hybridized solar energy with fossil power plant to improve reliability and efficiency. They developed a new concept of hybridization of solar energy with typical 100 MW–1000 MW coal-fired power plants which can preheat the feed water before entering the boiler at approximately 300°C. They used LS2 parabolic trough collectors with a concentration ratio of 71 and an optical efficiency of 0.76. They reported that due to solar hybridization, the net solar-to-electricity efficiency will increase.

Bijarniya *et al.* [63] demonstrated the concentrated solar power technique currently in use in India because conventional power plants face a number of challenges, including limited access to fuel, limited land for construction, and other environmental concerns. They stated that power generation with Concentrating Solar Power (CSP) is the best direction for India in the field of solar technology in the coming years.

Kumar *et al.* [64] reported an overview of the development of concentrated solar power in India. They stated there are more than 300 days a year with clear skies in India, which means that the country has an enormous solar power potential for the generation of solar electricity per watt of installed capacity. The Indian government has set a target of an additional 100,000 MW of solar power output by the year 2022. They stated that India is capable of establishing a CSP with a capacity of one thousand gigawatts. Several CSP plants across the country are operating effectively. They also stated that Jawaharlal Nehru National Solar Mission (JNNSM) was established by the Ministry of New and Renewable Energy（MNRE） of the Indian government to promote various applications of CSP and other solar applications in India.

Rubin et al. [65] examined historical development of two commonly used emission control technologies. These technologies are flue gas desulphurisation (FGD) and selective catalytic reduction (SCR) systems to control oxides of sulphur and nitrogen emissions respectively.

Prosin *et al*. [66] hybridised a coal based thermal power plant with concentrated solar thermal (CST) fields where boiler combustion air and feed water are heated by solar energy. They observed that it increases overall plant efficiency and reduces fuel consumption about 8% of annually total fuel consumption.

Shagdar *et al*. [67] integrated solar energy with a 300 MW coal-fired thermal power plant and performed energy and exergy based analysis at different unit load conditions to improve techno-economic and ecology aspects. They observed that thermal efficiency is enhanced by 7.19% with coal consumption reduction of 45.3 g /kWh. They suggested that optimal aperture area of the heliostat solar field is 330 330 m^2. They also reported that minimum levelized cost of electricity is Rs. 15.24/kWh.

Baloda and Soni [68] evaluated the technical, environmental, and economic aspects of incorporating concentrated solar energy for feed water heating into an existing 210MW coal-fired power station. They used solar energy to replace the steam extracted from the first stage of the turbine. They observed that integrating existing coal-fired power plants with solar energy lowers coal consumption and CO_2 emissions.

There have been few studies on solar hybridization, which involves installing parabolic solar troughs in place of the low pressure heaters（LPHs） for similar to the regeneration to improve the performance and lower emissions of a traditional coal−fired thermal power plant. Hence, an effort is made in this work where solar energy is integrated with a conventional coal fired power plant by deploying parabolic solar troughs at the place of the LPHs for similar to the regeneration.

1.4 Closing Remarks Out of the Literature Review

The above cited literature has been reviewed carefully for our basic understanding and on the basis of review relevant to the work of present thesis, it is being concluded as

i. The irreversibility arises in operation of thermal power plant has a detrimental effect to our environment together with a higher rate of pollutant emission. It is possible to reduce amount of such damage significantly by maintaining quality of energy or exergy in terms of enhancing efficiency of the plant in operation. Exergy analysis also known as Second Law of Thermodynamics analysis, should be performed alongside of the conventional energy analysis also known as First Law of Thermodynamics, while monitoring operational of a power station.
ii. It has also been observed from present review that literature reported detailed analysis on emission of pollutants and intending to reduce negative impact of emission to our environment is available rarely.
iii. Although installation of solar energy is very expensive, integration of solar energy as a part of a conventional power plant may enhance thermal efficiency of that plant. Literature related to solar thermal power plants is available a lot, however integration of the solar energy into conventional thermal power plant has not been reported.
iv. Furthermore, there is a crisis in using fossil fuels whereas demand of energy supply is increasing day–by–day. This requires operation of all existing power plants in a more efficient manner with a goal of enhancing their overall efficiency.

The above major concluding remarks out of the present review are our encouragement in identifying the theme of present thesis.

1.5 Objectives of the Present Thesis

At this present scenario, requisite demand of electricity is increasing day–by–day due to development of new technology and increasing population, hence spreading of the urban and city, and industry etc. This demand is being supplied mostly by operation of coal based thermal power plants. It is also observed that the installation of solar energy based power plants is minimal in India as these plants are very expensive in installation, and is in the developing stage, whereas supply of the electricity is to be maintained up to its demand at any cost. In this regard, operation of conventional thermal power plants is the only solution. Alongside to the supply of electricity by conventional power plants, there is a huge emission of pollutants into the environment – a real threat to the living beings and its biodiversity. This indicates a necessity of control in the operation of a conventional power plant, hence possible reduction in use of the coal, and possible use of solar energy by integrating it with the conventional power plant.

Based on the review of literature, the periodic identification of energy loss or energy destruction in the plant, especially in its components, in order to extend the possible attention to the specific component where the loss is high, is one of the remedial using energy and exergy analysis. Corrective to the loss may increase the thermal efficiency of the plant, and may reduce the emissions. Integration of solar energy may further help in increasing the thermal efficiency, then emission may reduce further to a standard level. Hence, in perspective to an Indian coal–based thermal power plant, primary objectives of the present thesis are set as:

i) To identify and quantify the loss of energy and irreversibility of a coal–based thermal power plant on the basis of energy and exergy analysis with a goal to reduce coal consumption as well as ash generation and harmful emissions.

This is mainly identification of components in the plant where irreversibility is high, accordingly operator may set their attention in overhauling or replacing respective components. The periodic overhauling or replacing of such damaged components on the basis of analysis will increase thermal efficiency, which is a reciprocal to coal consumption. This kind of handy remedial may reduce the coal consumption to a level, hence a reduction in harmful emissions.

ii) To integrate of solar energy in an existing thermal power plants in order to increase their overall thermal efficiency.

Although incorporating solar energy is very expensive, however by installing suitable solar troughs as a part of the existing power plant, the reheaters and/or regenerative heaters may be replaced. This will increase thermal efficiency of existing conventional power plants that also cuts down amount of coal uses, and lowering production of undesirable emissions such as carbon dioxide, sulphur dioxide, nitrogen oxide, ash etc.

1.6 Layout of the Thesis

The layout of the research works performed as content of the present thesis under the titled ***"A Study on the Thermodynamic Performance of a Coal-based Thermal Power Plant considering Energy and Exergy Analysis"*** is presented below chapter-wise.

Chapter-II: Energy and Exergy Analysis for Turbines (HPT, IPT and LPT) of a 500 MW Coal-based Thermal Power Plant – As output of a thermal power plant is being disbursed through its turbines, accordingly our preliminary understanding instigated to identify the energy losses or irreversibility in the turbines using energy and exergy analysis. This study has been extended for different unit load conditions: 100%, 80%, and 60%.

Chapter- III: Energy and Exergy Analysis for Individual Component of a 500 MW Coal-based Thermal Power Plant – Identification of energy losses or irreversibility for individual component of coal-based thermal power plant is also essential, accordingly energy and exergy-based analysis has also been executed for the individual component of plant under different unit loads: 100%, 80%, and 60%.

Chapter-IV: Emission Control for a 500 MW coal based Thermal Power Plant – It includes an analysis for reduction in consumption of coal, generation of ash, and emissions of SO_2 and CO_2 by improvement in the heat rate at different unit load conditions: 100%, 80%, and 60%, and it also includes identification of an ideal operating unit load condition.

Chapter-V: Integration of Solar Energy as a Part of Existing 500 MW Coal-based Thermal Power Plant- Excluding installation cost of solar heaters and available free solar energy, integration of solar energy into a conventional coal-based thermal power plant is considered here with a goal of maximizing its overall thermal efficiency, which also reduces the coal consumption, ash production, sulphur dioxide and carbon dioxide emissions.

Chapter–VI : Conclusion and Future Work.

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Chapter - II

Energy and Exergy Analysis

of Turbines of a 500 MW

Coal-based Thermal Power Plant

Chapter–II

Energy and Exergy Analysis of Turbines of a 500 MW Coal–based Thermal Power Plant

2.1 Introduction

The amount of energy used per person is an indicator for development and standard of living of a country. Therefore, the power plant industries play a pivotal role in economic development of the country. This issue attracts an interest to the researchers in order to find better methods of conserving energy and making use of the resources systematically. In the recent years, emission of greenhouse gases from the power plant industries increases global warming, which is a real threat to the planet. As reported by Demirbas [69], burning of fossil fuels is mainly responsible to the global warming (almost 98%) through CO_x emissions. However, the damaging effect of the power plants on environmental is possible to reduce by increasing efficiency of the plants [70], which leads to a strong need to maximize effective use of the energy in order to address the supply of the global energy demand as well as to maintain the environmental crises. An effective effort is to upgrade the efficiency and performance of already existing plants by analyzing and adjusting the existing components and energy resources in operation in the plants generating the electricity [70]. Initially, energy analysis based on the first law of thermodynamics is used extensively in design and performance evaluation of the energy related to engineering systems [70–72]. Since last few decades, an interest is growing in analysing the plants based on the second law of thermodynamics, is known as exergy analysis [70–73]. Following the mass and energy balances, the exergy analysis for any thermal plant is a useful tool for developing, evaluating, and optimizing performance of the plant [70–76]. The exergy analysis determines extent and location of exergy destructions, which helps in enhancing preexisting structures and related machineries, or to design and implement new ones [77]. This work is a cosideration of energy and exergy analysis of an existing power plant in order to identify the locations of exergy destruction and necessary inprovement of its efficiency. As a basic consideration, in the present chapter, only turbines of the plant are considered for analysis.

2.2 Consideration of Physical Problem

In this chapter, an energy and exergy analysis is performed for turbines of a coal based thermal power plant with a capacity of 500 MW owned by Damodar Valley Corporation which is located at Mejia of West Bengal in India. Fig.2.1 is a schematic presentation of the turbines of the plant. For present analysis, the necessary data have been collected. Table–2.1, Table–2.2 and Table–2.3 represent the essential thermodynamic parameters at 100%, 80% and 60% unit load conditions, respectively. A control volume is considered (enclosed by a continuous line as shown in Fig.2.1) includes three turbines those are coupled to a generator (G). These turbines are referred as high pressure turbine (HPT), intermediate pressure turbine (IPT), and low pressure turbine (LPT). After leaving boiler, steam contents high energy and high exergy flows to the HPT, which is represented as flow–1. After extracting work by expanding of the steam in the HPT, steam exits with comparably less energy and less exergy which is marked as flow–2. Then the exit steam flows back to the furnace again for reheating. The high–pressure heater receives a portion of the steam extracted from the high–pressure turbine (flow–3). The steam after reheating, which is denoted as flow–4, contents an increased amount of energy as well as exergy goes to the IPT to expand in this intermediate stage. After expanding in the IPT, low–pressure steam (flow–5) sends to the LPT for further expansion. The steam leaving from the IPT sends to the high–pressure heater, which is noted as flow–6. Steam denoted as flow–7 leaving from the IPT moves to the deaerator. Steams extracted from the low–pressure turbine and send to low–pressure heaters denoted as flow–9, flow–10, and flow–11. Steam exits from the LPT, denoted as flow–8, flows to the condenser in the system. As per the data, the wet steam with a dryness fraction of 0.91 is evacuated from the LPT, which is maintained at a low pressure of around 10.47 kPa (absolute).

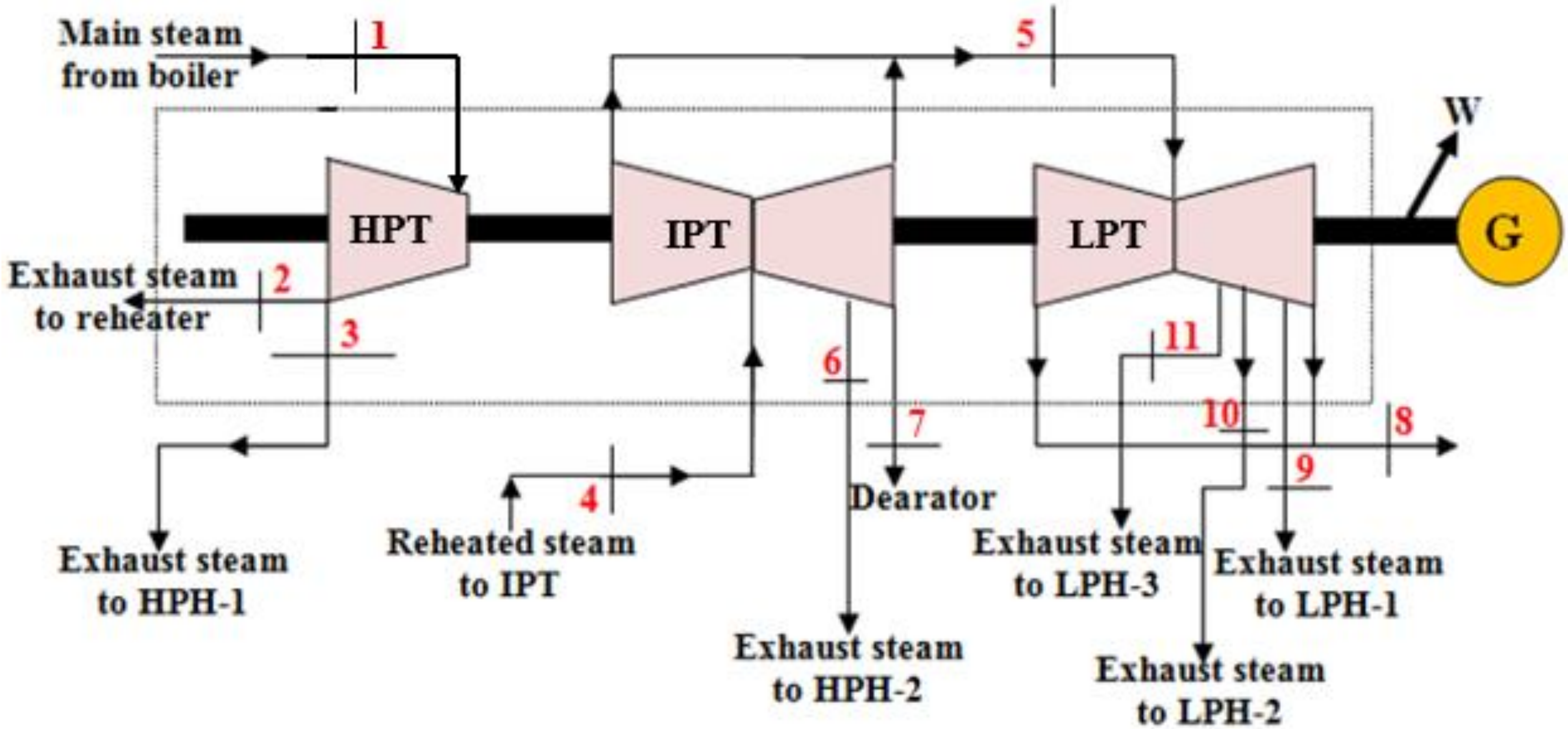

Figure – 2.1: Flow path in turbines for 500 MW capacity power plant

Table-2.1: Thermodynamic properties of the relevant flow at full load of a 500 MW capacity coal-fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (ºC)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	415.47	537		172.21	3389.77	6.39	1489.6	1408.34	618.88
2	369.9	337.4		45.62	3046.97	6.45	1128.31	1127.07	417.36
3	45.57	337.4		45.62	3046.97	6.45	1128.31	138.84	51.41
4	369.9	537		41.06	3529.43	7.19	1392.6	1305.53	515.12
5	308.59	289.1		7.25	3035.95	7.24	882.4	936.87	272.3
6	23.38	411.1		17.55	3276.44	7.23	1126.6	76.6	26.34
7	37.93	289.1		7.25	3035.95	7.24	882.4	115.15	33.47
8	264.48	46.71	0.9124	0.1047	2351.48	7.48	127.43	624.08	33.82
9	11.67	73.35	0.9536	0.36	2741.35	7.46	328.15	29.73	3.83
10	21.03	133.6		1.55	2851.39	7.33	560.7	57.59	11.79
11	11.41	190.9		2.87	3040.93	7.29	678.98	32.51	7.75

Table–2.2: Thermodynamic properties of the flows at 80% unit load of a 500 MW capacity coal–fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (°C)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	331.17	537		170	3389.77	6.39	1489.6	1122.59	493.31
2	298.61	329.9		36.49	3042.47	6.21	1126.64	908.551	336.43
3	32.56	329.9		36.49	3042.47	6.45	1126.64	99.06	36.68
4	298.61	537		32.87	3527.41	7.04	1391.96	1053.32	415.65
5	251.97	290.4		5.85	3040.93	7.02	881.24	766.10	222.05
6	17.72	412.1		14.14	3246.24	7.02	1125.48	57.52	19.94
7	28.92	290.4		5.85	3040.93	7.02	881.37	87.77	25.49
8	219.09	45.21	0.9229	0.1033	2349.42	7.31	126.29	514.73	27.67
9	8.19	72.26	0.9593	0.297	2545.11	7.24	327.28	20.84	2.68
10	16.08	134.4		1.246	2737.26	7.13	559.27	44.01	8.99
11	8.61	192.1		2.325	2846.24	7.12	677.42	24.51	5.83

Table– 2 . 3: Thermodynamic properties of the flows at 60% unit load of a 500 MW capacity coal–fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (ºC)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	250.73	537		170	3389.77	6.39	1487.45	849.92	372.95
2	229.66	322.7		28.09	3056.41	6.21	1125.32	701.94	258.44
3	21.07	322.7		28.09	3056.41	6.45	1125.32	64.40	23.71
4	229.66	537		25.32	3550.12	7.04	1390.46	815.32	319.33
5	194.69	292.1		4.53	3039.67	7.02	880.52	591.79	171.43
6	12.4	413.7		10.99	3206.20	7.02	1124.83	39.76	13.95
7	22.57	292.1		4.53	3039.67	7.02	880.52	68.61	19.87
8	171.98	45.21	0.941	0.1033	2442.62	7.31	125.42	420.08	21.57
9	4.79	72.26	0.9657	0.232	2532.16	7.24	326.25	12.13	1.56
10	11.66	135.7		0.966	2746.38	7.13	558.27	32.02	6.51
11	6.26	194		1.806	2857.67	7.12	676.32	17.89	4.23

2 . 3 Thermodynamic Modeling

A set of governing equations representing inside thermodynamics of the plant is used to carry out the present analysis. The respective set of thermodynamic equations includes energy balance, followed by an examining of the related exergy. These equations like mass continuity, energy balance, and exergy balance are presented by Cengel and Boles [90] under the steady state condition. As heat loss from turbines is reasonably low compared to the input thermal energy of the inlet steam, so it is neglected in calculation.

2 . 3 . 1 Mass Balance Equation

Rate of change of mass within the control volume is:

$$\frac{dm_{cv}}{dt} = \sum \dot{m}_{in} - \sum \dot{m}_{out} \tag{2.1}$$

Mass conservation under the steady state condition is:

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = 0 \tag{2.2}$$

2 . 3 . 2 Energy Balance Equation

The following equation based on the first law of thermodynamics, is used to present the rate of change in energy within the control volume and the energy in transition.

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \sum \dot{m}_{in}(h_{in} + \frac{V_{in}^2}{2} + gZ_{in}) - \sum \dot{m}_{out}(h_{out} + \frac{V_{out}^2}{2} + gZ_{out}) \tag{2.3}$$

On ignoring the change in kinetic energy and potential energy under the steady state condition, the equation for energy conservation is modelled as follows:

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out} h_{out} - \sum \dot{m}_{in} h_{in} \tag{2.4}$$

2 . 3 . 3 Exergy balance equation

On basis of the 2nd law of thermodynamics, the rate of change in exergy within the control volume is expressed as:

$$\frac{dX_{cv}}{dt} = \dot{X}_{heat} - \dot{W} + \sum \dot{m}_{in} \varepsilon_{in} - \sum \dot{m}_{out} \varepsilon_{out} - \dot{I} \tag{2.5}$$

The balance of exergy within the control volume under the steady state condition is reduced to

$$\dot{X}_{heat} - \dot{W} = \sum \dot{m}_{out}\varepsilon_{out} - \sum \dot{m}_{in}\varepsilon_{in} + \dot{I} \qquad (2.6)$$

Where $\dot{X}$ is the transfer rate of exergy, $\dot{I}$ is the destruction rate of exergy or irreversibility, and ε is specific exergy of the flow.

The net transfer of exergy ($\dot{X}_{heat}$) due to heat transfer at a temperature of T is presented as:

$$\dot{X}_{heat} = \sum (1 - \frac{T_0}{T})\dot{Q} \qquad (2.7)$$

2.3.4 Rate of irreversibility ($\dot{I}$)

There is a destruction in energy (a portion of energy unable to use it further) due to the irreversibility in the system, specifically in the turbines for present chapter, caused by the internal and external frictions experienced by the working fluid as it flows through the turbines.

$$\dot{I} = \dot{X}_{in} - \dot{X}_{out} - \dot{W}_{actuaal} \qquad (2.8)$$

2.3.5 Specific exergy of flow (ε)

The maximum rate of work accomplished by a steam before attaining to the equilibrium state with its surroundings is referred as the exergy of the stream. In this work, the exergy of steam per unit mass is calculated in each of the nodes in the control volume in order to determine irreversibility of the considered system. The specific exergy of any fluid stream is calculated [91] as:

$$\varepsilon = h - h_o - T_o(s - s_o) \qquad (2.9)$$

$$\dot{X} = \sum \dot{m}\varepsilon \qquad (2.10)$$

2.3.6 Shaft Power

Under the steady state condition and control volume mentioned in Fig. 2.1, the rate of shaft power output in the turbines is computed based on the energy balance. Thereafter, the exergy and exergy destruction rates are evaluated on basis of the exergy analysis. As per consequence; the shaft power output is calculated as:

$$\dot{W} = \dot{E}_{in} - \dot{E}_{out} \tag{2.11}$$

2.3.7 Energy Efficiency of Turbines

In practice, actual power output ($\dot{W}_{actual}$) is lower compare to the theoretical shaft power output ($\dot{W}$) due to heat losses into the ambient and incapability in converting of heat energy to work. Energy efficiency is a parameter to represent such losses. The energy efficiency of the turbines is, therefore, represented as follows.

$$\eta_I = \frac{\dot{W}_{actual}}{\dot{E}_{in}} \tag{2.12}$$

$$\dot{E} = \sum \dot{m} h \tag{2.13}$$

2.3.8 Exergy Efficiency of Turbines

The flow process involves both internal and external frictions, causes irreversibility. The exergy efficiency based on the 2nd law of thermodynamics is an indicating parameter of the irreversibility, presented as:

$$\eta_{II} = \frac{\dot{W}_{actual}}{\dot{X}_{in} - \dot{X}_{out}} \tag{2.14}$$

Where $\dot{W}_{actual}$ is actual power output and ($\dot{X}_{in} - \dot{X}_{out}$) is reversible power output.

2.3.9 Turbine Heat Rate (HR)

Heat rate (HR) is defined as the rate of heat supplied per unit power generation (kJ/kWh). The turbine heat rate (HR) is presented as:

$$HR = \frac{3600}{\eta_I} \tag{2.15}$$

2.4 Results and Discussion

In this section, a variation in the energy efficiency and the turbine heat rate as a function of unit load is presented in Fig. 2.2. It is mentioned here that an amount of external restraining torque that is applied to the turbine, is a definition of the unit load. It is observed here that value of the heat rate reduces with increase in the load, which indicates that the heat rate improves (discussed in details in Chapter-IV) as the unit load increases. When the plant is operating under 80% to 100% unit loads, the heat rate drops to its lowest possible level, and as a result, energy efficiency of the plant reaches to its highest possible level. The current investigation directs to a conclusion that the most cost-effective operating condition for a plant is when it operates at a greater or full unit load condition. In contrast, when a plant operates at 50% or 60% unit load, this leads to the most inefficient and costly working conditions.

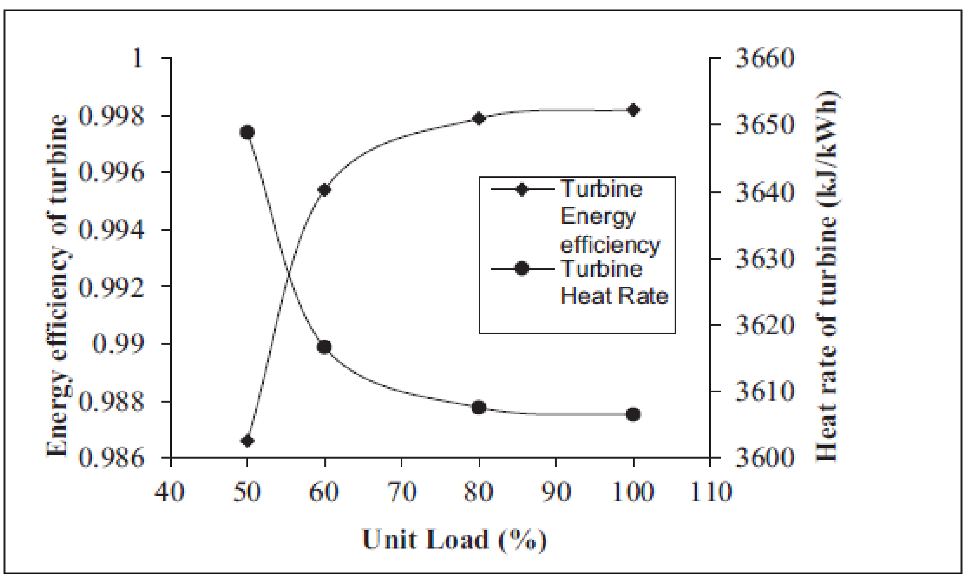

Figure 2.2: Variation of energy efficiency and heat rate of turbines with unit load

Contribution by an individual turbine to the total variation of cycle efficiency compare to its designed value is an important factor to be identified in order to an effective operation of the plant. In addition to this, an increase in the process irreversibility within a turbine results a direct loss of output power, which is a consequence to the loss of revenue. It is possible to minimize the loss by performing a routine maintenance of all turbines on the basis of exergy analysis. In order to determine the irreversibility in each of the turbines,both energy and exergy analysis are implemented in this chapter. Corresponding irreversibility for each of the turbinesis tabulatedintheTable–2.4 under a variety of the parameter pertaining to the unit load.Respective variation of the irreversibility for each individual turbine is presented in Fig.2.3. On basis of the irreversibility calculation, exergy efficiency for the individual turbine at varying unit load is also calculated, and shown in Fig. 2.4. It is noticed that the rate of irreversibility at LPT is noticeably higher than that of the HPT and IPT.While expanding, the steam with high thermal energy is flowing from HPT to IPT, and finally to LPT. Due to respective work output in the turbine, the quality of the steam changes from a condition of superheated steam to a wet steam. Now, a wet steam contains water droplets those increase irreversibilities in the flow, and hence, a sharp drop in the enthalpy at outlet of turbines is usually found. On the other hand, size of the LPT is comparably large, which also increases the external friction and hence, irreversibility. Due to the reason, irreversibility in the LPT is high. Thus, it is concluded here that a corrective measure is to be taken for the LPT in its maintenance and operational decisions. A necessary operational decision may be proper identifying and replacing the turbine blades in order to reduce the amount of exergy wasting in the LPT.In addition, after use of a certain number of years, the scale formed on the surfaces of turbine blades needs to be removed in order to improve the overall performance of the blades.

Table–2.4: The rate of irreversibility of the turbines at different unit load

Unit Load	Irreversibility in HPT (MW)	Irreversibility in IPT (MW)	Irreversibility in LPT (MW)
100%	3.36	4.52	15.72
80%	3.43	4.59	15.76
60%	3.52	4.65	15.89

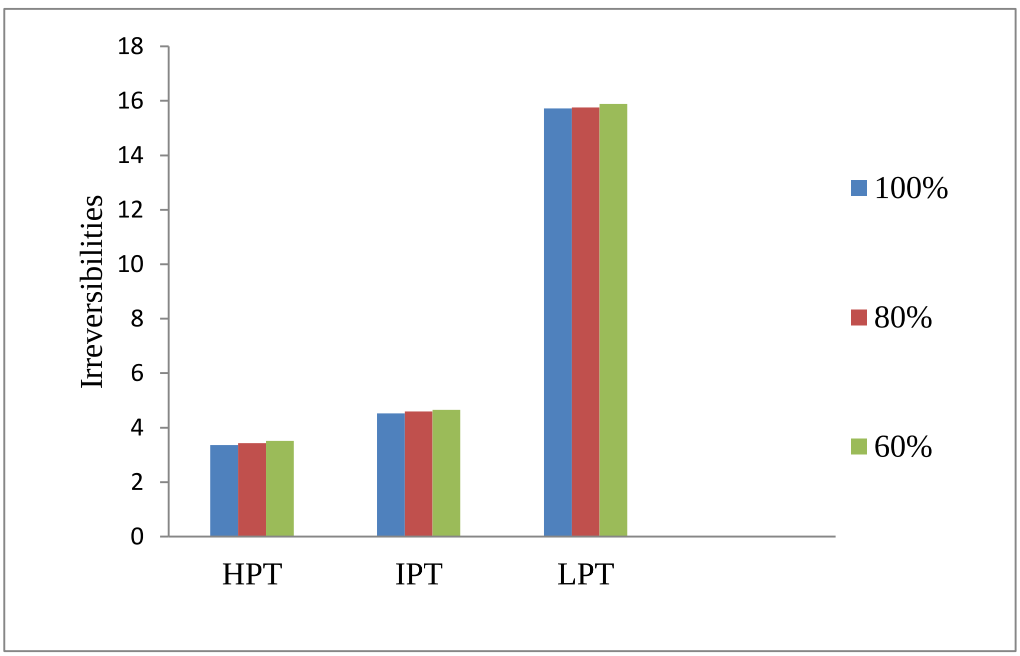

Figure 2.3: Rate of irreversibility of the turbines at different unit load

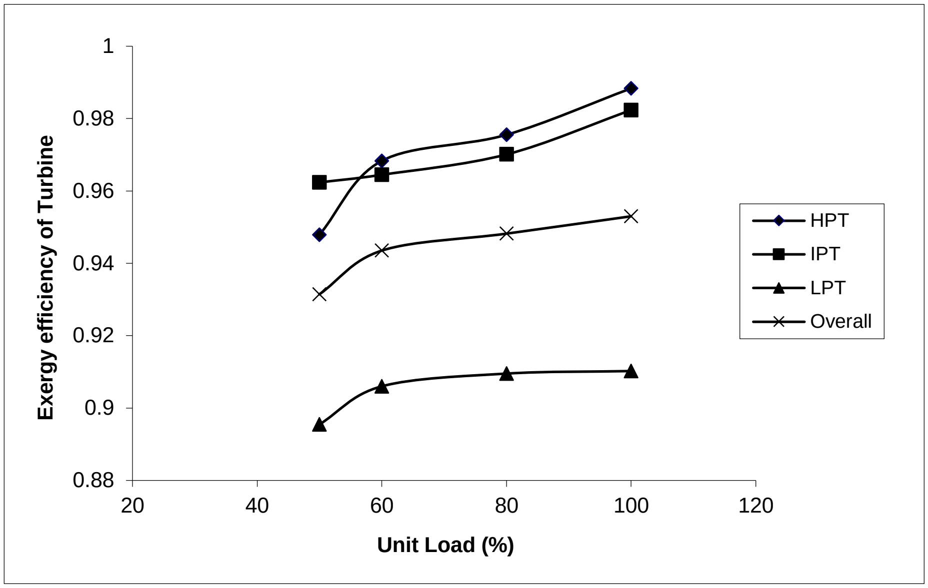

Figure 2.4: Variation of exergy efficiency with unit load

2.5 Conclusions

The energy and exergy analysis are implemented on the turbines of a 500 MW coal–based thermal power plant of the DVC, India. Only HPT, IPT, and LPT are considered within the control volume. A model using the thermodynamic laws is considered here for estimating performance of the turbines in the power plant at different unit load conditions (60%, 80% and 100%). It is seen that the irreversibility rate of the turbines increases with increase in the unit load. As predicted, the rate of irreversibility at LPT is more (22.15 MW at 100%–unit load) compare to than that of HPT and IPT. To minimize destruction in the exergy at the LPT, a proper replacement of blades might be a proper corrective method. It is observed that the exergy efficiency of the turbines is maximum at full unit load condition. It is found that maximum exergy efficiency of the HPT is 97.55% at 80% unit load whereas that of the LPT is 90.95% at 80% unit load and shows a minimum for all unit load conditions. Overall it is found that the heat rate is minimum for the higher loads (80% and 100%), which is an economical condition for operation. The subsequent reduction in the consumption of coal, generation of ash, emission of SO_2 and CO_2 due to improvement in heat rate is also predicted. Hence, out of the predictions, this work recommends operation of the plants at full load condition in order to obtain an effective plant efficiency.

2.6 References

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Chapter - III

Energy and Exergy Analysis

for

Individual Component of a 500 MW Coal based Thermal Power Plant

under

different unit load:

100%, 80% and 60%.

Chapter-III

Energy and Exergy Analysis for Individual Component of a 500 MWCoal-based Thermal Power Plant under different unit load: 100%, 80% and 60%.

3.1 Introduction

A crisis in available energy in the future has been described by many researchers and includes increasing electrical demand, reduction in fossil fuels used to lower green-house gas production, environmentally suitable power plants, and better efficiency. Steam power plants are especially vulnerable to these exigencies especially if they are older. It is observed that irreversibility ascends from the power plants is a form of environmental damage as it creates more emission of pollutants.

Thus methods to analyse performance is required to meet these challenges. It is already stated that, energy and exergy-based analysis of a power generating unit is most suitable and provides insight into the losses in various components of the unit.Exergy analysis helps in identifying the process irreversibility leading to the loss in useful work potential and thus pinpointing the areas where improvement can be sought. The exergy destruction rate or irreversibility rate can be considered as a benchmark by which losses in the plant and its individual components can be quantified and compared with the minimum possible irreversibility rate in that plant or its components [81]. Hence, exergy analysis along with energy analysis is also important for design, operation and maintenance of different equipment of a power plant.

In this chapter, the 1st law (energy) as well as the 2nd law (exergy) analysis of the turbine cycle (TC) and individual component of the system in a 500 MW coal-based power plant of DVC, India is considered. Accordingly, a thermodynamic model is developed for the analysis at various unit load conditions (100%, 80%, and 60%, etc.). The rate of exergy flow through control volume of turbines and system components is presented with the help of Grassman diagram. Then, the energy, i.e., the 1st law efficiency and HR are presented with different unit load. The irreversibility rate of each component within the turbine cycle are evaluated at various unit load conditions. Finally, the equipment with maximum irreversibility are pinpointed so that proper maintenance decisions can be taken.

3.2 Consideration of Physical Problem

In this study, a coal-fired steam power station of 500 MW of Damodar Valley Corporation, India is considered. The performance analysis of individual components of the station using energy and exergy analysis is conducted. Fig. 3.1 presents a diagrammatic representation of the turbine cycle of the thermal power plant. The relevant flows and properties data at different unit load, such as 100%, 80%, and 50% are summarized in Table 3.1, Table 3.2, and Table 3.3, respectively, along with the dead state condition. The cycle is linked to the generator and consists of a HPT, an IPT, and LPT. The data from the operations of the plant at different unit load are used to make an assessment of the flow of energy, the flow of exergy, and the flow of power. The HPT receives the high energy and exergy steam that is discharged from the boiler (flow-1). After being expanded in the HPT, the exhaust steam with lower energy and exergy (flow-2) is sent back to the boiler to be reheated. The IPT receives the superheated steam (flow-4), which has increased levels of both energy and exergy. Flow-3 depicts some of the steam that escapes from the HPT and flows into the HPH-2 for the purpose of feed water regeneration. After being expanded through the IPT, steam then flows to the LPT (flow-5) to undergo more expansion, which ultimately results in the generation of power. The LPT releases steam that has a dryness fraction of approximately 0.91 and the steam eventually enters a condenser (flow-8) at a pressure of approximately 10.47 kPa (abs). At a temperature that is lower than that of the ambient environment, the cold circulating water (CW) that comes from the cooling towers flows to the condenser (flow-a). The latent heat of condensation from the condenser is absorbed by the hot CW, which then transfers this heat to the cooling towers (flow-b). Flow-8, 9 and 10 depict steam bleeding from the LPT and flowing into the LPH-1, LPH-2, and LPH-3 for the purpose of regeneration. After heating feed water in the LPH-1, LPH-2 and LPH-3, steam returns back to the condenser through flow-14. The condensate flows out of the condenser at a low pressure and temperature (flow-23) and is pumped back to the deaerator by the CEP, which it does so by passing it through three different numbers of LPHs. The BFP draws suction from the deaerator (flow-20) and raises the pressure of the feed water (flow-21) to high value approximately 20.6 MPa so that it can flow into HPHs. Following that, the feed water is recirculated through flow-25 to the boiler, where it is used to produce steam, and the cycle thus continues. Because the attemperator regulates the temperature of the steam at the boiler outlet, the feed water from the BFP (flow-21) discharges back to the boiler. The present study takes place within a control volume, which is represented by a dashed line (---) in Fig.3.1. As a result, the flow of energy and exergy enters the control volume of the turbine cycle through flow-1 and 4, and the flow of energy and exergy exits the control volume through flow-2 and 25. By keeping the condenser back pressure at a low level, it is feasible to achieve a lower temperature while still losing heat to the surrounding environment. The temperature of the final feedwater (flow-25) that is being introduced into the boiler rises as it passes through the feed water heaters because heat is being transferred from the extracted steam at various intermediate stages of the turbines. This contributes to the addition of heat in the boiler, which is already operating at a relatively high temperature.

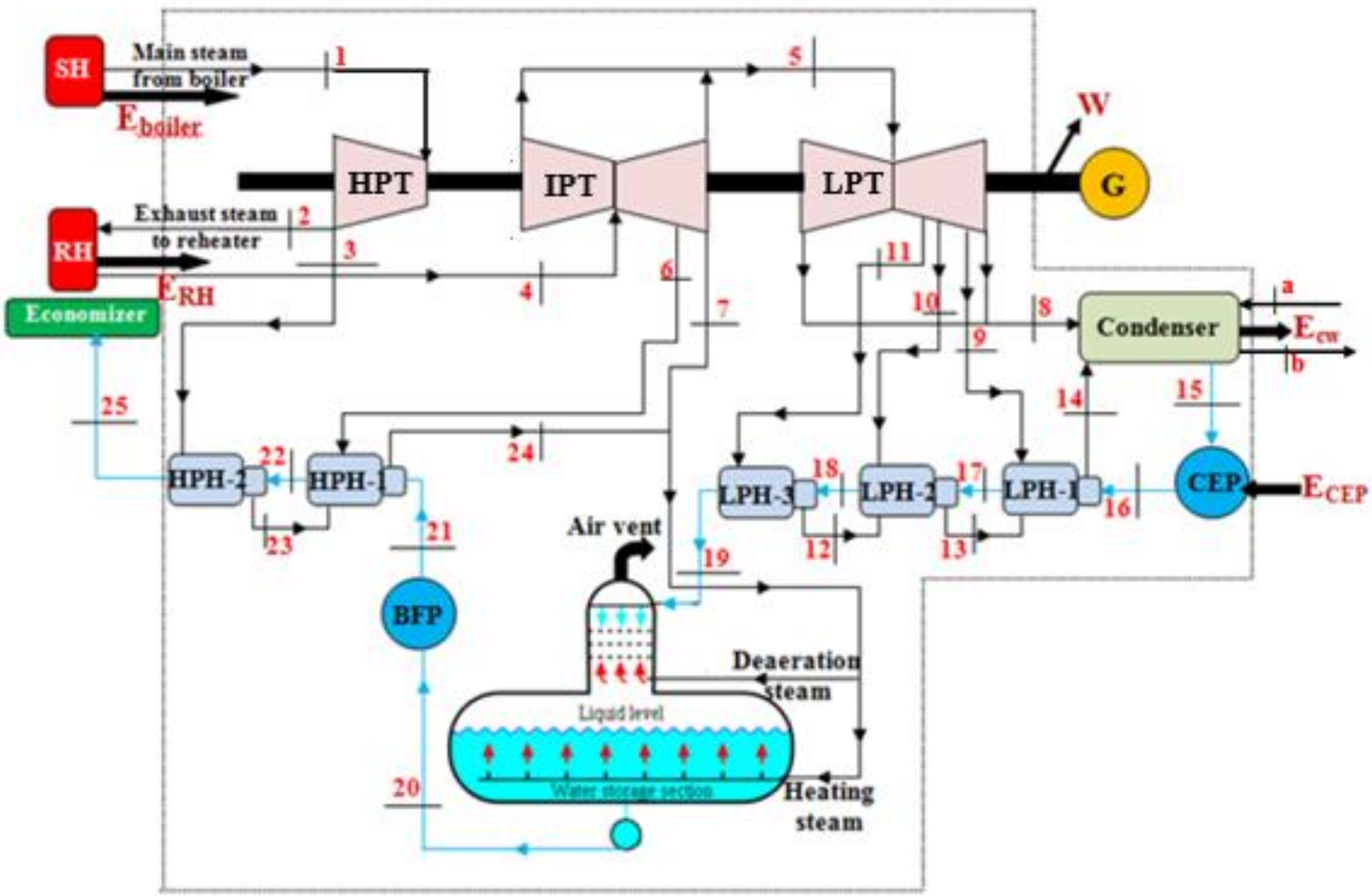

Figure - 3.1: Flow path of the turbine cycle of a 500 MW capacity coal-fired thermal power plant

Table-3.1: Thermodynamic properties of relevant flow at 100% unit load of a 500 MW coal-fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (ºC)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	415.47	537		172.21	3389.77	6.39	1489.6	1408.34	618.88
2	369.9	337.4		45.62	3046.97	6.45	1128.31	1127.07	417.36
3	45.57	337.4		45.62	3046.97	6.45	1128.31	138.84	51.41
4	369.9	537		41.06	3529.43	7.19	1392.6	1305.53	515.12
5	308.59	289.1		7.25	3035.95	7.24	882.4	936.87	272.3
6	23.38	411.1		17.55	3276.44	7.23	1126.6	76.6	26.34
7	37.93	289.1		7.25	3035.95	7.24	882.4	115.15	33.47
8	264.48	46.71	0.9124	0.1047	2351.48	7.48	127.43	624.08	33.82
9	11.67	73.35	0.9536	0.36	2741.35	7.46	328.15	29.73	3.83
10	21.03	133.6		1.55	2851.39	7.33	560.7	57.59	11.79
11	11.41	190.9		2.87	3040.93	7.29	678.98	32.51	7.75

12	11.41	190.5		2.73	468.6	7.32	37.26	5.35	0.43
13	32.44	73.4		0.36	307.27	0.99	17.58	9.97	0.57
14	44.11	51.1		0.13	213.93	0.72	4.7	9.44	0.21
15	334.85	46.1		0.1	194	0.65	5.63	63.73	1.85
16	334.85	46.3		22.27	195.8	0.65	7.43	64.33	2.44
17	334.85	68.6		22.27	288.96	0.94	14.17	94.93	4.66
18	334.85	106.9		22.27	449.79	1.38	43.88	147.77	14.42
19	334.85	126.5		22.27	532.82	1.6	61.35	175.05	20.16
20	415.47	160.8		6.51	679.06	1.95	103.29	282.13	42.91
21	415.47	164.3		207.51	706	1.96	127.25	293.32	52.87
22	415.47	202.3		207.51	870.85	2.32	184.82	355.76	75.5
23	45.57	207		17.96	884.07	2.4	174.2	40.29	7.94
24	68.95	169.1		7.75	715.27	2.03	115.66	49.32	797
25	415.47	253.7		207.51	1104.04	2.79	277.95	451.02	113.55

Table-3.2: Thermodynamic properties of relevant flow at 80% unit load of a 500 MW coal-fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (ºC)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	331.17	537		170	3389.77	6.39	1489.6	1122.59	493.31
2	298.61	329.9		36.49	3042.47	6.21	1126.64	908.551	336.43
3	32.56	329.9		36.49	3042.47	6.45	1126.64	99.06	36.68
4	298.61	537		32.87	3527.41	7.04	1391.96	1053.32	415.65
5	251.97	290.4		5.85	3040.93	7.02	881.24	766.10	222.05
6	17.72	412.1		14.14	3246.24	7.02	1125.48	57.52	19.94
7	28.92	290.4		5.85	3040.93	7.02	881.37	87.77	25.49
8	219.09	45.21	0.9229	0.1033	2349.42	7.31	126.29	514.73	27.67
9	8.19	72.26	0.9593	0.297	2545.11	7.24	327.28	20.84	2.68
10	16.08	134.4		1.246	2737.26	7.13	559.27	44.01	8.99
11	8.61	192.1		2.325	2846.24	7.12	677.42	24.51	5.83
12	8.61	106.2		2.21	444.34	7.18	35.33	3.83	0.30

13	24.69	69.5		0.27	290.788	0.86	16.64	7.18	0.41
14	32.88	67.6		0.11	282.42	0.69	6.20	9.29	0.20
15	267.45	46.1		0.1033	194	0.63	5.63	51.89	1.51
16	267.45	46.1		24.35	206.27	0.64	7.83	55.17	2.09
17	267.45	47.1		24.35	275.73	0.94	13.52	73.74	3.62
18	267.45	49.1		24.35	428.02	1.38	41.76	114.47	11.17
19	267.45	120.6		24.35	507.1	1.6	58.39	135.62	15.62
20	331.17	153.2		6.51	645.59	1.95	98.20	213.80	32.52
21	331.17	156.6		205.63	671.53	1.96	121.04	222.39	40.08
22	331.17	194.6		205.63	835.54	2.32	177.33	276.71	58.73
23	32.56	196		17.52	843.49	2.4	166.20	27.46	5.41
24	50.28	160.4		7.42	677.39	2.03	109.53	34.06	5.51
25	331.17	243.5		205.63	1055.62	2.79	265.76	349.59	88.01

Table-3.3: Thermodynamic properties of relevant flow at 60% unit load of a 500 MW coal-fired thermal power plant

Flow	Rate of Mass Flow (Kg/s)	Temperature (ºC)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/Kg)	Specific Entropy (s) (kJ/KgK)	Specific Exergy (ε) (kJ/Kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	250.73	537		170	3389.77	6.39	1487.45	849.92	372.95
2	229.66	322.7		28.09	3056.41	6.21	1125.32	701.94	258.44
3	21.07	322.7		28.09	3056.41	6.45	1125.32	64.40	23.71
4	229.66	537		25.32	3550.12	7.04	1390.46	815.32	319.33
5	194.69	292.1		4.53	3039.67	7.02	880.52	591.79	171.43
6	12.4	413.7		10.99	3206.20	7.02	1124.83	39.76	13.95
7	22.57	292.1		4.53	3039.67	7.02	880.52	68.61	19.87
8	171.98	45.21	0.941	0.1033	2442.62	7.31	125.42	420.08	21.57
9	4.79	72.26	0.9657	0.232	2532.16	7.24	326.25	12.13	1.56
10	11.66	135.7		0.966	2746.38	7.13	558.27	32.02	6.51
11	6.26	194		1.806	2857.67	7.12	676.32	17.89	4.23
12	6.26	98.9		2.21	414.22	6.69	32.94	2.59	0.21

13	17.92	64.2		0.27	268.61	0.79	15.37	4.81	0.28
14	22.71	62.4		0.11	261.08	0.64	5.73	5.93	0.13
15	207.06	46.1		0.1033	193.62	0.63	5.62	40.09	1.16
16	207.06	46.4		24.35	196.98	0.64	7.48	40.79	1.55
17	207.06	61.4		24.35	258.3	1.23	12.67	53.48	2.62
18	207.06	95.7		24.35	402.36	2.69	39.26	83.31	8.13
19	207.06	113.7		24.35	478.8	1.51	55.13	99.14	11.42
20	250.73	143.9		6.51	607.74	1.83	92.44	152.38	23.18
21	250.73	147.6		205.63	635.04	1.85	114.46	159.22	28.70
22	250.73	185.2		205.63	796.32	2.21	169.01	199.66	42.38
23	21.07	187.4		17.52	798.84	2.29	157.40	16.83	3.32
24	33.47	150.3		7.42	635.88	1.90	102.82	21.28	3.44
25	250.73	230.8		205.63	1000.86	2.64	251.97	250.95	63.18

Table-3.4: Thermodynamic properties of the dead state

	Temperature (°C)	Temperature (K)	Specific enthalpy (kJ/Kg)	Specific entropy (kJ/Kg-K)
Dead State	25	298	104.93	0.37

3.3 Thermodynamic Modeling

This analysis is carried out using a set of thermodynamic equations. The set of thermodynamic equations [78] largely consists of a study of energy, followed by a review of exergy. For the steady-state condition of flow, the equations for mass, continuity, energy balance, and finally exergy balance are already covered in Chapter-II. In addition to these equations for energy analysis, equations for irreversibility and second law efficiency of specific components, such as turbines, pumps, condensers, deaerators, and heaters, are taken into consideration for their performance analysis.

3.3.1 Equations of Energy analysis

The rate of net energy input to the cycle is $\dot{E}_{in}$. Where $\dot{E}$represents energy flow rate in MW. Suffixes RH, CEP and BFP represent re-heater, condensate extraction pump and boiler feedwater pump, respectively. Subsequently, the components of the right hand side of the eq. 16 are evaluated as:

$$\sum \dot{E}_{in} = \dot{E}_{boiler} + \dot{E}_{RH} + \dot{E}_{CEP} + \dot{E}_{BPF} \tag{3.1}$$

$$\dot{E}_{boiler} = \dot{m}_1 (h_1 - h_{25}) \tag{3.2}$$

$$\dot{E}_{RH} = \dot{m}_2 (h_4 - h_2) \tag{3.3}$$

$$\dot{E}_{BEP} = \dot{m}_{20} (h_{21} - h_{20}) \tag{3.4}$$

$$\dot{E}_{CEP} = \dot{m}_{15} (h_{16} - h_{15}) \tag{3.5}$$

The rate of net energy out from the cycle is formulated as:

$$\sum \dot{E}_{out} = \dot{E}_{CW} = \dot{m}_8 h_8 + \dot{m}_{14} h_{14} - \dot{m}_{15} h_{15} \tag{3.6}$$

3.3.2 The shaft power output

The shaft power output is obtained as:

$$\dot{W} = \sum \dot{E}_{in} - \sum \dot{E}_{out} \tag{3.7}$$

3.3.3 The first law (or energy) efficiency of the cycle

The first law (or energy) efficiency of the cycle is calculated as:

$$\eta_I = \frac{\dot{W}_{actual}}{\dot{E}_{in}} \tag{3.8}$$

3.3.4 Exergy analysis

The net exergy input rate to the cycle is calculated as:

$$\dot{X}_{in} = \dot{X}_{Boiler} + \dot{X}_{RH} + \dot{X}_{CEP} + \dot{X}_{BFP} \tag{3.9}$$

Where

$$\dot{X}_{Boiler} = \dot{m}_1(\varepsilon_1 - \varepsilon_{25}) \tag{3.10}$$

$$\dot{X}_{RH} = \dot{m}_2(\varepsilon_4 - \varepsilon_2) \tag{3.11}$$

$$\dot{X}_{CEP} = \dot{m}_{15}(\varepsilon_{16} - \varepsilon_{15}) \tag{3.12}$$

$$\dot{X}_{BFP} = \dot{m}_{20}(\varepsilon_{21} - \varepsilon_{20}) \tag{3.13}$$

3.3.5 The exergy output from the cycle

The exergy output from the cycle to the surroundings by the cooling water as a waste energy is derived as:

$$\sum \dot{X}_{out} = \dot{X}_{CW} = \dot{m}_8 h_8 + \dot{m}_{14}\varepsilon_{14} - \dot{m}_{15} h_{15} \tag{3.14}$$

3.3.6 Rate of irreversibility (I) or exergy destruction for the cycle

Rate of irreversibility ($\dot{I}$) or exergy destruction for the cycle is calculated as:

$$\dot{I} = \sum \dot{X}_{in} - \sum \dot{X}_{out} - \dot{W}_{actual} \tag{3.15}$$

3.3.7 The 2nd law (or exergy) efficiency for the cycle

Therefore, the 2nd law (or exergy) efficiency for the cycle is determined as:

$$\eta_{II} = \frac{\dot{W}_{actual}}{\sum \dot{X}_{in} - \sum \dot{X}_{out}} \tag{3.16}$$

3.3.8 Rate of irreversibility in individual components

Each component in the turbine cycle is considered in a separate control volume at steady state operation to evaluate the rate of irreversibility.

The equations to calculate the rate of irreversibility of different components in the TC are given below [78].

$$\dot{I}_{HPT} = \dot{X}_1 - \dot{X}_2 - \dot{X}_3 + \dot{W}_{HPT} \tag{3.17}$$

$$\dot{I}_{IPT} = \dot{X}_4 - \dot{X}_5 - \dot{X}_6 - \dot{X}_7 + \dot{W}_{IPT} \tag{3.18}$$

$$\dot{I}_{LPT} = \dot{X}_5 - \dot{X}_8 - \dot{X}_9 - \dot{X}_{10} - \dot{X}_{11} + \dot{W}_{LPT} \tag{3.19}$$

$$\dot{I}_{Turbines} = \dot{I}_{HPT} + \dot{I}_{IPT} + \dot{I}_{LPT} \tag{3.20}$$

$$\dot{\mathrm{I}}_{\mathrm{Condenser}} = \dot{\mathrm{X}}_8 + \dot{\mathrm{X}}_{14} - \dot{\mathrm{X}}_{15} \tag{3.21}$$

$$\dot{\mathrm{I}}_{\mathrm{BFP}} = \dot{\mathrm{X}}_{20} - \dot{\mathrm{X}}_{21} + \dot{\mathrm{W}}_{\mathrm{BFP}} \tag{3.22}$$

$$\dot{\mathrm{I}}_{\mathrm{LPH}} = \dot{\mathrm{X}}_{14} - \dot{\mathrm{X}}_{19} \tag{3.23}$$

$$\dot{\mathrm{I}}_{\mathrm{HPH}} = \dot{\mathrm{X}}_{21} - \dot{\mathrm{X}}_{25} \tag{3.24}$$

$$\dot{\mathrm{I}}_{\mathrm{Deaerator}} = \dot{\mathrm{X}}_7 + \dot{\mathrm{X}}_{19} + \dot{\mathrm{X}}_{24} - \dot{\mathrm{X}}_{20} \tag{3.25}$$

3.3.9 Exergy efficiency of individual Components

The equations to calculate 2nd law efficiency of different components in the turbine cycle are given here

$$\eta_{II.HPT} = 1 - \frac{\dot{I}_{HPT}}{\dot{X}_1 - \dot{X}_2 - \dot{X}_3} \tag{3.26}$$

$$\eta_{II.IPT} = 1 - \frac{\dot{I}_{IPT}}{\dot{X}_4 - \dot{X}_5 - \dot{X}_6 - \dot{X}_7} \tag{3.27}$$

$$\eta_{II.LPT} = 1 - \frac{\dot{I}_{LPT}}{\dot{X}_5 - \dot{X}_8 - \dot{X}_9 - \dot{X}_{10} - \dot{X}_{11}} \tag{3.28}$$

$$\eta_{II.Condenser} = 1 - \frac{\dot{I}_{Condenser}}{\dot{X}_8 + \dot{X}_{14} - \dot{X}_{15}} \tag{3.29}$$

$$\eta_{II.BFP} = 1 - \frac{\dot{I}_{BFP}}{\dot{W}_{BFP}} \tag{3.30}$$

$$\eta_{II.LPH} = 1 - \frac{\dot{I}_{LPH}}{\dot{X}_{14}} \tag{3.31}$$

$$\eta_{II.HPH} = 1 - \frac{\dot{I}_{HPH}}{\dot{X}_{21}} \tag{3.32}$$

$$\eta_{II.Deaerator} = 1 - \frac{\dot{I}_{Deaerator}}{\dot{X}_7 + \dot{X}_{19} + \dot{X}_{24} - \dot{X}_{20}} \tag{3.33}$$

All related calculations in the thesis is performed on Excel platform.

3.4 Results and discussion

Analysing the turbine cycle depicted in Fig.3.1 using a thermodynamic model, such as the model described in Section 3.3, with a reference temperature of 298 K and a reference pressure of 101.3 kPa are the two parameters that are taken into consideration. In Fig.3.1, the various states of the steam are each represented by a flow number from 1 to 25 for the purpose of analysing the performance of the turbine cycle. During the analysis, one should determine which areas of the system have the highest level of irreversibility and identify which parts of the system have the highest level of performance. In light of this, the rate of energy flow as well as exergy flow is analysed at each state with reference to the data collected from DVC handbook (Table 3.1, 3.2 and 3.3). Following that, the rate of inflow and outflow of energy, exergy, and irreversibility of the cycle are evaluated at 100%, 80%, 60%, and 50% of the rated unit load, as shown in Table-3.5. According to Table-3.5, it was discovered that the rate of net energy input is nearly 2.21 times that of the power output in the cycle (1,123.54 MW at 507 MW load) when the unit load is at 100%, but it is approximately 2.4 times when the unit load is at 50%. This finding is made after comparing the rate of net energy input with the power output of the cycle. The investigation also revealed that a sizeable portion of the net energy input is expelled from the cycle in the condenser (about 54.84% at 100% unit load), and the remaining amount is the work output rate from the turbine cycle. When compared to the power output of the turbine cycle, the amount of electrical power produced at the generator terminal is approximately 2% lower. This could be because of the losses that are related with the power consumption of the generator as well as the auxiliary in the cycle.

Table 3.5:Energy and exergy analysis of the turbine cycle at various unit load

Unit Load (%)	**Energy in,** $\sum \dot{E}_{in}$ **(MW)**	**Exergy in,** $\sum \dot{X}_{in}$ **(MW)**	**Energy rejection to condenser,** $\dot{E}_{out}$ **(MW)**	**Exergy rejection to condenser,** $\dot{X}_{out}$ **(MW)**	**Work done,** $\dot{W}$ **(MW)**	**Exergy destruction,** $\dot{I}$ **(MW)**
100	1123.54	607.36	616.22	35.46	507.32	64.58
80	924.031	499.20	509.19	33.81	414.841	50.549
60	735.825	386.177	416.109	27.19	319.716	39.271
50	614.051	310.53	359.08	24.00	254.971	31.559

In addition, the 1st law and 2nd law efficiencies of the cycle are evaluated under a variety of unit load conditions and are depicted in Fig.3.2. It has been observed that both the efficiencies improve when there is a greater increase in the unit load. However, there is a noticeable gap between the two efficiencies, and this gap widens as the load gets higher. According to Table-3.6, it has been determined that the exergy rejection at a specific unit load is lower than the energy rejection to the condenser. This finding was made after it was observed that the energy rejection to the condenser is higher. Because of this, the cycle efficiency according to the first law is lower at a given unit load, while its efficiency according to the second law is higher. It has also been discovered that both the energy efficiency and the exergy efficiency are rather low for a low unit load (for example, 50%).

This is because, under conditions of low unit load, both the energy and exergy rejection rates increase to a greater extent. Usually, the design of any turbine blades (rotary and stationary blades) is performed on the full load condition. A full load mass flow rate results higher velocity in the flow passages, which leads to a high Reynolds number flow. On other side, running of a power plant on reduced load leads to a low Reynolds number flow causes flow separation, vortex formation etc. those are responsible in generating entropy due to high friction and other related losses. This is one of the major reasons for getting low thermal efficiency at low load conditions.

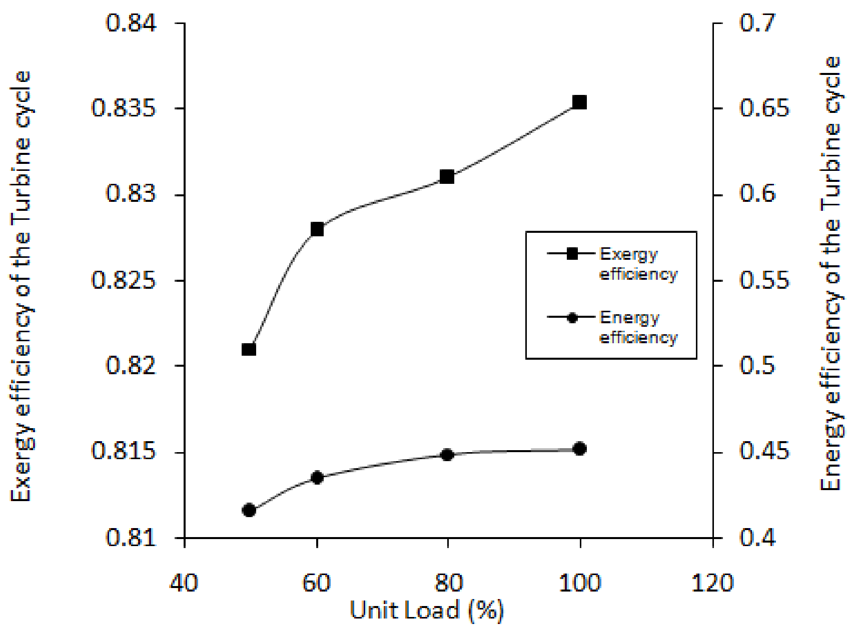

Figure- 3.2: The 1st law and 2nd law efficiencies of the turbine cycle with unit load

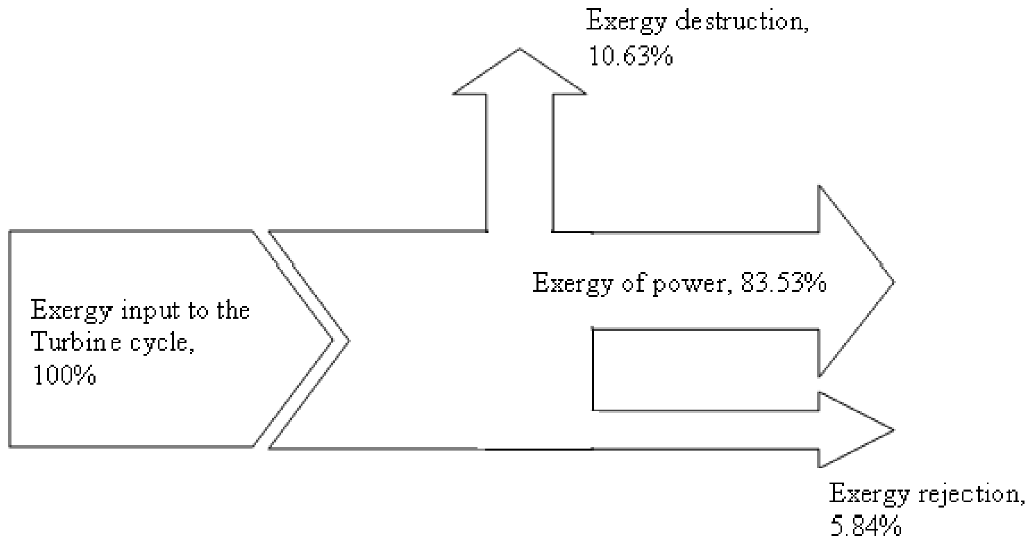

Figure - 3.3: Exergy flow through the turbine cycle at full load by Grassman diagram

The Grassman diagram [80] in Fig.3.3 depicts the usual exergy flow rates for the turbine cycle when it is operating at 100% load. It can be shown that the rates of power output, irreversibility, and exergy rejection from the turbine cycle are around 0.8353, 0.1063, and 0.0584 times that of the net exergy input rate, respectively (607.36 MW at 507 MWe unit load). In Table 3.6, the exergy components are represented as a percentage of the cycle for each of the different unit load conditions. It has been observed that the exergy destruction rises with decreasing unit load.

Table 3.6: Exergy flow in the turbine cycle at various unit load

Unit load (%)	Exergy of power (%)	Exergy rejection (%)	Exergy destruction (%)
100	83.53	10.63	5.84
80	83.12	10.13	6.75
60	82.78	10.16	7.06
50	81.99	10.13	7.88

Table 3.7: The irreversibility rate in the individual component at different unit load conditions

Unit Load(%)	HPT (MW)	IPT (MW)	LPT (MW)	CEP (MW)	LPH-1 (MW)	LPH-2 (MW)	LPH-3 (MW)	HPH-1 (MW)	HPH-2 (MW)	Deaerator (MW)	BFP (MW)
100%	3.36	4.52	15.72	0.37	0.97	2.19	0.57	2.01	5.87	2.24	3.96
80%	3.43	4.59	15.76	0.29	0.93	2.08	0.57	2	5.83	2.24	3.92
60%	3.52	4.65	15.89	0.29	0.92	2.08	0.55	2	5.82	2.21	3.91
50%	3.57	4.66	15.98	0.27	0.93	2.09	0.57	2.01	5.84	2.23	3.92

The contribution of the separate components to total variation in cycle efficiency, compared to its design value, is identified in various irreversibility sources of the components in the cycle seen in the exergy analysis. Increase in irreversibility in a component is considered directly as the loss of power and therefore, the loss of revenue. However, this loss of revenue of a plant is minimized by proper maintenance of components through the exergy analysis. The irreversibility for the components of the TC is presented in Fig. 3.4 at 500 MW of unit load. The irreversibility rate for the LPH-2 is more compared to the LPH-1 and LPH-3. This is because the steam flow rate to LPH 2 is nearly two times than that of the LPH-1 and LPH-3 (flow-15, flow-16 and flow-17 shown in Table-3.1). On the other hand, the 2nd law analysis is carried out for HPH-1 and HPH-2, and it reveals that consumption of exergy in HPH-2 is noticeably increased compared to its design value. Therefore, some corrective actions are required in both the operation decision and maintenance decision. To minimize the exergy destruction in the HPH-2, the operation decision includes properly adjusting drip level and venting of air. Sometimes, after a number of years of service, scale forms over the heating surfaces which cause deterioration of the heater performance. Hence, proper maintenance decisions such as replacement or retubing improves the performance of the heaters. The decisions during operation and maintenance based on the 2^{nd}law analysis of systems are more effective.

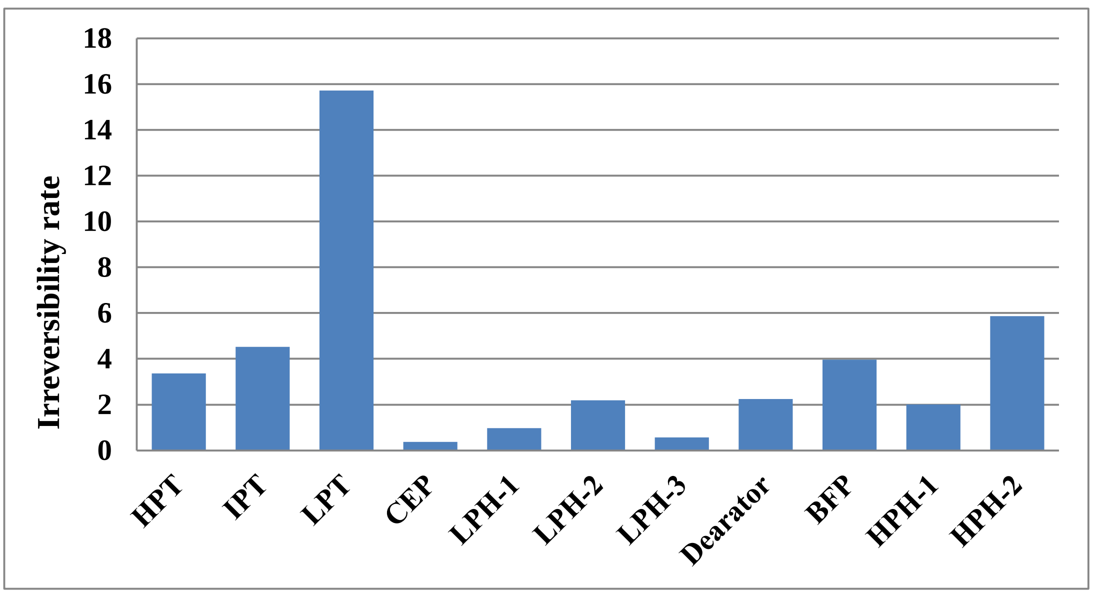

Figure - 3.4: Irreversibility rate (MW) in individual components at 100% unit load

3.5 Conclusions

In the present work, the 1st law and 2nd law analysis of the turbine cycle of a coal-based steam power plant in India is analyzed. A thermodynamic model is considered to estimate the performance of the cycle as well as of the system components. A detailed study for losses of energy as well as exergy in the cycle and its components is presented at full unit load, 80%, 60% and 50% unit load. Hence, the 1st law and 2nd law efficiencies, and HR for the cycle are presented as a function of unit load. It is observed that the rate of energy input is about 2.21 times of the power output from the turbine cycle for the 100% unit load whereas it is about 2.4 times that at a unit load of 50%. It is also noticed that both the 1stlaw and 2nd law efficiency are comparatively low at a low unit load. The authors conclude it is more economical to operate the plant at a higher unit load. In this paper we present the rate of exergy flow through the turbine cycle by the Grassman diagram. The irreversibility rate is more in the cycle at low unit load. The irreversibility rate for the individual component of thecycle is also presented at 500 MW of unit load. It is found that the irreversibility rate for the LPTs and HPH-2 is more compared to the other system components of the turbine cycle. To minimize the exergy destruction in the HPH-2, proper adjustment of drip level, venting of air, and replacement or re-tubing of the heaters would be advantageous. Since turbine HR decreases and unit load increases, it follows that HR minimizes at full load resulting in a more economical operating mode.

3.6 References

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Chapter - IV

Analysis on Emissions and Ash Generation of a 500 MW Coal based Thermal Power Plant at different unit load

Chapter-IV

Analysis on emissions and ash generation of a 500 MW Coal based Thermal Power Plant at different unit load

4.1 Introduction

Reduction of harmful pollutants in the environment and ensuring availability to the global energy demand lead to parameter in order to balancing them, which is a challenge worldwide [82]. As per available literature, the air pollution is a severe problem to us. Although particulate matters (PMs), CO_2, SO_x, NO_x, and other types of pollutants in air are released by the different industrial facilities, construction and demolition activities etc. [83, 84], but the major pollutants mainly carbon dioxide (CO_2), sulfur dioxide (SO_2), nitrogen oxides (NO_x), carbon monoxide (CO), volatile organic compounds (VOCs) etc. are released by coal based thermal power plants [83-85]. Due to their tiny size, these particles frequently enter into the human organs where they affect the organs gradually.Their adverse effects lead human body in many chronic diseases including respiratory problems, weakened immune systems, cardiovascular diseases, cancer etc.- all of these lead human to early death [87-90]. According to a recent study on health detrimental issues affects by the air pollution, 0.76 million overall deaths is recorded due to the ambient air pollution in year of 2020 [82-86]. In addition to the above, carbon dioxide as a greenhouse gas is a major cause to the global warming. Despite of knowing to the fact that a coal-fired power plant emit a variety of dangerous pollutants continuously, till we rely on the electricity which is generated out of present existing power plants only because of electricity out of the coal fired power plants is less expensive than that of the other sources.

In India too, the coal based power plants are a major source of the environmental pollution. According to the reports [94], the power sectors emit 51% SO_2, 43% carbon dioxide (CO_2), 20% oxides of nitrogen (NO_x), and 7% PM to the environment. Reduction of such emissions is a major challenge, now-a-days. In this context, the government of Indian has introduced a necessary step by announcing the "Environment (Protection) Amendment

Rules, 2015" in order to reduce pollution in environment by variety of sources including the thermal power plants.

A systematic understanding of the emissions out of a coal based thermal power plant is essential in order to identify the sources and hence, its necessary control. In this chapter, an effort is included accordingly to analyse emissions and generation of ashes out of a coal fired power plant, consequent to minimize their effect in environmental pollution.

4.2 Mathematical and Chemical Modeling

In Chapter–II, the conservations of mass, energy, and exergy are explained in details. The related equations for the rate of irreversibility, specific exergy flow, shaft power, energy efficiency, exergy efficiency, turbine heat rate etc. as discussed in Chapter-II, are utilized here for analysis also.

In a coal fired thermal power plant, coal is the primary fuel for the combustion and generating to work as heat source. The necessary heat input [78] is calculated as

Heat input = Rate of coal consumption × GCV (4.1)

where GCV is gross calorific value of the coal. The gross calorific value (GCV) is calculated by the Dulong's formula [72] as

$$\mathrm{GCV(in\ kJ)} = 338.0672\times \mathrm{C} + 1443.184\times \mathrm{H} - 180.435\times \mathrm{O} + 92.8848\times \mathrm{S} \qquad (4.2)$$

The chemical constituents of a coal mentioned in the eq. (50) are given in Table-4.1in weight % based on the ultimate analysis of the coal [72] used in standard power stations.

Then corresponding heat rate [72] and heat rate improvement [95] is calculated as

Heat rate (HR) = Heat input/Generator power (4.3)

Heat rate improvement in increase of unit load by ΔX %

= HR at X% unit load - HR at (X+ΔX) % unit load (4.4)

Therefore, the coal saving based on heat rate improvement is calculated as

Coal saving = HR improvement × Generator power /GCV (4.5)

Corresponding to above coal saving, related reduction in CO_2 and SO_2 emissions and ash generation are evaluated as:

Based on chemical balance of product and reactants, it is found that 1kg carbon generates 44/12 kg of CO_2 = 3.66 kg. Hence reduction in emission of CO_2 is evaluated as:

CO_2 emission reduction = 3.66 × Carbon percentage in coal × Coal saving (4.6)

Further, it is well found that 1kg sulphur generates 64/32 kg of SO_2 = 2.0 kg. Therefore, reduction in emission of SO_2 is calculated as:

SO_2 emission reduction= 2.0 ×Sulphur percentage in coal × Coal saving (4.7)

Corresponding to the coal saving, the reduction in ash generation is calculated as:

Ash generation reduction = Coal saving × Ash percentage in coal (4.8)

Table-4.1: Ultimate analysis of a standard coal [72]

Chemical constituents	Weight (%)
Carbon (C)	42.82
Nitrogen (N)	0.82
Hydrogen (H)	2.65
Sulphur (S)	0.34
Oxygen (O)	7.35
Moisture (M)	6.2
Ash (A)	39.82

4.3 Results and Discussion

Based on the model presented in the section 4.2, corresponding reduction in generation of ash and emission of greenhouse gas with respect to improving in the heat rate is presented in this section. Table 4.2 shows reduction in consumption of coal, emission of CO_2, SO_2 and generation of ash by improving the heat rate through stepwise increase in the unit load (row-1, 2 of the Table 4.2). As unit load increases, consumption of fuel is reduced significantly (row-3 of the Table 4.2). Consequent reduction in CO_2, SO_2, and generation of ash is evaluated and presented in Table 4.2 (row-4, 5, 6). On comparison of the results in the table, a conclusion is drawn that generation rate of pollutants and ash decreases to a significant value with increase in unit load. As found, at the 100% of the unit load, reduction rate in emission of CO_2, SO_2, and ash formation is maximum.

Table- 4.2: Improvement in heat rate (HR) and corresponding reduction in emissions with increase in unit load

Sl. No	Parameters	Values		
1		Increase in unit load (50% to 60%)	Increase in unit load (60% to 80%)	Increase in unit load (80% to 100%)
2	HR improvement in kJ/kWh	17.50	48.33	106.01
3	Reduction of coal consumption in kg/day	267.84	762.48	1230.72
4	Reduction in CO_2 emission in kg/day	419.86	1195.25	1929.25
5	Reduction in SO_2 emission in kg/day	1.71	4.88	7.88
6	Reduction in ash generation in kg/day	107.14	304.99	492.29

4.4 Conclusions

The heat rate improves at higher load (80% and 100%), which is an economical condition of the operation. The subsequent reductions in consumption of the coal, ash generation, and emission of SO_2 and CO_2 due to the improvement in heat rate are predicted. As seen, the generation rate of pollutant gases and ash decreases to a significant value with increase in the unit load. At 100% unit load, the reduction of coal consumption is 1230.72 kg per day. Accordingly, under this full load condition, reduction rate for $CO_2$is 1929.25 kg/day, for SO_2 is 7.88 kg/day, and for generation of ash is 492.29 kg/day – enough to reduce environmental pollution. On basis of the results, it is concluded that it would be more economical when any plant operates at its rated capacity of unit load than part load.

4.5 References

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Chapter - V

Integration of Solar Energy as a Part of the Existing 500 MW Coal-based Thermal Power Plant

Chapter V

Integration of Solar Energy as a Part of the Existing 500 MW Coal−based Thermal Power Plant

5.1. Introduction

Continuous increase in population and growing standard of living are the main causes in rise of the energy demand in term of electricity. In this regard, Mittal [96] reported that the demand of electricity is increasing by about 6-7% per year. On the other hand, the ecology and human health are greatly affected by utilization of different fossil fuels as energy sources. In addition to this, there is limited stock of fossil fuels in the earth and they are depleted day by day. The best way to solve this problem is to use free energy from the sun. India receives a large amount of free energy from the Sun throughout the year. The Ministry of New and Renewable Energy (MNRE) says that the India has a lot of potential to use the solar energy. As per MNRE, India is receiving about 5,000 trillion kWh of solar energy per year, which is about 4 to 7 kWh per m^2 per day [97]. Accordingly, the use of renewable energy is increasing day by day to meet our energy demands. In today's world, the solar energy is one of the most promising renewable energy sources. There are various collectors to receive and utilize of the solar energy. Parabolic solar trough is a most useful type of the solar collectors, which can be integrated with conventional coal-based thermal power plants in order to reduce consumption of coals. The reduction in the coal consumption reduces emissions in the environment, and balances demand of electricity. In India, there are several number of coal based thermal power plants those can be modified slightly to make it into the solar aided coal fired power generation (SACFPG) to increase performance and reduces harmful emissions [98-102].

As available, there are few studies [47-68] on the solar hybridization, which mainly involves installing of the parabolic solar troughs in place of low pressure heaters (LPHs) for similar to the regeneration to improve performance and reduce emissions of a traditional coal-fired thermal power plant. So, in this chapter, an effort is furnished to compare the energetic and exergetic efficiency as well as coal consumption, ash generation, SO_2 and CO_2 emissions, and heat rate improvement of a 500 MW coal-fired power plant before and after the solar hybridization of the plant by deploying the parabolic solar troughs at the place of the LPHs for similar to the regeneration.

5.2. Consideration of Physical Problem

A 500 MW coal-fired steam power plant of DVC, India is taken under consideration in this study. Fig.5.1 shows a typical integration and installation of the parabolic solar troughs in place of the low-pressure feed water heaters. The parabolic troughs are made of parabolic shaped reflective mirror, a tubular receiver, and supporting structures. The reflective mirror reflects all incident radiation of the sun towards the tubular receiver which contains a heat transfer fluid (HTF) to carry heat. In order to receive the maximum reflection, the tubular receiver is placed at focus of the parabola and the flowing fluid (HTF) inside of the receiver is getting heated.

On integration of the solar energy in place of feed water heaters, the parabolic solar troughs heat the feed water before it passes through economizer and enters the boiler drum. Due to this arrangement, there is no need to bleed steams from the LP turbine. Accordingly, both mass flow rate of steam passes through the LP turbine and amount of work produced by it increase. Coal consumption is also reduced, leading to less ash generations well as less harmful emissions into the air.

As shown in Fig. 5.2, the parabolic solar troughs are arranged in series in a single row, and several rows are considered on the basis of present requirement. The series arrangement of the parabolic trough collectors increases temperature of heat transfer fluid whereas the rows arrangement increases the mass flow rate of the HTF. The solar radiation reflected by the parabolic mirror of the parabolic trough collectors, and heated the heat transfer fluid flowing through the tubular receiver.These hot heat transfer fluid is then used to heat the boiler feed water through a counter flow heat exchanger which is placed between condenser and deaerator. In present analysis, Therminol (VP-1) is considered as the heat transfer fluid.

Hence, in this chapter, a detailed analysis is performed for an existing coal based 500 MW thermal power plant by predicting its performance, overall thermal efficiency, coal consumption rate before and after integrating the parabolic solar trough collectors at the place of LPHs in order to heat the feed water. In this regard, the necessary thermodynamic data used in the analysis are shown in the Table 5.1.

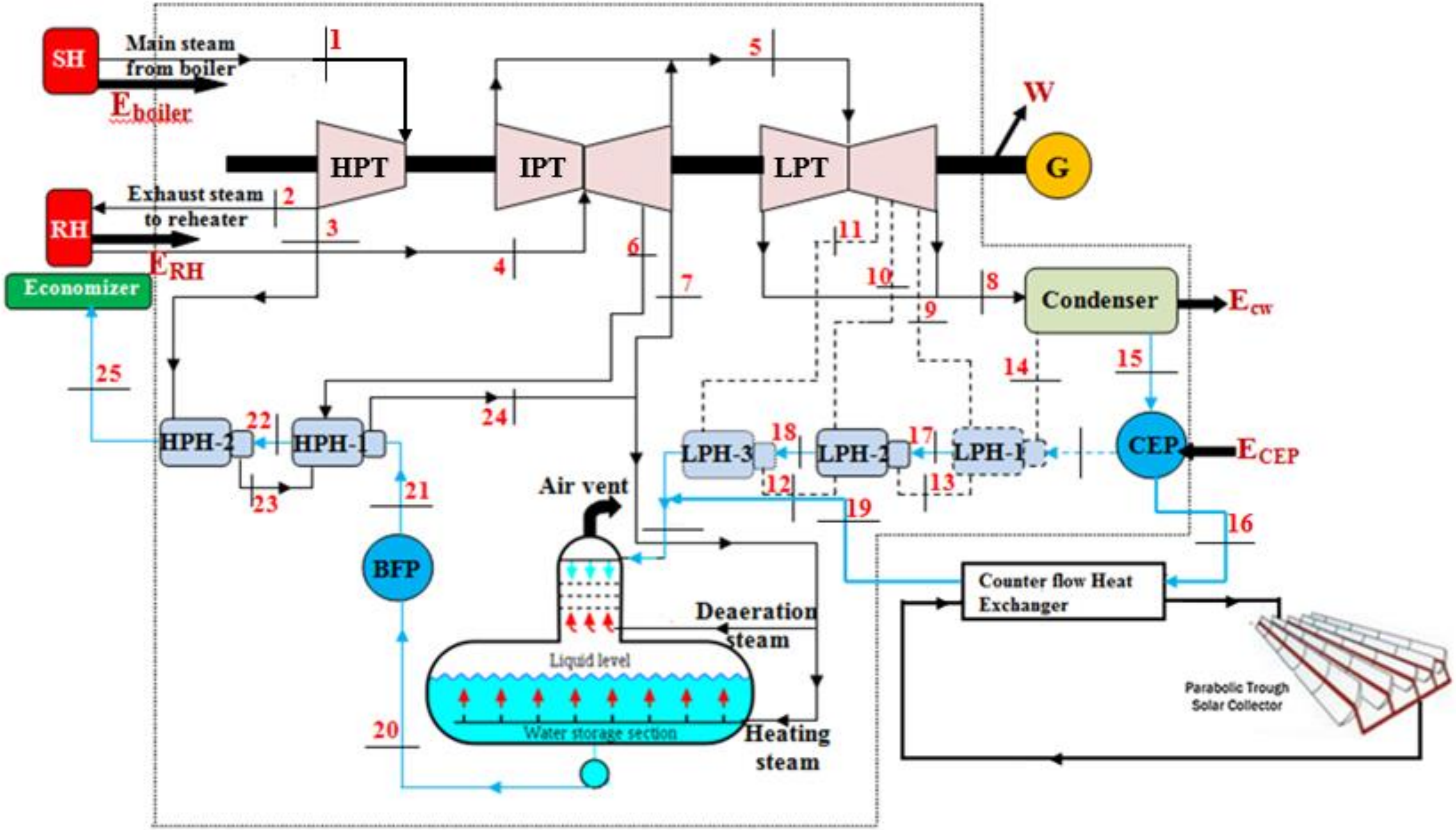

Figure 5.1: Plant cycle of a 500 MW coal based thermal power plant integrated with parabolic solar troughs in place of LP heaters

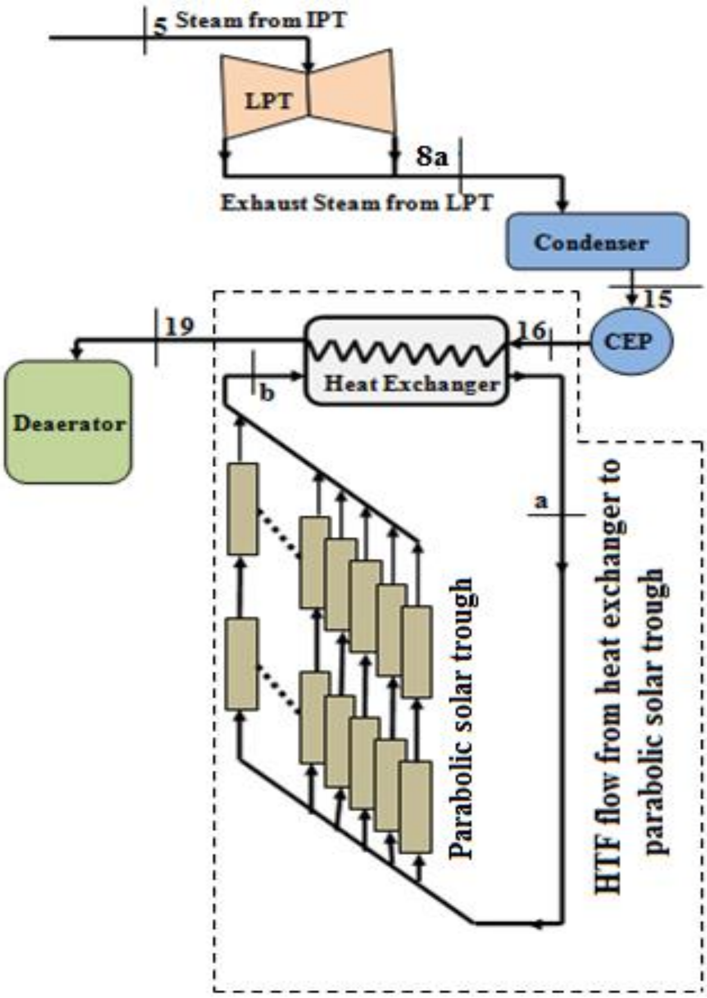

Figure 5.2: Schematic arrangement of parabolic solar troughs in place of LPHs

5.3 Description of the parabolic solar trough of LS-3 type

These are different types of parabolic trough collectors. Types-LS are mostly in use for applications. The LS-3 parabolic trough collector s chosen for this study, as shown in Fig. 5.3, because of the fact that these collectors are bigger in diameter than the types of LS-1 and LS-2, widely used, efficient, and available in market. The LS-3 troughs are used in the power plant to replace the LP heaters. The axis is assumed horizontal, and runs north to south. It helps to track the sun radiation from east to west. Further with a special coating, the stainless steel receiver tube is able to absorb more solar radiation. The conduction heat loss is prevented with use of a 66-mm glass tube surrounding the receiver tube. The thermo-physical properties of a LS-3 type parabolic trough are presented in Table 5.2.

Table 5.1: Thermodynamic properties of different flows in the turbine cycle at full load condition

Flow	Rate of Mass Flow (kg/s)	Temperature (°C)	Dryness Fraction (x)	Pressure (Bar)	Specific Enthalpy (h) (kJ/kg)	Specific Entropy (s) (kJ/kgK)	Specific Exergy (ε) (kJ/kg)	Energy Flow rate (E) (MW)	Exergy Flow rate (X) (MW)
1	415.47	537		172.21	3389.77	6.39	1489.6	1408.34	618.88
2	369.9	337.4		45.62	3046.97	6.45	1128.31	1127.07	417.36
3	45.57	337.4		45.62	3046.97	6.45	1128.31	138.84	51.41
4	369.9	537		41.06	3529.43	7.19	1392.6	1305.53	515.12
5	308.59	289.1		7.25	3035.95	7.24	882.4	936.87	272.3
6	23.38	411.1		17.55	3276.44	7.23	1126.6	76.6	26.34
7	37.93	289.1		7.25	335.95	7.24	882.4	115.15	33.47
8a	334.85	46.71	0.9124	0.1047	2351.48	7.48	127.43	624.08	33.82
15	334.85	46.1		0.1	194	0.65	5.63	63.73	1.85
16	334.85	46.3		22.27	195.8	0.65	7.43	64.33	2.44
19	415.47	126.5		22.27	532.82	1.6	61.35	175.05	20.16
20	415.47	160.8		6.51	679.06	1.95	103.29	282.13	42.91
21	415.47	164.3		207.51	706	1.96	127.25	293.32	52.87
22	415.47	202.3		207.51	870.85	2.32	184.82	355.76	75.5
23	45.57	207		17.96	884.07	2.4	174.2	40.29	7.94
24	68.95	169.1		7.75	715.27	2.03	115.66	49.32	7.97
25	415.47	253.7		207.51	1104.04	2.79	277.95	451.02	113.55

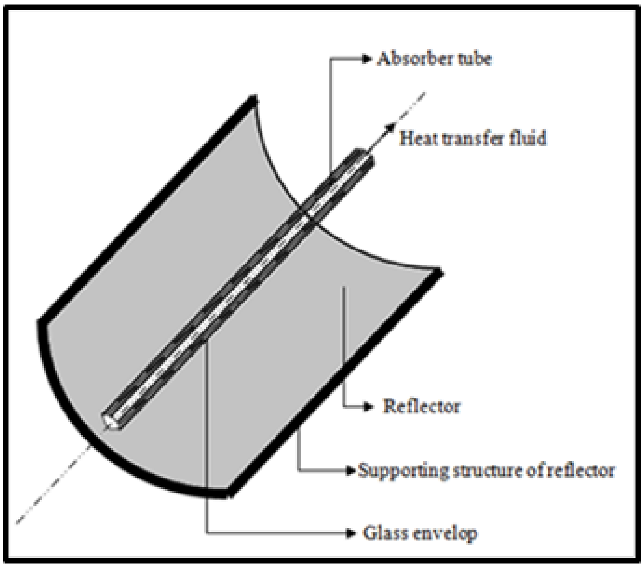

Figure 5.3: Parabolic solar trough of LS-3 type

Table 5.2: Input parameters for analysis of the parabolic troughs [103]

Parameter	Symbol	Values
Single collector width	W	5.76 m
Single collector length	L	12.27 m
Receiver inner diameter	$D_{r.i}$	0.066 m
Receiver outer diameter	$D_{r.o}$	0.07 m
Cover inner diameter	$D_{c.i}$	0.115 m
Cover outer diameter	$D_{c.o}$	0.121 m
Mass flow rate per collector trough	m_{HTF}	0.8 Kg/s
Global solar radiation	G_b	450 W/m^2
Temperature rise by single collector		165ºC
Inlet temperature of heat transfer fluid	$T_{HTF.o}$	400ºC
Outlet temperature of heat transfer fluid	$T_{HTF.in}$	70ºC
Inlet temperature of feed water	$T_{w.in}$	46.71ºC
Outlet temperature of feed water	$T_{w.o}$	126.5ºC
Mass flow rate of feed water	m_w	308.59 kg/s
Specific heat of HTF	$C_{p.HTF}$	2.821 kJ/kg K
Effectiveness of the counter flow heat exchanger	ε	0.86

5.4 Replacing of the LPHs by the parabolic solar troughs

In the present case, series arrangement of the parabolic trough collectors is considered for increase the temperature of heat transfer fluid and row arrangement is considered to increase the mass flow rate of the HTF. In present analysis, Therminol (VP-1) is considered as the heat transfer fluid. Considering a standard counter flow heat exchanger for exchanging the heat between feed water heater and heat transfer fluid, the related flow parameters are calculated.

The mass flow rate of HTF through the heat exchanger, and the number of collector modules connected in row and hence, in series are presented in Table 5.8. It is found that energy of about 103 MW is received by the feed water from HTF. To replace the LPHs, 161 collector modules are to be connected in row and related number of rows is 2. Total 322 collector modules are required to increase the feed water temperature to that of an existing conventional thermal power plant.

Table 5.3: Mass flow rate of HTF and number of collector modules connected in row and series

Parameter	**Quantity**
Heat received by feed water	102921.6 kJ/s
Total mass flow rate of HTF through HE	128.55 Kg/s
Number of collector modules in row	161
Temperature rise in a row	165°C
Number of rows in series	2
Total number of collectors	322

5.5 Thermodynamic Analysis

In concern to the present analysis, the equations for mass, energy and exergy balance, rate of irreversibility, specific exergy flow, shaft power, energy efficiency, exergy efficiency, turbine heat rate etc. are utilized as modeled and discussed in the Chapter–II in addition to the thermodynamic equations required for the energy and exergy analysis of the 500 MW coal based thermal power plant while the plant is integrated with solar energy using the parabolic trough collectors replacing the LP heaters.

The necessary governing equations related to amount of coal saving, reduction in emissions such as CO_2, SO_2 and reduction in ash generation, are discussed in the Chapter-IV. All those equations are utilized further to compare the coal saving, reduction in emissions and ash generation before and after the integration of solar energy with the plant.

However, the additional energy balance in correspondence to the mass balance is presented the following sub-sections.

5.5.1Work output before solar integration *(* $\dot{W}_{bi}$ *)*

The work output by the turbines before solar integration is calculated as*:*

$$\dot{W}_{bi} = (\dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3) + (\dot{m}_4 h_4 - \dot{m}_5 h_5 - \dot{m}_6 h_6 - \dot{m}_7 h_7) + (\dot{m}_5 h_5 - \dot{m}_8 h_8 - \dot{m}_9 h_9 - \dot{m}_{10} h_{10} - \dot{m}_{11} h_{11}) \tag{5.1}$$

5.5.2 Work output after the solar integration *(* $\dot{W}_{ai}$ *)*

The work output by the turbines after solar integration is calculated as*:*

$$\dot{W}_{ai} = (\dot{m}_1 h_1 - \dot{m}_2 h_2 - \dot{m}_3 h_3) + (\dot{m}_4 h_4 - \dot{m}_5 h_5 - \dot{m}_6 h_6 - \dot{m}_7 h_7) + (\dot{m}_5 h_5 - \dot{m}_{8a} h_{8a}) \tag{5.2}$$

5.5.3 Improvement in work output due to the solar integration

Here, the solar energy is used to heat the feed water using the parabolic solar troughs**.** The steam flow rate through the turbines increases due to integration of the solar energy and, hence, work output by the turbines also increases**.** The increase in the work output is calculated as*:*

$$\dot{W}_{improvement} = \dot{W}_{ai} - \dot{W}_{bi} \tag{5.3}$$

5.6 Results and Discussion

Since bleeding of steam from the LP turbine is no longer required due to replacement of LPHs by the parabolic trough collectors, the mass flow rate of steam through the LPT increases. The work output of the turbines as well as the total energy efficiency improves due to this integration of the solar energy; the related improvements are presented in Table 5.4. It is found that the overall efficiency of the plant increases from 35.5% to 37.62% in integration of solar energy or replacing the LPHs by solar troughs.

Table 5.4: Overall work output and energy efficiency at full load condition before and after integration of solar troughs

Condition	Overall work output of turbines (MW)	Overall efficiency
Before integration of the solar energy	512.3	35.5%
After integration of the solar energy	548.52	37.62%

The conclusion drawn out of the Table 5.4 includes a detailed energy and exergy analysis. The necessary calculation on the flow rate of energy, exergy and irreversibility before and after integration of the solar energy is presented in the Table 5.5 and Table 5.6, respectively. Based on the data of Table 5.5 and Table 5.6, it is found that, at 100% unit load condition, the work output increases by an amount of 36.21 MW whereas the irreversibility in the turbines decreases by an amount of 3.63 MW, due to integration of the solar energy. The overall efficiency of the plant is then increased by 2.12 %. At 80% unit load condition, the work output increases by 34.1 MW, and irreversibility of the turbines decreases by 6.61 MW. On 60% unit load condition, the work output increases by 25.88 MW and irreversibility of the turbines decreases by 4.03 MW.The differences in work output and irreversibility rate before and after integration of the solar energy are presented in the form of bar charts in Fig. 5.4 and Fig. 5.5, respectively.

Table 5.5: Flow rate of energy, exergy and irreversibility before integration of the solar energy

Unit Load (%)	Energy input in MW	Energy output in MW	Exergy input in MW	Exergy output in MW	Work done in MW	Irreversibility in MW	Actual work in MW
100	3651.75	3139.44	1405.33	847.71	512.31	45.31	497.37
80	2929.48	2524.78	1156.02	713.31	404.7	38.01	397.19
60	2233.35	1929.98	886.93	557.05	303.37	26.51	297.65

Table 5.6: Flow rate of energy, exergy and irreversibility after integration of the solar energy

Unit Load (%)	Energy input in MW	Energy output in MW	Exergy input in MW	Exergy output in MW	Work done in MW	Irreversibility in MW	Actual work in MW
100	3651.75	3103.22	1405.33	815.1	548.52	41.68	531.46
80	2929.48	2490.68	1156.02	684.82	438.8	31.40	427.33
60	2233.35	1904.1	886.93	553.08	329.25	22.48	321.91

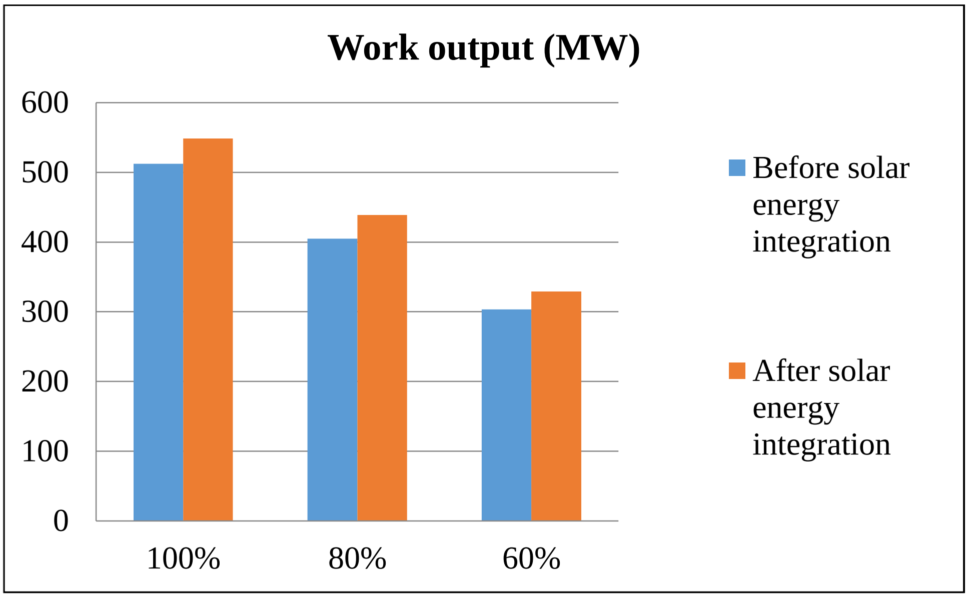

Figure 5.4: Work output at different unit loads before and after integration of the solar energy

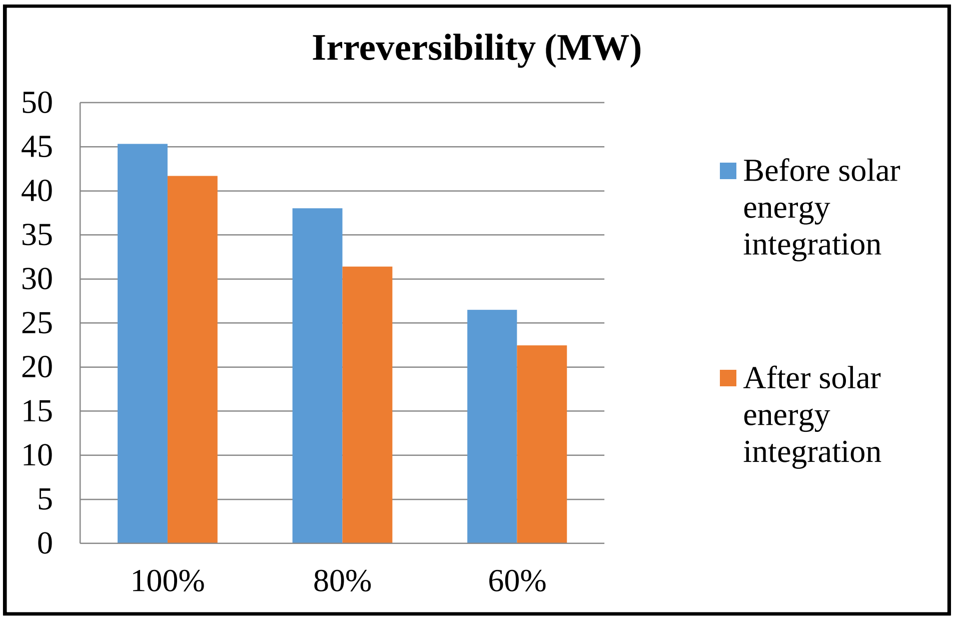

Figure 5.5: Turbine irreversibility at different unit loads before and after integration of the solar energy

Control of emissions and reduction in the consumption of coal are also a recent emphasizing issue in controlling the environmental pollution. Integration of the solar energy to the existing power plant is also a beneficial. Accordingly, this work calculated the related heat rate improvement, reduction in emissions of the CO_2 and SO_2, and reduction in ash generation. Table 5.7 presents the necessary calculation before integration of the solar energy whereas Table 5.8 presents corresponding values after the solar integration. Those values are also presented graphically in Fig. 5.6 and Fig. 5.7. According to the Table 5.7 and Table 5.8, on integration of solar energy with the 500 MW conventional thermal power plant and an increase in unit load from 80% to 100%, the turbine heat rate is improved by 6.36 kJ/kWh compare to without integration of the solar integration. On an increase of unit load from 60% to 80%, the turbine heat rate is improved by 2.8 kJ/kWh and for an increase in unit load from 50% to 60%, the turbine heat rate is improved by 1.09 kJ/kWh in case of integration of the solar energy with existing coal based thermal plants.

Table 5.7: Heat rate *(*HR*)* improvement, reduction in emission of CO_2 and SO_2, and reduction of ash generation before integration of the solar energy

Sl. No	**Parameters**	**Before integration of solar energy**		
1		**Increase in unit load (50% to 60%)**	**Increase in unit load (60% to 80%)**	**Increase in unit load (80% to 100%)**
2	HR Improvement in kJ/kWh	17.50	48.33	106.01
3	Reduction of coal consumption in kg/day	267.84	762.48	1230.72
4	Reduction in CO_2 emission in kg/day	419.86	1195.25	1929.25
5	Reduction in SO_2 emission in kg/day	1.71	4.88	7.88
6	Reduction in ash generation in kg/day	107.14	304.99	492.29

Table 5.8: Heat rate (HR) improvement, reduction in emission of CO_2 and SO_2, and reduction of ash generation after integration of the solar energy

Sl. No	Parameters	After integration of solar energy		
1		Increase in unit load (50% to 60%)	Increase in unit load (60% to 80%)	Increase in unit load (80% to 100%)
2	HR Improvement in kJ/kWh	18.59	51.13	112.37
3	Reduction of coal consumption in kg/day	283.7	805.55	1301.63
4	Reduction in CO_2 emission in kg/day	443.12	1254.42	2042.18
5	Reduction in SO_2emission in kg/day	1.92	5.13	8.25
6	Reduction in ash generation in kg/day	113.51	321.85	519.02

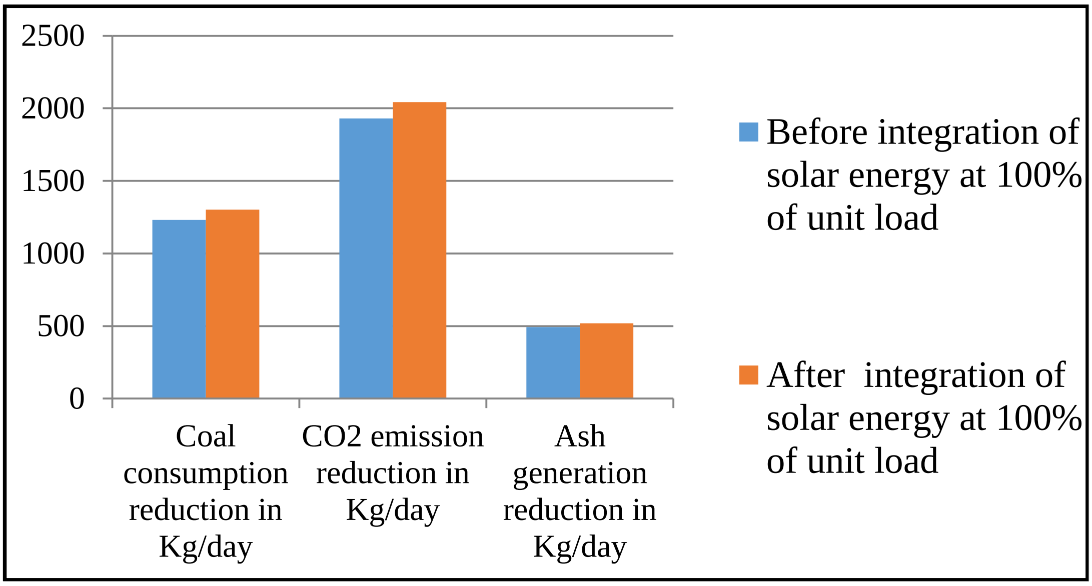

Figure 5.6: Comparison of coal consumption reduction, reduction in CO_2 emission and ash generation before and after integration of the solar energy

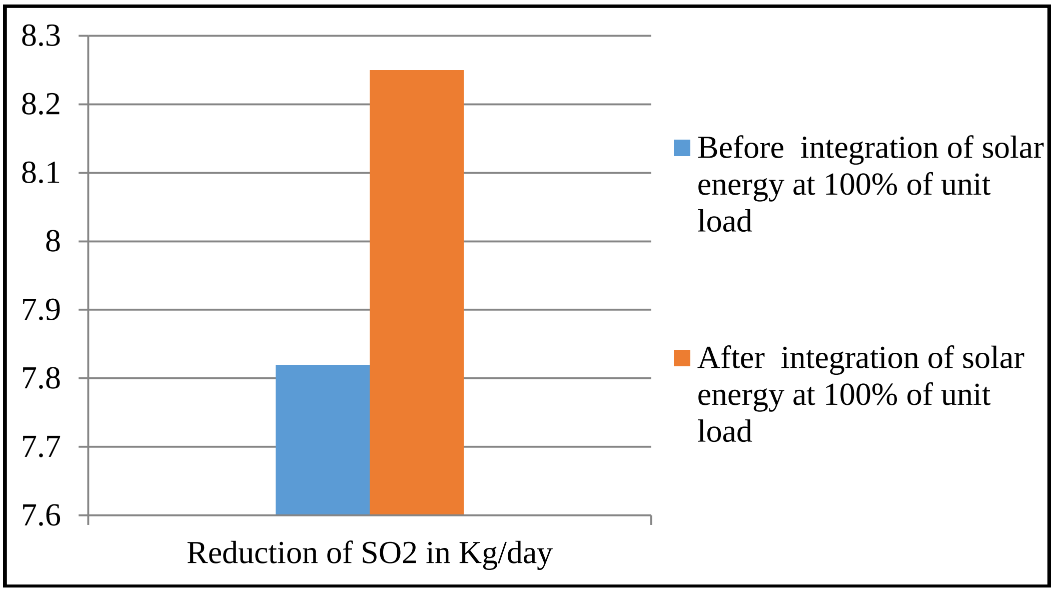

Figure 5.7: Comparison in reduction of SO_2 emission before and after integration of the solar energy

5.7 Conclusions

The current study reported that installing parabolic trough collectors at the place of low pressure heaters (LPHs) for feed water heating improves the performance of existing plant and reduces coal consumption as well as the harmful emissions. It is noticed that the output of the cycle is increased by 36.21 MW, and overall thermal efficiency is improved by 2.12% simply by integrating the solar energy with the 500 MW coal based thermal power plant. This was accomplished without affecting any of the other components of the cycle. The results also indicate that, at increase of unit load increased from 80% to 100%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 70.91 kg/day, 112.85% kg/day, 0.37 kg/day and 26.73 kg/day, respectively on integration of the solar energy. On unit load increased from 60% to 80%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 43.04 kg/day, 59.17% kg/day, 0.25 kg/day and 16.86 kg/day, respectively. At unit load increased from 50% to 60%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 15.86 kg/day, 23.26 kg/day, 0.21 kg/day and 6.37 kg/day, respectively. Therefore, utilization and integration of the solar energy in a conventional thermal power plant is beneficial in term of reduction in both greenhouse gas emissions and pollutants on surrounding environment.

5.7 References

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Chapter - VI

Conclusions and Future Work

Chapter VI

Conclusions and Future Work

6.1 Summary of the thesis

A crisis in the available energy in the near future has been reported by many researchers on contrast to the increase in demand of electrical energy, which emphasis on substantial reduction in the use of fossil fuels by developing an environmentally suitable power plant with a better efficiency. Effective power plant additionally reduces emission of the green-house gas and reduces pollutants in the air. As reported, existing steam power plants are more vulnerable in polluting the environment, and subsequent a cause to the global warming, if they are older. Thus, a method for systematic analysis is required to meet the challenge in order to reduce coal consumption, enhance performance of the existing power plants, reduce emission, and hence, to reduce the global warming.The energy and exergy-based analysis of a power generating unit is mostsuitable and provides insight into the losses in various components of the unit.The exergy analysis determinesextent and location of the exergy destructions, which helps in enhancing pre-existing structures and related machineries, or to design and implement new ones. This work is a cosideration of the energy and exergy analysis of an existing power plant in order to identify the locations of exergy destruction and necessary inprovement of its efficiency.

As a basic consideration, in the chapter - II, only turbines of a 500 MW coal fired thermal power plant of Damodar Valley Corporation (DVC), India is considered. The analysis is used for determining performance of turbines (HPT, IPT and LPT) at different unit load conditions (60%, 80% and 100%). A control volume, which includes three turbines those are coupled to a generator, is considered. A set of governing equations representing insight thermodynamics of the plant is used to carry out the analysis. These equations are mass continuity, energy balance, and exergy balance; modelled following the basics of thermodynamics under steady state condition. It is found that irreversibility rate of the turbines increases with an increase in unit load. The rate of irreversibility of the LPT (low pressure turbine) is higher than that of HPT (high pressure turbine) and IPT (intermediate pressure turbine). It is also found that, the exergy efficiency of the HPT is maximum, and the exergy efficiency of the LPT is minimum for all unit load conditions. It is concluded here that a proper replacement of blades may be advantageous to minimize the destruction in exergy of the LPT. It is also concluded out of the predictions that a plant runs at full load is the most economical condition of operation because the exergy efficiency of the turbines is maximum at this condition of full unit load.

In the Chapter - III, the 1st law (energy) as well as the 2nd law (exergy) analysis of the turbine cycle and individual component of the system in a 500 MW coal-based power plant of DVC, India is considered. Accordingly, a thermodynamic model is developed for the analysis at various unit load conditions (100%, 80%, and 60%, etc.). At the steady-state condition of the flow, the equations for mass continuity, energy balance, and finally exergy balance are considered. In addition to these, equations for irreversibility and 2nd law efficiency of specific components such as turbines, pumps, condensers, deaerator, and heaters are taken into consideration for their performance analysis. It is found that the rate of net energy input is nearly 2.21 times than that of the power output in the turbine cycle when the unit load is 100%, but it is approximately 2.4 times when the unit load is 50%. It is observed that the irreversibility rate for the LPH-2 (LPH stands for low pressure heater) is more compared to the LPH-1 and LPH-3. It is also found that the rate of irreversibility of LPT and HPH-2 (HPH stands for high pressure heater) is higher than other components of the turbine cycle. So it is concluded that some corrective actions are required in both the operation and maintenance decisions. To minimise the exergy destruction in the HPH-2, the operation decision may include properly adjusting drip level and venting of air. Sometimes, after a number of years of service, the scale forms over the heating surfaces which cause deterioration of the heater performance. Hence, proper maintenance decisions such as replacement or retubing may improves the performance of the heaters.

In the Chapter - IV, an effort is given to evaluate the reduction in coal consumption, ash generation and greenhouse gas emissions such as CO_2, SO_2 due to improvement in heat rate. A reduction in coal consumption, also in CO_2, SO_2 emissions, and in ash generation is found with improved heat rate of the turbine while the unit load is increased. With increase in unit load, the fuel consumption reduces significantly. The heat rate is minimum for higher loads (80% and 100%), which is an economical condition of the operation. The subsequent reduction in consumption of coal, in generation of ash, in emission of SO_2 and CO_2 due to improvement in heat rate is predicted further. At 100% unit load, reduction in coal consumption is 1230.72 kg per day. Under full unit load condition, reduction rates for CO_2 (1929.25 kg/day), SO_2, (7.88 kg/day) and generation of ash (492.29 kg/day) are maximum.

It is already discussed in the literature review section (as discussed in Chapter-I) that an integration of solar energy with the conventional existing power plants possibly reduces the harmful emissions. In order to reduce emissions of the harmful gases and to increase thermal efficiency of the existing thermal power plants, an integration of solar energy is considered in this work. Accordingly, in chapter -V,a detailed analysis is performed for an existing coal based 500 MW thermal power plant by predicting its performance, overall thermal efficiency, coal consumption rate before and after integrating the parabolic solar trough collectors in the place of LPHs in order to heat the feed water. Parabolic solar troughs are integrated in place of LPHs. Parabolic solar troughs receive heat energy from the sun, and heat the boiler feed water before it passes through the economizer and enters the boiler drum. As a result, steam bleeding from different stages of the LPT for heating the feed water is eliminated. It is noticed that the output of the cycle is increased by 36.21 MW, and overall thermal efficiency is improved by 2.12% simply by integrating the solar energy with the 500 MW coal based thermal power plant. This was accomplished without affecting any of other components of the cycle. The results also indicate that, in increase of unit load from 80% to 100%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 70.91 kg/day, 112.85% kg/day, 0.37 kg/day and 26.73 kg/day, respectively on integration of the solar energy. On unit load increased from 60% to 80%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 43.04 kg/day, 59.17% kg/day, 0.25 kg/day and 16.86 kg/day, respectively. At unit load increased from 50% to 60%, additional reduction in coal consumption, additional reduction in CO_2, SO_2 emission and additional reduction in ash generation are 15.86 kg/day, 23.26 kg/day, 0.21 kg/day and 6.37 kg/day, respectively. Therefore, utilization and integration of the solar energy in a conventional thermal power plant is beneficial in term of reduction in both the greenhouse gas emissions and pollutants on surrounding environment.

In brief, implementation of present model as presented in the thesis is essential in order to perform effectively by an existing coal based thermal power plant, which reduces coal consumption, reduces emissions, and moreover reduces the global warming.

6.2 Scope of the future work

- Further study and research on integration of the solar energy with the existing coal fired thermal power plant towards the practical implementation.
- Develop and design of parabolic solar troughs which can effectively integrated at the place of LPHs as well as HPHs to get better performance of anexisting coal fired thermal power plant.
- After integrating the solar energy at the place LPHs as well as HPHs, develop a thermodynamic model to evaluate reduction in coal consumption as well as reduction in ash generation and also reduction in harmful emissions like CO_2, SO_2 etc.