OPTIMIZATION OF A COAL FIRED THERMAL POWER PLANT USING THERMOECONOMIC TECHNIQUE

THESIS

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by

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CANDIDATE'S DECLARATION

I hereby declare that this thesis entitled **OPTIMIZATION OF A COAL FIRED THERMAL POWER PLANT USING THERMOECONOMIC TECHNIQUE** by **MUKESH GUPTA**, being submitted in fulfilment of the requirements for the Degree of Doctor of Philosophy in MECHANICAL ENGINEERING, at Faculty of Engineering & Technology of YMCA University of Science & Technology Faridabad, in November 2017, is a bonafide record of my original work carried out under guidance and supervision of **Dr. RAJ KUMAR, PROFESSOR, MECHANICAL ENGINEERING** and has not been presented elsewhere.

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CERTIFICATE OF THE SUPERVISOR

This is to certify that this Thesis entitled **OPTIMIZATION OF A COAL FIRED THERMAL POWER PLANT USING THERMOECONOMIC TECHNIQUE** submitted in fulfilment of the requirement for the Degree of Doctor of Philosophy in **MECHANICAL ENGINEERING** under Faculty of Engineering & Technology of YMCA University of Science & Technology, Faridabad, in November 2017, is a bonafide record of work carried out under my guidance and supervision.

I further declare that to the best of my knowledge, the thesis does not contain any part of any work which has been submitted for the award of any degree either in this university or in any other university.

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ABSTRACT

The electricity sector in India supplies the world's 6^{th} largest energy consumer, accounting for 3.4% of global energy consumption by more than 17% of the global population. Due to the fast paced growth of Indian economy, there has been an average increase of 3.6% in the energy demand per annum over the last 30 years. In December 2010, the installed power generation capacity of India stood at 165,000 MW and the per capita energy consumption was 612 KW. Hence, it can be concluded that India is an energy deficient country and to meet the energy requirements of the fast developing economy some urgent steps need to be taken.

Performance evaluation of the thermal power plants has been a great challenge for engineers & scientists over the years. Conventional evaluation techniques used for these plants are based on the first law analysis. Extensive research in this field suggests that a more effective way of evaluating a power plant could be the second law analysis, also known as the exergy analysis. In the recent years, economic principles have been introduced in combination with the exergy principles to assess the performance of thermal power plants. This new and more realistic evaluation technique is called exergoeconomics or thermoeconomics. The exergy analysis predicts the thermodynamic performance of an energy system whereas the exergoeconomic technique estimates the unit cost of products and quantifies the monetary losses because of irreversibilities associated with the various components of the power plant.

The current study involves second law based exergy analysis of 210 MW coal based thermal power plant and 25MW open cycle gas turbine power plant. Thermodynamic laws of mass and energy conservation laws have been applied, to each component and the plant, to derive the energy and exergy balance equations.

Exergoeconomics analysis has been performed for the coal based and gas turbine power plants. The economic analysis is combined with the exergy analysis. The cost balance equations for

each component have been derived. These equations, along with some auxiliary relations, have been solved for the average unit cost for all flows and the total cost flow rate has been determined for all the components of the plant.

From the exergy based analysis critical components have been determined where maximum exergy destruction takes place. Optimization for these components has been done by combining the results from exergy analysis and exergoeconomics. Different thermodynamic parameters which affect the performance of these components have been considered and optimization has been done by analyzing the effect of variation in these thermodynamic parameters on unit product cost of the components.

For the coal based thermal power plant, boiler and steam turbine have been found to be the critical components where maximum exergy destruction takes place. For the boiler, hot air temperature and feed water temperature are the most important parameters which affect its performance significantly. Effect of variation of these parameters on the performance of the boiler has been done and optimization has been achieved based on the effect of hot air temperature on the unit product cost of the boiler and air pre heater as a trade- off between these two values. For the steam turbine, analysis has been done considering the effect of inlet steam temperature on its performance. Optimization has been achieved by analyzing the effect of inlet steam temperature on unit product cost of the steam turbine and boiler as a trade- off between these two values.

For the open cycle gas turbine power plant, combustion chamber and gas turbine have been found to be the critical components where maximum exergy destruction takes place. Effects of compressor pressure ratio and air inlet temperature on the performance of compressor, combustion chamber and gas turbine have been analyzed. Optimization has been achieved for the open cycle gas turbine power plant as best balance between the unit product cost of the compressor and combustion chamber as functions of compressor pressure ratio and unit product costs of combustion chamber and gas turbine as functions of turbine inlet temperature. A brief comparison has been made in the setup and working principles of the coal fired and open cycle gas turbine power plants from thermodynamic and economic viewpoint.

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NOMENCLATURE

SYMBOLS

c	Cost per unit of exergy ($/ MJ or Rs./ MJ)
c_p	Specific heat (kJ/ kg K)
$\dot{C}$	Cost rate associated with exergy ($/ h or Rs/ h)
e	Specific exergy (MJ/ kg)
E	Exergy (MJ)
$\dot{E}$	Rate of exergy flow (MW)
F	Fuel
h	Specific enthalpy (MJ/ kg)
i	Interest rate
I	Irreversibility
k	Unit exergetic cost (component)
$\dot{m}$	Mass flow rate (kg/ s)
N	Number of mixture components
p	Pressure (bars)
P	Product
P_0	Atmospheric pressure
PW	Present worth
$PWF\,(i,n)$	Present worth factor
$\dot{Q}$	Heat transfer rate

R	Universal gas constant (kJ/ kg K)
s	Specific entropy (kJ/ kg K)
S	Entropy (MJ/ K)
S_n	Salvage value ($ or Rs.)
T	Temperature (K)
T_0	Atmospheric temperature
TIT	Turbine inlet temperature
U	Internal energy (MJ)
v	Specific Volume (m^3)
V	Volume (m^3)
W	Power (MW)
Z	Monetary flow rates ($/ h or Rs./ h)
ε	Second law or exergetic efficiency

SUBSCRIPTS

0	Ambient conditions
1, 2....	State points
ac	Air compressor
b	Boiler
baux	Auxiliary work supplied to boiler
c	Condenser

cc	Combustion chamber
e	Exit
f	Fuel
F	Fuel
gt	Gas turbine
i	Inlet
k	kth component
net	Net output
OP	Overall Plant
p	Product
P	Product
L	Loss
q	Heat
T	Total
t	Turbine
taux	Auxiliary work supplied to turbine

SUPERSCRIPTS

CH	Chemical exergy
KN	Kinetic energy
OM	Operation and maintenance cost
PH	Physical exergy
PT	Potential exergy
T	Thermal
tot	Total
W	Work or electricity

CHAPTER-1

INTRODUCTION

1.1 BACKGROUND

Energy is one of the most fundamental parts of our universe. Energy has come to be known as a `strategic commodity' and any uncertainty about its supply can threaten the functioning of the entire economy, particularly in developing economies. India's substantial and sustained economic growth is placing enormous demand on its energy resources. The demand and supply imbalance in energy sources is pervasive requiring serious efforts by Government of India to augment energy supplies as India faces possible severe energy supply constraints. Energy requirement in our country is increasing at a very rapid rate. Achieving energy security in this strategic sense is of fundamental importance not only to India's economic growth but also for the human development objectives that aim at alleviation of poverty, unemployment and meeting the Millennium Development Goals (MDGs).

The electricity sector in India supplies the world's 6^{th} largest energy consumer, accounting for 3.4% of global energy consumption by more than 17% of the global population. Due to the fast paced growth of Indian economy there has been an average increase of 3.6% in the energy demand per annum over the last 30 years.

The total installed capacity for electricity generation in the country has increased from 145755 MW as on 31.03.2006 to 284,634 MW as on 31.03.2014, registering a compound annual growth rate (CAGR) of 7.72% (https://en.wikipedia.org/wiki/Electricity_sector_in_India accessed on 20/12/2015.). There has been an increase in generating capacity of 17990 MW over the last one year, the annual increase being 6.75%. The highest rate of annual growth (11.66%) from 2012-13 to 2013-14 in installed capacity was for Thermal power. At the end of March 2014, thermal power plants accounted for an overwhelming 70.25% of the total installed capacity in the country, with an installed capacity of 199,947 MW.

Coal production in the country during the year 2013-14 was 565.77 million tons (MTs) as compared to 556.40 MTs during 2012-13, registering a growth of 1.68%. Considering the trend of production from 2005-06 to 2013-14, it is observed that coal production in India was about 407.04 MTs during 2005-06, which increased to 565.77 MTs during 2013-14 with a CAGR of 3.73%. During the same period the CAGR of Lignite was about 4.33% with production increasing from 30.23 MTs in 2005-06 to 44.27 MTs in 2013-14.

Total Electricity generation in the country, from utilities and non-utilities taken together during 2013-14 was 11,79,256 GWh. Out of this 8,53,683 GWh was generated from thermal and 1,34,731 GWh was from hydro and 34,200 GWh was generated from nuclear sources. The total consumption of energy from conventional sources increased from 23,903 Peta joules during 2012-13 to 24,071 Peta joules during 2013-14, showing an increase of 0.70%. Hence it can be safely concluded that India is an energy deficient country and to meet the energy requirements of the fast developing economy.

Power Crisis has been a long clamour in India and this seems to persist for the coming decade or so. Beyond optimistic illusions, ground realities are too fierce to be accepted. Seeds of improvement are however being planted at all possible arenas which can be broadly classified as:

1. Power Generation
2. Power Transmission &
3. Power Distribution

The thermal power plants contribute maximum towards electricity generation in the country and mostly based on two types of technologies:

1. Steam Power Plants
2. Gas turbine power plants

1.1.1 STEAM POWER PLANTS

Steam power plants used in India are primarily coal based. All of them use the conventional drum type boilers. These steam power plants can be classified into two categories i.e. condensing or non- condensing. In the condensing steam power plants, the outlet steam from the turbine is discharged to a condenser whose pressure is maintained below atmospheric. To cater the accelerated need of the country, Ultra Mega Power Plants (UMPPs) have been proposed. These are giant power plants of 4000 MW at one place. In addition to these, a number of Mega Power Projects (More than 1000 MW) are also being promoted. The unit size of power plants has also experienced a supercritical shift in technology. Thus focus is on more efficient supercritical units of 660 MW and 800 MW. However, considering imported coal as the main fuel, unit size can go up to 1000 MW. As the domestic coal available for power sector is having very high (40 to 45%) ash content feasibility of higher size units is yet to be examined. Meanwhile after 500MW / 600 MW (subcritical) the next suitable supercritical size was decided 660 MW or 800 MW for coal based thermal power plants.

1.1.2 GAS TURBINE POWER PLANTS

Currently the total capacity of gas turbine power plants in India is about 26699.9 MW which is increased by 51.3% as compare to the year 2011 in which it was 13711.27 MW. Gas turbine power plants can be classified as open cycle and combined cycle plants. Power plants based on the combined cycle are generally based on Rankine or Brayton thermodynamic cycles.

Gas turbine systems used nowadays comprise of four major components: compressor, combustion chamber, gas turbine and a generator as illustrated in Figure 1.1. Gas turbines for power generation can be either industrial which is of heavy frame or can be of aero derivative. Aero derivative gas turbines have higher initial cost and more sensitive to compressor inlet temperatures as compared to the industrial gas turbines. The functioning of a gas turbine power plant includes the compression process where large volumes of air are compressed using multistage compressors to high pressures. The compressed air is then fed to the combustion chamber which uses natural gas as fuel to increase the temperature of the air. This high pressure and temperature air is then used to directly run the turbines which are coupled with the generators to produce electricity.

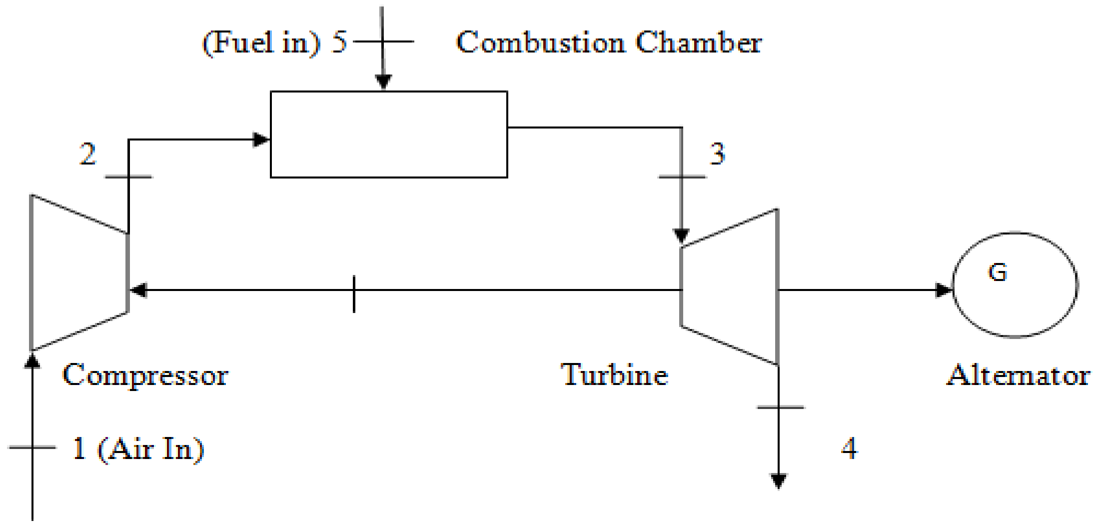


Figure 1.1 Open cycle gas turbine power plant

Gas turbine power plants in India generally use natural gas as fuel because of its environment friendly nature coupled with cost effectiveness. Out of total production of natural gas in the country about 40% is used in the gas turbine power plants. GAIL is the main source of fuel for most of these power plants. Most of the gas production in the country comes from the western sector. The gas obtained from the shores is sweetened by removing the sulphur content. This process is generally done at Hazira.

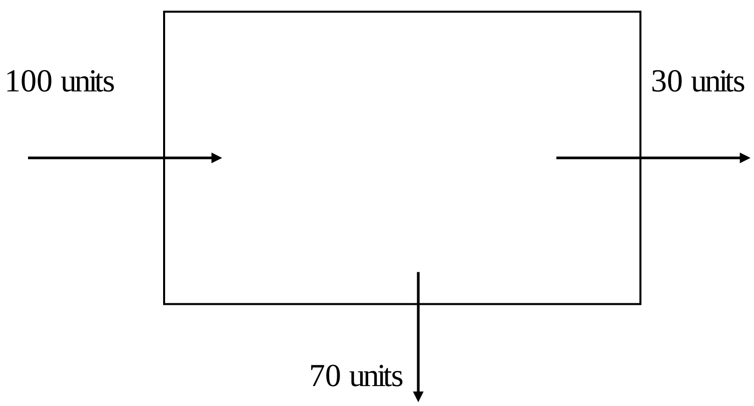
Performance evaluation of the thermal power plants has been a great challenge for the engineers & scientists for the years. Conventional evaluation techniques used for these plants are based on the first law analysis. Extensive research in this field suggested that a more effective way of evaluating a power plant could be second law analysis, also known as the exergy analysis. In the recent years, economic principles have been introduced in combination with the exergy principles to assess the performance of thermal power plants. This new and more realistic evaluation technique is called the exergoeconomics or thermoeconomics. The exergy analysis usually predicts the thermodynamic performance of an energy system whereas exergoeconomics estimates the unit cost of products and quantifies the monetary losses due to irreversibilities associated with various components of the power plant.

The aim of the current study is to combine the thermodynamic and economic evaluation of already existent power plants. For this, a coal based power plant and an open cycle gas turbine power plant have been considered for thermodynamic analysis and exergoeconomic analysis and optimization. The results obtained in the current study can further be used for R & D endeavors towards conservation of energy.

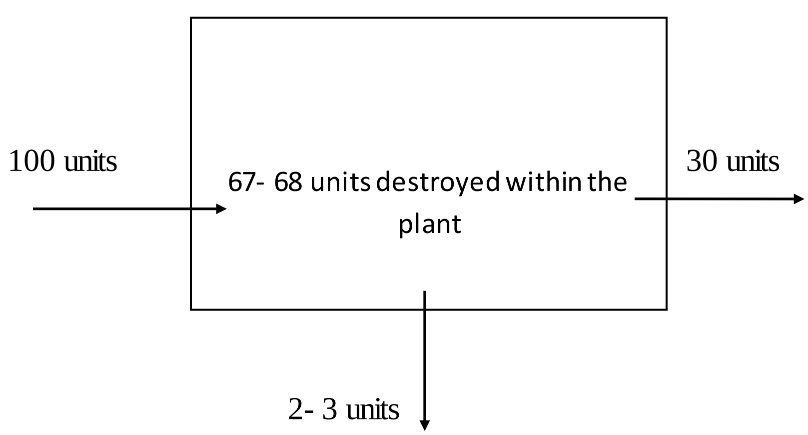
1.2 CONCEPTS AND APPLICATIONS OF EXERGETIC THEORY AND EXERGOECONOMIC ANALYSIS

In the last few years there has been an increased awareness among the scientific community about the importance of energy to human race and there has been a growing concern about the rate at which the non-renewable sources of energy are getting depleted. Hence greater emphasis is being laid on analyzing the performance of various thermodynamic systems. This evaluation is primarily done using various thermodynamic principles which have been used for a long time. For the last half century there has been a serious problem of energy crisis which has forced the scientific community to look beyond the traditional concepts of thermodynamic analysis and look for better ways to analyze the energy production methods. This has led to the development of analysis techniques based on the concept of exergy which has been found to be a much better tool to evaluate and analyze the performance of thermal systems.

To understand the various analysis techniques based on the concept of exergy, a clear distinction needs to be drawn between the conventional concept of energy and the concept of exergy. According to the first law of thermodynamics, energy is conserved in every device or thermodynamic process and it cannot be destroyed. The concept of destruction which cannot be applied to energy has been successfully applied to exergy. The distinction between the concepts of energy and exergy is illustrated in Figure 1.2. The analysis based on energy concept is shown in Figure 1.2 (a). it shows that out of 100 energy units which are entering the system with fuel only 30 get converted into electricity and the rest are released to the atmosphere. Similar analysis done using the exergy concept provides a different picture. Figure 1.2 (b) shows that out of 100 exergy units available only 30 units get converted to electricity and the remaining 70 need to be accounted for. Careful analysis shows that out of these 70 units, 67-68 units are destroyed in the various plant components by various forms of irreversibilities and only 2 to 3 units are actually discharged to the atmosphere. This analysis shows that the results obtained using the conventional energy concept can be misleading. The exergy analysis not only provides the results in terms of losses incurred in a thermal system but it also allows the analyst to identify the actual source of this loss by highlighting the system inefficiencies in terms of irreversibilities.



(a) Energy Basis



(b) Exergy Basis

Figure 1.2 Analysis of thermal systems using the concepts of energy and exergy

It has been found that it is much easier to assign monetary values to exergy as compared to energy. Serious misevaluations have resulted when researchers have tried to assign monetary values to energy.

Some of the important characteristics of exergy are as follows:

- Exergy is the maximum theoretical work that can be extracted from a thermal system comprising of many sub systems as the system passes from a given state to equilibrium with its surroundings which pass through the dead state. A system in complete equilibrium with its environment has no exergy. No difference appears in temperature, pressure etc., hence there is no driving force for any process.
- The exergy of a system increases more it deviates from its environment.
- When energy loses its quality, exergy is destroyed. Exergy is that part of energy which is useful and has economic value and is worth managing carefully.
- Exergetic efficiencies are a measure of approach towards the ideal state which is not necessarily true for energy.
- The exergy of any thermal system can be expressed as the sum of chemical and thermo-mechanical exergy. Thermo- mechanical exergy is further classified as physical, kinetic and potential energy.

The major difference between energy and exergy analysis of a thermal system is that whereas the energy based analysis is worked around the first law of thermodynamics, any analysis based on the concept of exergy is centered round the second law of thermodynamics. Over the years more and more researchers are shifting their focus towards the exergy based analysis of thermal systems.

The total exergy of any thermal system is derived as follows:

$$E = E^{PH} + E^{K} + E^{P} + E^{CH} \tag{1.1}$$

The superscripts, *PH, K* and *P* represent the physical, kinetic and potential exergies. The first three terms on the right hand side combine to give the thermo- mechanical exergy of the system. Superscript *CH* refers to the chemical exergy. It has been observed that exergy is an intensive property hence it is much more convenient to work with it on unit of mass or molar basis. The total specific exergy on the mass basis is given as follows:

$$e = e^{PH} + e^{K} + e^{P} + e^{CH} \tag{1.2}$$

where, e denotes the specific exergy.

Considering the system to be at rest relative to its environment ($e^K = e^P = 0$), the total physical exergy of the system is given as:

$$E^{PH} = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) \tag{1.3}$$

where, *U, p, V, T* and *S* denote the internal energy, pressure, volume, temperature and entropy respectively at the specified state. Subscript *0* denotes the atmospheric condition.

The chemical exergy for a closed system is given as:

$$E^{CH} = \sum_R N_R(\mu_{R0} - \mu_R^0) \tag{1.4}$$

where, *N* is the number of moles of species *R* in the mixture and μ denotes the chemical potential.

The total exergy of the system can be represented as:

$$E = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) + \sum_R N_R(\mu_{R0} - \mu_R^0) \tag{1.5}$$

The total specific exergy of a closed system is given as:

$$e = (h - h_0) - T_0(s - s_0) + \sum_R N_R(\mu_{R0} - \mu_R^0) \tag{1.6}$$

where *h* and *s* represent the specific enthalpy and entropy respectively.

Exergy analysis is used to analyze the thermodynamic performance of any energy system and it also provides information about the efficiency of individual components by accurately quantifying the entropy generation in different components. For a energy process considering "*P*" as the product and "*F*" as the fuel, the following equation is always satisfied:

$$F - P = I \geq 0 \tag{1.7}$$

where I is the irreversibility associated with the system.

The exergetic efficiency of the system is given as:

$$\eta_{exergetic} = P / F \leq 1 \tag{1.8}$$

The inverse of exergetic efficiency represents the unit exergetic cost of the product and is given as:

$$k_p = F / P \geq 1 \tag{1.9}$$

Monetary evaluation of energy systems is done by incorporating two types of interrelated environments (1) The physical environment and (2) The economic environment characterized by the market prices. In such an analysis, the relevant variable is the exergetic cost which provides information about the actual amount of exergy that is needed to produce the desired product.

During the economic analysis, various factors such as market prices (c_p) and cost of depreciation, maintenance and installation of productive units (Z) are taken into account. Any improvement in the efficiency of the system usually results in higher capital investment.

1.3 CHAPTERWISE ORGANIZATION OF THE THESIS

The chapter wise summary of the thesis is as follows:

Chapter 1: Introduction

This chapter provides an overview of the current energy generation trends in the country, basic concepts of power plants, exergy, exergetic costs and exergoeconomic evaluation of power plants. The chapter also discusses the proposed objectives of the current study and in the end the chapter wise organization of thesis has been discussed.

Chapter 2: Literature Review

This chapter provides in detail the relevant literature survey which has been carried out to understand the various aspects such as thermodynamic analysis of power plants, economic analysis of power plants and optimization of power plants from both thermodynamic and economic view points. Different thermal systems analyzed by various researchers have been included in this chapter.

Chapter 3: Thermodynamic Analysis of Coal Based Thermal Power Plant

This chapter includes the analysis of the 210 MW coal based thermal power plant from thermodynamic considerations. Mass and energy conservation laws have been applied to each component and also for whole system. The energy and exergy analysis for the base condition of 210 MW condition has been performed to pinpoint the location and magnitude of process irreversibilities. Further, the deviation of first and second law efficiency has been estimated. The expressions for exergetic efficiency have been developed for critical components as functions of important parameters.

Chapter 4: Exergoeconomic Analysis and Optimization of Coal Fired Thermal Power Plant

This chapter deals with the exergoeconomic analysis for 210 MW thermal power plant. In this analysis, mass and energy conservation laws have been applied to each component. Quantitative balance of exergies and exergetic costs for each component and for whole system was considered carefully. The exergy- balance and cost balance equation has been used in these analyses. Programming in Excel has been used to arrive at the optimal solutions. These optimal solutions have been achieved as best balances between the unit product costs of different components.

Chapter 5: Thermodynamic Analysis of Open Cycle Gas Turbine Power Plant

This chapter deals with the thermodynamic analysis of a 25 MW open cycle gas turbine power plant. The energy and exergy analysis for the base condition of 25 MW condition has been performed to pinpoint the location and magnitude of process irreversibilities. Further, the deviation of first and second law efficiency has been estimated. The expressions for exergetic efficiency have been developed for critical components as functions of important parameters.

Chapter 6: Exergoeconomic Analysis and Optimization of Open Cycle Gas turbine Power Plant

This chapter deals with exergetic and thermoeconomic analysis for a 25 MW open cycle gas turbine power plant. Programming in Excel has been used to arrive at the optimal solutions. These optimal solutions have been achieved as best balances between the unit product costs of different components. In this analysis, mass and energy conservation laws were applied to each component. Quantitative balance of exergies and exergetic costs for each component and for whole system was considered carefully.

Chapter- 7: Comparison of Coal Fired and Open Cycle Gas Turbine Power Plants

In this chapter a brief comparison has been between the setup and working principles of the coal fired and open cycle gas turbine power plants from thermodynamic and economic viewpoint.

Chapter 8: Overall Conclusions and Future Scope of Work

This chapter deals with the overall conclusions of the current study and explores the future scope of work related to the field.

CHAPTER- 2

LITERATURE REVIEW

2.1 INTRODUCTION

In the current study, literature review has been done extensively to understand the work done by various researchers on the following three major aspects of thermal systems in general and power plants in particular:

a) Thermodynamic analysis and optimization of power plants.
b) Economic analysis of power plants.
c) Exergo- economic optimization of power plants.

2.2 THERMODYNAMIC ANALYSIS OF POWER PLANTS

Petrakopoulou et. al. (2016) analyzed the four hybrid systems that couple a reference –biomass and photovoltaic power plant with four different structures of a steam electrolysis system for hydrogen production. The integration of different structures of the electrolysis process is found to result in operational penalties due to added irreversibilities, intrinsic to the electrolysis process and the reduction of the biomass plant efficiency from the extraction of low-pressure steam used to evaporate the electrolyzer feed water.

Yuferov et. al., (2015) analyzed the influence of efficiency factor and medium-entropy temperatures on the specific mass of a space power plant. It has been found that sensitivity functions represent criteria relations that define the optimality and similarity range for space nuclear power plants. A form has been proposed for recording the specific characteristics with explicit interrelationships between target functions and design variables. Atsonios et. al., (2015) investigated and optimized the lignite pre-drying concepts. The main process parameters examined have been the heat source for drying and the respective drying medium. Different concepts have also been examined which include the utilization of preheated air as heating medium and the optimized integration of a heat pump as a heat source for the drying process. Xu Liang and Jingqi (2015) developed a generic approach to calculate the thermodynamic properties of the flue gases online, based on its composition estimation. Mohammadi et. al., (2015) performed the energy and exergy analyses of boiler blow down heat recovery on a steam power plant in Iran. Two different optimization algorithms including GA and PSO have been established to increase the plant efficiency. The decision variables which have been taken under consideration include extraction pressure from steam turbine and temperature and pressure of boiler outlet stream.

Fu et. al., (2015) performed a systematic study on direct combustion coal to power processes with respect to thermodynamic, technical and economic factors. Traditional exergy analysis focuses on irreversibilities in existing processes whereas the new methodology investigates the thermal efficiency from its theoretical maximum to practical values by adding various irreversibilities one by one. Liu et. al., (2015) designed a lignite-fired power plant integrated with a vacuum dryer, by thermodynamically analyzing a reference case of a 1000 MW power plant. The results of the study show that the net efficiency of power plant can be increased if a low-pressure steam or heat pump is used to provide the drying heat source. Hagi et. al., (2015) proposed a full integration procedure which is suited for both new built and retrofit coal-fired power plants by means of easy-to-use correlations, which link heat demand to production loss and waste heat availability to production increase, taking their exergy content into account. This approach provides an analytical tool which allows a quick and realistic evaluation of a given concept or process layout, without the need of a detailed full power plant model. Olaleye et. al., (2015) performed steady state simulation and exergy analysis of supercritical coal-fired power plant (SCPP) integrated with post-combustion CO_2 capture (PCC). The analyses show that the once-through boiler exhibits the highest exergy destruction. The turbine subsystems show lower exergy destruction compared to the boiler subsystem but more significance in fuel-saving potentials of the system. Four cases of the integrated SCPP-CO_2 capture configuration have been considered for reducing thermodynamic irreversibilities in the system. This study shows that improvement in turbine performance design and the driving forces responsible for CO_2 capture can help improve the rational efficiency of the integrated system.

Yang et. al., (2015) proposed a new conceptual boiler cold-end design integrated with the steam cycle in a 1000 MW CFPP, in which the preheating of air was divided into high-temperature air pre- heater (HTAP), main air pre- heater (MAP). In the proposed boiler cold-end design, the flue gas waste heat was not only used to heat condensed water, but also to further preheat the combustion air. The air temperature at the air pre- heater outlet increases and part of the steam bleeds with high exergy can be saved. Promes et. al., (2015) investigated the steady state operation and performance of a 253 MW Integrated Coal Gasification Combined Cycle (IGCC) that is based on the design and operating parameters of the existing Willem-Alexander plant in Buggenum, the Netherlands. For performance optimization of such plants, an extensive base case model of the IGCC has been developed and validated with actual process data. The model accurately predicts the mass flows, temperatures and pressures. Exergy losses are experienced to the greatest extent in the gasifier and the combustion chamber of the gas turbine which indicates that efficiencies of mature IGCC's could be further optimized.

Saghafifar and Gadalla (2015) (a) performed a comparative analysis to signify the advantages and disadvantages of Maisotsenko gas turbine cycle (MGTC) as compared to humid air gas turbine cycles. MGTC performance is evaluated based on a simple recuperated gas turbine cycle. Further, sensitivity analysis has been done to investigate the effect of different operating parameters on the overall cycle performance. Hafdhi et. al., (2015) performed the energetic and exergetic analysis for a steam turbine power plant of an existing Phosphoric Acid Factory. A numerical code has been established using EES software to perform the calculations required for the thermal and exergy plant analysis. The main sources of irreversibility found are the melters, followed by the heat exchangers, the steam turbine generator and the pumps. The maximum energy efficiency is obtained for the blower followed by the heat exchangers, the deaerator and the steam turbine generator. The effects of high pressure steam temperature and pressure on the steam turbine generator energy and exergy efficiencies have been analyzed.

Mletzko and Kather (2014) worked on a conventional natural gas-fired combined cycle plant with a reheat gas turbine and presented a modified version for oxy-fuel operation. They deduced that higher pressure ratio results in a higher net efficiency of the plant. Mohapatra & Sanjay (2014) compared the impact of two different methods of inlet air cooling (vapor compression and vapor absorption cooling) integrated to a cooled gas turbine based combined cycle plant. Airfilm cooling has been adopted as the cooling technique for gas turbine blades. A parametric study of the effect of compressor pressure ratio, compressor inlet temperature, turbine inlet temperature, ambient relative humidity and ambient temperature on performance parameters of plant has been carried out. Optimal values of above mentioned thermodynamic parameters have been obtained. Memon et. al., (2014) modeled a gas turbine cycle to investigate the effects of important operating parameters like compressor inlet temperature (CIT), turbine inlet temperature (TIT) and pressure ratio (PR) on the overall cycle performance and CO_2 emissions. Effects of these thermodynamic parameters have been investigated on the exergy destruction and exergy efficiency of the cycle components. Multiple polynomial regression models have been developed to correlate the response variables and predictor variables. The operating parameters have then been optimized.

Oko and Wang (2014) developed a detailed dynamic model of a 500 MW coal-fired subcritical power plant using gPROMS based on first principles. This model is able to predict plant performance reasonably from 70% load level to full load. Analysis showed that implementing load changes through ramping introduces less process disturbances than step change. This model is useful for providing operator training and for process troubleshooting among others. Wang et. al., (2014) proposed that the design trade-offs between thermodynamics and economics of energy conversion systems can be more effective by combining a superstructure and mixed-integer non-linear programming (MINLP) techniques. This idea was successfully applied to supercritical coal-fired power plants to investigate the economically-optimal designs at each efficiency level. Arriola-Medellín et. al., (2014) applied the combined pinch and exergy approach to analyze the operation and design of a typical steam power plant. It quantifies the total, avoidable an unavoidable exergy loss for the equipment, which means, the potential for equipment improvement. On the other hand, the analysis of cross pinch heat transfer in the process identifies additional losses of energy due to the inefficient design of the heat recovery system.

Liu et. al., (2013) proposed that Lignite pre-drying seems to be an attractive way to tackle issues such as a low plant thermal efficiency, a high investment in construction of the lignite-fired power plant, etc. They performed a thermodynamic analysis of two pre-drying methods (both boiler flue gas drying and steam drying). Results show that both pre-drying methods can improve the plant thermal efficiency. Geete and Khandelwala (2013) performed the thermodynamic analysis of 120 MW thermal power plant at particular inlet pressure (124.61 bar) and at different inlet temperatures The correction curves for power and heat rate have been generated for combined effect of inlet pressure and different inlet temperatures. Aydin (2013) developed the exergetic sustainability indicators in order to determine sustainability aspects of gas turbine engine (GTE) based power plant. Steam turbine cycle results in improvement of overall efficiency and reviewed exergetic sustainability indicators evidently. Decrease of waste exergy ratio leads to decrease of environmental effect factor and increase both exergetic efficiency and exergetic sustainability index.

Silva et. al., (2011) developed a thermodynamic information system for diagnosis and prognosis of an existing power plant. This system is based on an analytic approach that informs the current thermodynamic condition of all cycle components. The effects induced by components anomalies and repairs in other components efficiency have been taken into consideration owing to the use of performance curves and corrected performance curves together with the thermodynamic data collected from the distributed control system. Xu et. al., (2011) presented a theoretical framework for the energy analysis and exergy analysis of the solar power tower system using molten salt as the heat transfer fluid. Energy losses and exergy losses in each component and in the overall system are evaluated to identify the causes and locations of the thermodynamic imperfection. Kaushik et. al., (2011) presented a comparison of energy and exergy analyses of thermal power plants stimulated by coal and gas. The comparison provides a detailed review of different studies on thermal power plants over the years. This review also throws light on the scope for further research and recommendations for improvement in the existing thermal power plants.

Regulagadda et. al., (2010) performed thermodynamic analysis of a subcritical boiler–turbine generator for a 32 MW coal-fired power plant. A parametric study is conducted for the plant under various operating conditions, including different operating pressures, temperatures and flow rates, in order to determine the parameters that maximize plant performance. They calculated that boiler and turbine irreversibilities yield the highest exergy losses in the power plant. Environmental impact and sustainability analysis have also been performed and presented.

Godoy et. al., (2010) obtained the optimal designs for a CCGT power plant for a wide range of power demands and different values of the available heat transfer area. These thermodynamic optimal solutions have found within a feasible operation region by means of a non-linear mathematical programming (NLP) model, where decision variables can vary freely. Technical relationships among them have been used to systematize the optimal values of design and operative variables for a CCGT power plant into optimal solution sets.

Gupta and Kaushik (2010) presented the energy and exergy analysis for the different components of a proposed conceptual direct steam generation (DSG) solar–thermal power plant (STPP). It has been observed that the maximum energy loss is in the condenser followed by solar collector field. The maximum exergy loss is in the solar collector field while in other plant components it is small. Liu et. al., (2010) evaluated the power-generation efficiency of major thermal power plants in Taiwan during 2004–2006 using the data envelopment analysis (DEA) approach. A stability test was conducted to verify the stability of the DEA model. The most important variable in the DEA model is the ‘‘heating value of total fuels’’.

Ray et. al., (2010) performed the exergy analysis of a 500 MW steam turbine cycle of an operating power plant under the design and off-design conditions with different degrees of superheat and reheat sprays. The analysis helps in identification of the contribution of individual equipment in the overall increase of exergy destruction under off-design condition. Exergy analysis has also been performed using off-line performance guarantee (PG) tests conducted before and after a unit overhauling. Pre-overhauling exergy efficiency figures of the major cycle equipment have been compared with their respective design values to assess the need and extent of maintenance work, whereas post-overhaul exergy data has been used to quantify the compliance with the guaranteed performance.

Ganpathy et. al., (2009) presented the exergy analysis to identify the magnitudes and the locations of real energy losses, in order to improve the existing systems, processes or components. The exergy losses occurred in the various subsystems of the plant and their components have been calculated using the mass, energy and exergy balance equations. The distribution of the exergy losses in several plant components during the real time plant running conditions has been assessed to locate the process irreversibility. Aljundi (2009) performed the energy and exergy analysis of Al-Hussein power plant in Jordan. He analyzed the system components separately to identify and quantify the sites having largest energy and exergy losses. The effect of varying the reference environment state on this analysis has also been presented. The performance of the plant has been estimated by a component wise modeling and a detailed break-up of energy and exergy losses for the considered plant. For a moderate change in the reference environment state temperature, no drastic change has been noticed in the performance of major components and the main conclusion remained the same; the boiler is the major source of irreversibilities in the power plant. Chemical reaction is the most significant source of exergy destruction in a boiler system which can be reduced by preheating the combustion air and reducing the air–fuel ratio. Erdem et. al., (2009) analyzed the performance of nine thermal power plants under control governmental bodies in Turkey, from energetic and exergetic viewpoint. The considered power plants are fed by low quality coal. Thermodynamic models of the plants have been developed based on first and second law of thermodynamics. Energetic simulation results of the developed models are compared with the design values of the power plants in order to demonstrate the reliability. Design point performance analyses based on energetic and exergetic performance criteria have been performed for all considered plants. The main sources of thermodynamic inefficiencies as well as reasonable comparison of each plant to others have been identified.

Oktay (2009) calculated the exergy efficiencies, irreversibilities, and improvement factors of turbine, steam generator and pumps for the selected plant in Turkey. Comparison between conventional and fluidized bed power plant has been made and improving techniques have also given for conventional plants. Kelly et. al., (2009) proposed that the exergy destruction occurring within a component can be split into two parts: (a) endogenous exergy destruction due exclusively to the performance of the component being considered and (b) exogenous exergy destruction caused also by the inefficiencies within the remaining components of the overall system.

Som and Dutta (2008) made a comprehensive review pertaining to fundamental studies on thermodynamic irreversibility and exergy analysis in the processes of combustion of gaseous, liquid and solid fuels. The need for such investigations in the context of combustion processes in practice has been stressed upon and then the various approaches of exergy analysis and the results arrived at by different research workers in the field have been discussed. It has been observed that the major source of irreversibilities is the internal thermal energy exchange associated with high temperature gradients caused by heat release in combustion reactions.

Kanoglu et. al., (2007) presented an extensive overview of various energy- and exergy-based efficiencies used in the analysis of power cycles. Vapor and gas power cycles, cogeneration cycles and geothermal power cycles have been examined in the study, and consideration has been given to different cycle designs. The many approaches that can be used to define efficiencies are provided and their implications have been discussed. Improvements of the management of energy in power plants that stem from understanding the efficiencies better have been described. Utlu and Hepbasli (2007) presented the energy and exergy utilization efficiencies in the Turkish utility sector over a period from 1990 to 2004. Energy and exergy analyses have been performed for eight power plant modes which are based on the actual data over the period studied. Sectoral energy and exergy analyses have been conducted to study the variations of energy and exergy efficiencies for each power plant through the years, and overall energy and exergy efficiencies have been compared for these power plants.

Chen et. al., (2004) performed a performance analysis and optimization of a open-cycle regenerator gas-turbine power-plant. The analytical formulae for the relation between power output and cycle overall pressure-ratio have been derived by taking into account, the eight pressure-drop losses in the intake, compression, regeneration, combustion, expansion and discharge processes and flow process in the piping, the heat-transfer loss to the ambient environment, the irreversible compression and expansion losses in the compressor and the turbine, and the irreversible combustion loss in the combustion chamber. The power output has been optimized by adjusting the mass-flow rate and the distribution of pressure losses along the flow path.

2.3 ECONOMIC ANALYSIS OF POWER PLANTS

Manzolini et. al., (2015) assessed the economic advantages of an innovative solvent for CO_2 capture on state-of-the-art solvents. The CESAR-1 solvent, which is an aqueous solution of 2-amino-2-methylpropanol (AMP) and piperazine (PZ), is applied both to advanced supercritical pulverised (ASC) coal and natural gas combined cycle (NGCC) power plants with post-combustion CO_2 capture units. The methodology includes process model developments using commercial simulation programs, which determine the thermodynamic properties of the selected power plants and the performance of the CO_2 capture units. Barigozzi et. al., (2015) conducted a techno-economical parametric analysis of an inlet air cooling system applied to an aero-derivative Gas Turbine (GT) for a combined cycle power plant (CC). A 55 MW combined cycle power plant with a GE LM6000 gas turbine was assumed as a reference case. Operational hours and power output augmentation were higher in hotter climates; wet climates required huge thermal storages, thus increasing the investment cost. The best techno-economic performance is attained for sites with high temperature combined with low relative humidity, typical of desert areas. The parametric analysis showed that the size of cooling storage is a very important parameter for the economical revenue.

Brouwer et. al. (2015) quantified and compared the technical and economic performance of power plants for four distinctly different future scenarios. They observed that future low-carbon power systems will have large shares of intermittent renewable sources (19–42%) and also a 2–38% higher variability in residual load compared to the baseline scenario. Hence the power plant operation will be more variable which will reduce their efficiency by 0.6–1.6% compared to the full-load efficiency. Enough flexibility is present in future power systems for accommodation of renewables due to advances in power plant flexibility and interconnectors. As a result, generators with CCS have a large market share (23–64% of power generated)

Osikowska et. al. (2015) built detailed thermodynamic models of oxy-fuel power plants with gross power of approximately 460 MW. For the selected structure of the system, an economic analysis of the solutions was developed. This analysis accounts for different scenarios of the functioning of the Emission Trading Scheme and includes detailed estimates of the investment costs in both cases. As an indicator of profitability, the break-even price of electricity was used primarily. A system with a hybrid air separation unit has slightly better economic performance. The break-even price of electricity in this case is approximately 3.4 €/MW h less than for the system with a cryogenic unit. Bolatturk et. al. (2015) performed the exergy and thermoeconomic analyses of Turkey-based Cayırhan thermal power plant. Thermodynamic properties of the inlet and outlet points of each unit in thermal plant have been specified via EES package program. Thermal and second law efficiencies of thermal power plant have been found respectively as 38% and 53%. In the thermal power plant, the highest amounts of exergy losses are found in; the boiler, turbine groups, condenser, heater group and pump groups. The highest amount of exergy loss costs have been seen respectively in boiler, turbine group and condenser. Elsafi (2015) demonstrated the exergy and exergoeconomic analysis of commercial-size direct steam generation parabolic trough solar thermal power plant. For steam power cycles, reheating might be necessary to avoid the wetness of steam which shortens the lifetime of the turbines. The non-reheating configuration as well as reheating by steam–steam heat exchanger has been considered. For each component, exergy and exergy-costing balance equations have been formulated based on a proper definition of fuel–product–loss.

Zare (2015) investigated and compared the performance of three configurations of Organic Rankine cycle (ORC) for binary geothermal power plants from the viewpoints of both thermodynamics and economics. To assess the cycles' performances, thermodynamic and exergoeconomic models have been developed and a parametric study has been carried out prior to the optimization with respect to the total product cost minimization, as the objective function. Also, a profitability evaluation of the investigated systems is performed based on the total capital investment and payback period. Guandalini et. al. (2015) analyzed the potential of a grid balancing system based on different combinations of traditional gas turbine based power plants. Power-to-gas is a promising solution to balance the electric grid, based on water electrolysis, which can effectively contribute to reducing the uncertainty of dispatch plans. Different economic scenarios have been assessed, leading to a set of optimal sizes of the proposed system, using a statistical approach in order to estimate wind farm productivity and forecasting errors, as well as each component load conditions. Xu et. al. (2015) proposed a partially-underground tower type boiler design, which has nearly half of the boiler embedded underground, thereby significantly reducing the boiler height and steam pipeline lengths. Thermodynamic and economic analyses have been conducted on a 1000 MW advanced double reheat steam cycle. Results show that compared to the reference power plant, the power plant with the proposed tower-type boiler design could reduce the net heat rate by 18.3 kJ/kWh and could reduce the cost of electricity (COE) by $0.60/MWh.

Khorshidi et. al. (2015) considered the different configurations of auxiliary units to partially or totally meet the energy requirements for MEA post-combustion capture in a 500 MW sub-critical coal-fired plant. The auxiliary unit is either a boiler or a combined heat and power (CHP) unit, providing both heat and electricity. Using biomass in auxiliary units, the grid loss is reduced without increasing fossil fuel consumption. The results show that using a biomass CHP unit is more favorable than using a biomass boiler both in terms of CO_2 emission reductions and also power plant economic viability. By using an auxiliary biomass CHP unit, both the emission intensity and the cost of electricity would be marginally lower than for a coal plant with capture. Emission reductions can also occur if CO_2 is captured both from the coal plant and the auxiliary biomass CHP, which results in negative emissions. However, high incentive schemes (or a low biomass price are required to make CO_2 capture from both the coal plant and the auxiliary biomass CHP unit economically attractive. Budisulistyo and Krumdieck (2015) presented a pre-feasibility design investigation for a binary geothermal power plant by using a typical geothermal resource in New Zealand. Thermodynamic and economic analyses have been conducted for key cycle design options. The net electrical power output (W_{net}) and the ratio of W_{net} to total Purchased Equipment Cost (PEC) have been used as the objective function to select the best thermo-economical designs.

Oyedepo et. al. (2014) conducted the performance evaluation and economic analysis of a gas turbine power plant in Nigeria for the period 2001–2010 The simple performance indicator developed to evaluate the performance indices and outage cost for the station can also be applied to other power stations. Measures to improve the performance indices of the plant have been suggested such as training of operation and maintenance (O & M) personnel regularly, improvement in O & M practices, proper spare parts inventory and improvement in general housekeeping of the plant. Tola and Pettinau (2014) compared, from the technical and economic points of view, the performance of three coal fired power generation technologies: (i) ultra-supercritical (USC) plant equipped with a conventional flue gas treatment (CGT) process, (ii) USC plant equipped with SNOX technology for the combined removal of sulphur and nitrogen oxides and (iii) integrated gasification combined cycle (IGCC) plant based on a slurry-feed entrained-flow gasifier. Technical assessment has been carried out by using simulation models implemented through Aspen Plus and Gate-Cycle tools, while the economic assessment has been performed through a properly developed simulation model. Franco and Vaccaro (2014) discussed and analyzed the perspectives of future development of geothermal power plants, mainly of small size for the exploitation of medium–low temperature reservoirs. A key element for the design of a geothermal plant for medium temperature geothermal source is the definition of the power of the plant (size): this is important in order to define not only the economic plan but the durability of the reservoir also. They proposed a method for joining energetic and economic approaches. The result of the combined energetic and economic analysis is interesting particularly in case of Organic Rankine Cycle (ORC) power plants in order to define a suitable and optimal size and to maximize the resource durability.

Cormos (2014) concluded that coal-based power generation sector is facing important changes to implement energy efficient carbon capture technologies to comply with emission reduction targets for transition to low carbon economy. He proposed CaL (Calcium Looping) as one of the innovative carbon capture options which is able to deliver low energy and cost penalties. The study evaluates how the integration of post-combustion calcium looping influences the economics of power plants providing up-dated techno-economic indicators. Coal-based combustion plants operated in both sub- and super-critical steam conditions have been evaluated in the study. As benchmark options used to quantify the carbon capture energy and cost penalties, the same power generation technologies have been evaluated without CCS (Carbon capture and storage). The power plant concepts investigated in the paper generates around 545–560 MW net power with at least 90% carbon capture rate. Introduction of CaL technology for CO_2 capture results in a 24–42% increase of specific capital investment, the O&M costs are increasing with 24–30% and the electricity cost with 39–48% (all compared to non-CCS cases).

Xuan Do et. al. (2014) evaluated and compared the economic feasibility of three different configurations of a woodchips power plant based on the circulating fluidized-bed (CFB) gasification: (1) a gas engine, (2) a gas turbine, and (3) gas & steam turbines. A comprehensive model of the power plant has been developed which employs the process simulator, Aspen Plus. The economic feasibility has been analyzed in terms of the payback period (PBP), return on investment (ROI), and discount cash flow rate of return (DCFROR).

Osikowska et. al. (2013) performed the economic analysis of different structures of the oxy-fuel system and the reference air-fired power plant by using a newly developed computational algorithm built in the Excel environment. The algorithm uses a Break Even Point (BEP) method, focusing mainly on determining a break-even price of electricity. Siefert and Litster (2013) performed the exergy and economic analyses of two advanced fossil fuel power plants configurations: an integrated gasification combined cycle with advanced H_2 and O_2 membrane separation including CO_2 sequestration (Adv. IGCC–CCS) and an integrated gasification fuel cell cycle with a catalytic gasifier and a pressurized solid oxide fuel cell including CO_2 sequestration (Adv. IGFC–CCS). The goal of the exergy analysis was to evaluate the power generation and the exergy destruction of each of the major components. They estimated the capital, labor, and fuel costs of these power plants, and then calculated the internal rate of return on investment (IRR) and the levelized cost of electricity (LCOE). In the Adv. IGFC–CCS case, we chose a configuration with anode gas recycle back to the gasifier, and then varied the SOFC pressure to find the optimal pressure under this particular configuration.

Godoy et. al. (2013) proposed a reduced model as a strategy for simplifying the resolution of the rigorous multi-period model. Trends in the system behavior have been identified, enabling the reduction of the multi-period formulation into a system of non-linear equations plus additional constraints, which allows easily computing accurate estimations of the optimal values of the design variables as well as the time-dependent operative variables. El Nasr et. al. (2013) discussed an advanced solid-based adsorption and a techno-economic evaluation methodology in order to compare the advantages of this process to the conventional process. Some indications of the expected technical and economic benefits of the process have also been discussed. Alavijeh et. al. (2013) examined the natural gas fired power plants in Iran. The characteristics of thirty two gas turbine power plants and twenty steam power plants have been considered. Their emission factor values have been compared with the standards of Energy Protection Agency, Euro Union and World Bank. Emission factors of gas turbine and steam power plants show that gas turbine power plants have better performance than the steam power plants. For economic analysis, fuel consumption and environmental damages caused by the emitted pollutants have been considered as cost functions and electricity sales revenue has been taken as the benefit function. Stopatto et. al. (2012) presented a procedure aimed at evaluating extra cost related to flexible operation, and at assisting the management decision about power plants′ operation and maintenance scheduling. The procedure, predicts the residual life of the most critical components while considering the effects of creep, thermo-mechanical fatigue, welding, corrosion and oxidation. It also permits the choice of different future strategies for plant management and evaluate the residual life and the economic effects for each of them.

Muller et. al. (2011) analyzed the heating systems of four European countries: Austria, Finland, The Netherlands, and Sweden. They concluded that the overall consumer costs for different heating options are in the same range. They derived an overall standard deviation of about 8%. Analysis demonstrates that the share of capital costs on total heating cost increases with lower exergy input. They conclude that, for the case of modern cost effective heating systems, the substitution rate between exergy and capital is in the vicinity of 2/3. This means that by reducing the average specific exergy input of the applied energy carriers by one unit, the share of capital costs on the total costs increases by 2/3 of a unit. Sciubba (2011) described a procedure that leads to the calculation of “exact” values of both econometric coefficients, based on detailed exergy- and monetary balances of the Society to which the EEA is applied. It is shown that both α and β depend on the consumption patterns, the technological level and the life- and socio-economic standards of each country. It is also shown that the values are substantially different for developed (OECD) and underdeveloped countries, and representative samples of values are calculated and critically analyzed. Uson and Valero (2011) concluded that the thermoeconomic diagnosis of operating energy intensive systems is the determination of fuel consumption, the identification of the causes of its increment from design conditions and the quantification of the effect of each one of these causes. Besides data acquisition and monitoring systems, a thermoeconomic diagnosis methodology is also needed. These methods are based on thermodynamic indicators instead of thermoeconomic parameters. The comparison is based on the diagnosis of a 3 * 350 MW coal-fired-power plant for a time span of more than 6 years. Results show that quantitative causality analysis is able to quantify the effects of all variables while the others are only suitable for the most influential ones.

Gorji- Bandpi et. al. (2010) performed the thermoeconomic analysis and exergoeconomic evaluation for a 140 MW gas turbine power plant based on the second law of thermodynamics. Cost rates of exergy streams for any state of cycle and unit cost of the final product in the power plant have been calculated. The effects of a change in the compressor pressure ratio, gas turbine inlet temperature, and turbine and compressor isentropic efficiency on the unit cost of the product have also been studied. The analysis shows the deep relation of the unit cost to the change in these parameters. Yari (2010) conducted a comparative study of the different geothermal power plant concepts, based on the exergy analysis for high-temperature geothermal resources. With respect to each cycle, a thermodynamic model has been developed. Based on the exergy analysis, a comparative study was done to clarify the best cycle configuration. The performance of each cycle has been discussed in terms of the second-law efficiency, exergy destruction rate, and first-law efficiency. Cafaro et. al. (2010) performed a study regarding the thermoeconomic monitoring of the bottoming cycle of a combined cycle power plant, using real historical data. The software is able to calculate functional exergy flows (y), their related costs (c) (using the plant functional diagram); after that non dimensional parameters for the characteristic exergonomic indices.

Zang et. al. (2007) presented a progressive separation procedure of the induced effects for power plant system diagnosis based on structural theory and symbolic thermoeconomics. The malfunction/dysfunction analysis and the fuel impact analysis of the structural theory as well as an improved induced malfunction evaluation method have been applied to a 300 MW pulverized coal fired power plant located in Yiyang (Hunan Province, China). The dysfunctions induced by the malfunctions are separated by the malfunction/dysfunction analysis from the irreversibility increases in the components. The effects of the malfunctions on each component and the whole plant have also been evaluated by using the fuel impact analysis.

Zang et. al. (2006) proposed a cost analysis method based on thermoeconomics for a 300 MW pulverized coal fired power plant located in Yiyang (Hunan Province, China). This method can provide a detailed analysis for cost formation of the power plant as well as the effects of different operating conditions and parameters on the performance of each individual component. To perform the thermoeconomic analysis of the plant, a simulator has been developed. With the thermodynamic properties of the most significant mass and energy flow streams being obtained from the plant, this simulator can reproduce the cycle behavior for different operating conditions with relative errors less than 2%. Verda (2006) analyzed the information that can be obtained by considering different accuracy levels in the thermoeconomic diagnosis problem. A progressive elimination of effects that impede a clear identification of anomalies causing performance degradation has been performed. The analysis has been applied to combined cycle plant.

Rosen and Dincer (2003) (a) investigated the relation between capital costs and thermodynamic losses for devices in modern coal fired, oil-fired and nuclear electrical generating stations. Thermodynamic loss rate-to-capital cost ratios have been used to show that, for station devices and the overall station, a systematic correlation exists between the capital cost and exergy loss but not between capital cost and energy loss or external exergy loss. The possible existence is indicated of a correlation between the mean thermodynamic loss rate-to-capital cost ratios for all of the devices in a station and the ratios for the overall station, when the ratio is based on total or internal exergy losses. Rosen and Dincer (2003) (b) developed several thermodynamic relations between energy and exergy losses and capital costs for thermal systems and equipment and applied them to a modern coal fired electrical generating station. The application considers the overall station and the following station devices: turbine generators, steam generators, preheating devices and condensers. The data suggest that an important parameter is the ratio of the thermodynamic loss rate to capital cost. The relative spread in the ratio values for different devices is seen to be large when it is based on energy loss and small when it is based on exergy loss.

Kwak et. al. (2001) performed the exergetic and thermoeconomic analyses for a 500-MW combined cycle plant. Mass and energy conservation laws were applied to each component of the system. The exergoeconomic model, which represented the productive structure of the system considered, was used to visualize the cost formation process and the productive interaction between components. The computer program developed in the study can be used to determine the production costs of power plants, such as gas- and steam-turbines plants and gas-turbine cogeneration plants.

Lior (1997) proposed that adding superheat to the nearly saturated steam increases the amount of power generated by at least 70%, the plant efficiency by at least 16%, the plant effectiveness by at least 6%, and reduces the cost of generated electricity by at least 32%. These features make fossil-fuel superheat of nuclear power plants interesting both for new plants and for retrofit of existing nuclear plants. The super heater accounts for the major portion of exergy destruction in the system excluding the reactor, with the extraction turbine taking second place, and the optimization of their combination will lead to even better system performance.

Tsatsaronis (1993) discussed the development, state-of-the-art and applications of exergy analysis and thermoeconomics (exergo-economics). The study reviews the history of exergy analysis and thermoeconomics, the performance evaluation of an energy system from the viewpoints of the second law of thermodynamics and thermoeconomics as well as applications of thermoeconomic optimization techniques.

2.4 OPTIMIZATION OF POWER PLANTS

Ruiz et. al. (2015) (a) presented a multiple criteria study about the efficiency improvement of the auxiliary services. They considered the economic investment and the net present value, as economic criteria, together with the energy saving criterion. In the multi-objective model proposed, the energy model has been validated using several measures taken in a 1100 megawatts coal power plant. Vandani et. al. (2015) performed the energy and exergy analyses of boiler blow down heat recovery. Two different optimization algorithms including GA and PSO are established to increase the plant efficiency. The decision variables are extraction pressure from steam turbine and temperature and pressure of boiler outlet stream. The optimization results show that temperature and pressure of boiler outlet stream have a higher effect on the exergy efficiency of the system in respect to the other decision variables. Ruiz et. al. (2015) (b) proposed a multi objective optimization problem to determine the most suitable strategies to maximize the energy saving, to minimize the economic investment and to maximize the Internal Rate of Return of the investment. Solving this real-life multi objective optimization problem with a decision maker presents several challenges and difficulties. The idea is to start with the approximation of the Pareto optimal set using evolutionary multi objective optimization. The next step is aiding the decision maker to explore the efficient set and to identify the subset of solutions which fits her/his preferences and finally the search is concentrated for new solutions into the most interesting part of the efficient set with the help of a preference-based evolutionary algorithm. This allows building of a flexible scheme that is progressively adapted to the decision maker's reactions until he is able to find the most preferred solution.

Guedez et. al. (2015) introduced a comprehensive methodology for designing solar tower power plants based on a thermoeconomic approach which allows the true optimum trade-off curves between cost, profitability and investment to be identified while simultaneously considering several operating strategies as well as varying critical design parameters in each of the aforementioned blocks. The methodology has been presented by means of analyzing the design of a power plant for the region of Seville. Johnston et. al. (2015) proposed a methodology to economically optimize sizing of Energy Storage Systems (ESSs) whilst enhancing the participation of Wind Power Plants (WPP) in network primary frequency control support. The methodology has been designed flexibly, so that it can be applied to different energy markets and to include different ESS technologies. The methodology includes the formulation and solving of a Linear Programming (LP) problem. Saghafifar and Gadalla (2015) (b) integrated MGTC as the bottoming cycle to a topping simple gas turbine as Maisotsenko bottoming cycle (MBC). Thermodynamic optimization has been performed to illustrate the advantages and disadvantages of MBC as compared with air bottoming cycle (ABC). Detailed sensitivity analysis has been done to present the effect of different operating parameters on the proposed configurations' performance.

Wang et. al. (2014) (a) concluded that the design trade-offs between thermodynamics and economics of energy conversion systems can be more effective by the combination of superstructure and mixed-integer non-linear programming (MINLP) techniques. The front of decision space showing the optimal sets of economic behavior and system efficiency with different corresponding optimal system structures and process variables provides additional and useful information on cost-effective design of thermal systems. Wang et. al. (2014) (b) proposed an enhanced differential evolution with diversity- preserving and density-adjusting mechanisms, and a newly-proposed algorithm for searching the decision space frontier in a single run, to conduct the multi-objective optimization of large scale, supercritical coal-fired plants. The uncertainties associated with cost functions have been discussed by analyzing the sensitivity of the decision space frontier to some significant parameters involved in cost functions.

Sowa et. al. (2014) presented a modeling approach for P2H systems, as a component of virtual power plants, with a high share of renewable energies. The operation strategies are evaluated with respect to economic and technical aspects and uncertainties in generation and load. The operation strategies of P2H systems are shown with regard to market integration of renewable energies within a virtual power plant and the provision of ancillary services. Guedez et. al. (2014) proposed the methodology for thermoeconomic optimization of solar power plants. In order to analyze the market role, three scenarios were modeled, with low, medium and high penetrations of non dispatchable renewables (i.e. wind and solar photovoltaics). The demand that cannot be met by these variable sources is met by a solar thermal power plant with heat provided either by a solar field and storage system or a back-up gas burner. For each scenario, the size of the solar field and storage were varied in order to show the trade-off between the levelized generation costs of the system. Li et. al. (2014) proposed a strategy for simplifying the resolution of the rigorous economic optimization problem of power plants based on the economic optima distinctive characteristics which are used to describe the behavior of the decision variables of the power plant on its optima. This approach results in a mathematical formulation shaped as a system of non-linear equations and additional constraints which is able to easily provide accurate estimations of the optimal values of the power plant design and operative variables.

Groniewsky (2013) concluded that the basic concept in applying numerical optimization methods for power plants optimization problems is to combine a state-of-the-art search algorithm with a powerful, power plant simulation program to optimize the energy conversion system from both economic and thermodynamic viewpoints. Improvement in the energy conversion system, by optimizing the design and operation and studying interactions among plant components, requires the investigation of a large number of possible design and operational alternatives while state-of-the-art search algorithms can assist in the development of cost-effective power plant concepts.

Godoy et. al. (2011) determined the optimal combined cycle gas turbine power plants characterized by minimum specific annual cost values by means of a non-linear mathematical programming model. As the technical optimization allows identifying trends in the system behavior and unveiling optimization opportunities, selected functional relationships are obtained as the thermodynamic optimal values of the decision variables which are systematically linked to the ratio between the total heat transfer area and the net power production. Feng et. al. (2011) developed a model by combined cooling, heating and power (CCHP) plant composed of an irreversible closed Brayton cycle and an endoreversible four-heat-reservoir absorption refrigeration cycle by using finite time thermodynamics. The irreversibilities considered in the CCHP plant include heat-resistance losses in the hot-, cold-, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers as well as non-isentropic losses in the compression and expansion processes. Based on the finite time exergoeconomic analysis method, profit rate optimization has been carried out by searching for the optimal compressor pressure ratio and the optimal heat conductance distributions of the seven heat exchangers for a fixed total heat exchanger inventory and with the help of Powell arithmetic.

Chen et. al. (2011) investigated the finite time exergoeconomic performance of a combined cooling, heating and power generation (CCHP) plant composed of one endoreversible closed Brayton cycle and one endoreversible four-heat-reservoir absorption refrigeration cycle by using finite time thermodynamics. Heat conductance distribution among hot-, cold-, thermal consumer-, generator-, absorber-, condenser- and evaporator-side heat exchangers and compressor pressure ratio have been optimized by taking the maximum profit rate as the objective. Rovira et. al.

(2011) showed a methodology to achieve thermoeconomic optimizations of CCGT power plants taking into account the frequent off-design operation of the plant. The aim of the work is to improve the thermoeconomic design of the power plant by means of considering a more realistic annual operation scenario. The methodology has been applied to optimize several CCGT configurations operating under different scenarios of energy production.

Spelling et. al. (2011) developed a dynamic model of a pure-solar combined-cycle power plant in order to allow the determination of the thermodynamic and economic performance of the plant. The model has been used for multi-objective thermoeconomic optimization of both the power plant performance and cost, using a population-based evolutionary algorithm. In order to examine the trade-offs that must be made, two conflicting objectives have been considered i.e. minimal investment costs and minimal levelized electricity costs. Ahmadi and Dincer (2011) developed a multi-objective optimization approach in which certain exergetic, economic and environmental parameters have been considered through two objective functions, including the gas turbine exergy efficiency, total cost rate of the system production including cost rate of environmental impact. The thermoenviroeconomic objective function is minimized while the power plant exergy efficiency is maximized using a power full developed genetic algorithm. The results of optimal designs are obtained as a set of multiple optimum solutions, called 'the Pareto optimal solutions'. Ploumen et. al. (2011) carried out the thermodynamic analysis with KEMA's flow sheeting package SPENCE®. In all considered cases the thermal input has been taken as 2400 MWth. Improvement of efficiency of coal fired power station technology can reduce the amount of CO_2 emitted significantly. The study shows that with current available technology and improvements an additional emission reduction of almost 10% can be realized by applying the USC 700 + MC. Compared to the world wide average an emission reduction of 66% can be achieved without CCS.

Dipama et. al. (2010) presented a multi-objective optimization method that permits solutions that simultaneously satisfy multiple conflicting objectives to be determined. The optimization process is carried out by using an evolutionary algorithm developed around an innovative technique that consists of partitioning the solution search space into parallel corridors. Within these corridors, ''header'' solutions are trapped to be then involved in a reproduction process of new populations by using genetic operators. Godoy et. al. (2010) determined the optimal designs of a CCGT power plant characterized by maximum second law efficiency values for a wide range of power demands and different values of the available heat transfer area. These thermodynamic optimal solutions have been found within a feasible operation region by using a non-linear mathematical programming (NLP) model, where decision variables can vary freely. Technical relationships among them have been used to systematize the optimal values of design and operative variables into optimal solution sets, named as optimal solution families.

Wang et. al. (2009) examined the exergy analysis for a cogeneration system, and a parameter optimization for each cogeneration system is achieved by means of genetic algorithm (GA) to reach the maximum exergy efficiency. The optimum performances for different cogeneration systems are compared under the same condition. Results of the study show that the exergy losses in turbine, condenser, and heat recovery vapor generator are relatively large, and reducing the exergy losses of these components could improve the performance of the cogeneration system.

Sahoo (2008) optimized a cogeneration system that produces 50 MW of electricity and 15 kg/s of saturated steam at 2.5 bar using exergoeconomic principles and evolutionary programming. The analysis shows that the product cost, cost of electricity and steam, is 9.9% lower with respect to the base case. This is achieved, however, with 10% increase in capital investment. Moreover, it is important to note that the additional investment can be paid back in 3.23 years.

Palazzi et. al. (2007) presented a thermo-economic optimization method that systematically generates the most attractive configurations of an integrated system. In this methodology, the energy flows are computed using conventional process simulation software. The system is integrated using the pinch based methods which rely on optimization techniques. This defines the minimum of energy required and sets the basis to design the ideal heat exchanger network. A thermo-economic method is then used to compute the integrated system performances, sizes and costs. This allows performing the optimization of the system with regard to two objectives: minimize the specific cost and maximize the efficiency.

Li et. al. (2006) performed the thermo-economic optimization of a tri- generation system by simultaneous consideration of the thermodynamic, economic and emission criteria regarding both CO2 and NOx. Technologies such as gas turbine, internal combustion engine, absorption chiller and gas boiler have been considered as options of the plant optimum configuration. System Net Present Value (NPV) has been taken as the objective to be maximized.

Durmayaz et. al. (2004) reviewed the optimization studies of thermal systems, that consider various objective functions, based on finite-time thermodynamics and thermoeconomics. They concluded that the irreversibilities originating from finite-time and finite-size constraints are important in the real thermal system optimization. Since classical thermodynamic analysis based on thermodynamic equilibrium do not consider these constraints directly hence it is necessary to consider the energy transfer between the system and its surroundings in the rate form. Chen et. al. (2004) performed the performance analysis and optimization of a open-cycle regenerator gas-turbine power-plant. The analytical formulae for the relation between power output and cycle overall pressure-ratio have been derived taking into account the eight pressure-drop losses in the intake, compression, regeneration, combustion, expansion and discharge processes and flow process in the piping, the heat-transfer loss to the ambient environment, the irreversible compression and expansion losses in the compressor and the turbine, and also the irreversible combustion loss in the combustion chamber. The power output has been optimized by adjusting the mass-flow rate and the distribution of pressure losses along the flow path.

Silviera and Tuna (2003) presented a thermoeconomic analysis of cogeneration plants, applied as a rational technique to produce electric power and saturated steam. The main aim of this new methodology is to find out the minimum Exergetic Production Cost (EPC), based on the Second Law of Thermodynamics. The variables selected for the optimization were the pressure and the temperature of the steam leaving the boiler in the case of using steam turbine, and the pressure ratio, turbine exhaust temperature and mass flow in the case of gas turbines. Valdes et. al. (2003) proposed a possible way to achieve a thermoeconomic optimization of combined cycle gas turbine (CCGT) power plants. The optimization has been done using a genetic algorithm, which has been tuned applying has been applied to a single pressure CCGT power plant. Once tuned, the optimization algorithm has also been used to evaluate more complex plants, with two and three pressure levels in the heat recovery steam generator (HRSG). The variables considered for the optimization were the thermodynamic parameters which establish the configuration of the HRSG. Two different objective functions have been proposed: one minimizes the cost of production per unit of output and the other maximizes the annual cash flow.

Valero et. al. (2002) presented a complete thermoeconomic diagnosis based on the structural theory of thermoeconomics. The methodology is applied to the Escucha power plant, which is a 160 MW conventional coal fired power plant sited in Aragon (Spain). The methodology is validated using a specific simulator of the Escucha power plant. The simulator reproduces the cycle behavior for different operating conditions, either in design or in off design conditions. The error has been found to be lower than 1% in most of cases. The simulated results, i.e. temperatures, pressures, mass flow rates, power and so on, are considered as plant measured and validated values.

Habib et. al. (1999) presented a first and second-law procedure for the optimization of the reheat pressure level in reheat regeneration thermal-power plants. The procedure is general in form and has been applied for a thermal-power plant having two reheat pressure levels and two open-type feed- water heaters. The second-law efficiency of the steam generator, turbine cycle and plant have been evaluated and optimized. The irreversibilities in the different components of the steam generator and turbine cycle sections have been evaluated and discussed. El- Sayed (1999) concluded that raising the efficiency cost-effectively (Thermoeconomics) is a multi-disciplinary problem in which thermodynamics interfaces other disciplines of knowledge which in this particular case are design, manufacture and economics. He developed a communication/optimization strategy, via the concept of costing equations, using which, the system can be analyzed and optimized for minimum cost within the domain of thermodynamics.

2.5 RESEARCH GAP

After going through the available literature, it has been found that lot of work has been done by various researchers on the thermodynamic aspects of different types of power plants, very little work has been done on the economic aspects, especially, considering the second law of thermodynamics. Even less work has been done to achieve the optimization of plant performance from exergoeconomic viewpoint. Hence this study has been done to combine the thermodynamic and economic aspects of plant performance to achieve the optimal state.

2.6 OBJECTIVE OF THE THESIS

Any system design and analysis combines the use of principles from different fields such as thermodynamics, heat and mass transfer, fluid mechanics and mechanical design. In the current study, thermodynamic aspect of system design has been taken care of.

In the current study a coal fired thermal power plant and an open cycle gas turbine power plant have been considered for analysis. The analysis done in this study includes thermal analysis, exergetic cost based exergoeconomic analysis and optimization of the different components of the plant.

Current work deals with the following:

- Development of methods to reduce the heat rate of existing power plants.
- Preparation of case studies quantifying heat rate reductions resulting from the methods described herein.
- Survey of existing plants and published literature to assess heat rate reductions typically achieved in the existing coal based power plants in India.
- Maximizing the overall exergetic efficiency of the plant.
- Development of order-of-magnitude capital, fixed, and variable operations and maintenance (O&M) cost estimates for the modifications associated with the methods described for typical 210 MW coal plants.
- Minimizing the cost of generated power.
- To perform the detailed first law & second law based exergy analysis of cycles to get Proper insight from thermodynamics point of view and to point out the expected energy saving by identifying the exergy losses in the components of the system.

- To perform the exergo- economic or thermo- economic analysis of the power plant.
- To study the effects of decision variables on exergy, exergetic cost and thermo- economic variables.
- Optimization of thermo- economic variables.

CHAPTER- 3

THERMODYNAMIC ANALYSIS OF COAL FIRED TEHRMAL POWER PLANT

3.1 INTRODUCTION

The utility electricity sector in India had an installed capacity of 303 GW as of 31 May 2016. Renewable Power plants constituted 28% of total installed capacity and Non-Renewable Power Plants constituted the remaining 72%. The gross electricity generated by utilities is 1,106 TWh (1,106,000 GWh) and 166 TWh by captive power plants during the 2014–15 fiscal. The gross electricity generation includes auxiliary power consumption of power generation plants. India became the world's third largest producer of electricity in the year 2013 with 4.8% global share in electricity generation surpassing Japan and Russia.

During the year 2014-15, the per capita electricity generation in India was 1,010 kWh with total electricity consumption (utilities and non utilities) of 938.823 billion or 746 kWh per capita electricity consumption. Electric energy consumption in agriculture was recorded highest (18.45%) in 2014-15 among all countries. The per capita electricity consumption is lower compared to many countries despite cheaper electricity tariff in India.

The total installed power generation capacity in the country, which is the sum of utility capacity, captive power capacity and other non-utilities, is illustrated in Table 3.1 (https://en.wikipedia.org/wiki/Electricity_sector_in_India accessed on 20/12/2015.).

Statistics show that India is an energy deficient country and drastic measures need to be undertaken to overcome this problem. One way of doing that is to analyze the existent plants from thermodynamic and economic viewpoint for better performance and efficient utilization of resources.

Table 3.1 Power Generation Capacity in India over the years

Installed Capacity as on	Thermal (MW)				Nuclear (MW)	Renewable (MW)			Total (MW)	% Growth (on yearly basis)
	Coal	Gas	Diesel	Sub-Total Thermal		Hydel	Other Renewable	Sub-Total Renewable		
31-Dec-1947	756	-	98	854	-	508	-	508	1,362	-
31-Dec-1950	1,004	-	149	1,153	-	560	-	560	1,713	8.59%
31-Mar-1956	1,597	-	228	1,825	-	1,061	-	1,061	2,886	13.04%
31-Mar-1961	2,436	-	300	2,736	-	1,917	-	1,917	4,653	12.25%
31-Mar-1966	4,417	137	352	4,903	-	4,124	-	4,124	9,027	18.80%
31-Mar-1974	8,652	165	241	9,058	640	6,966	-	6,966	16,664	10.58%
31-Mar-1979	14,875	168	164	15,207	640	10,833	-	10,833	26,680	12.02%
31-Mar-1985	26,311	542	177	27,030	1,095	14,460	-	14,460	42,585	9.94%
31-Mar-1990	41,236	2,343	165	43,764	1,565	18,307	-	18,307	63,636	9.89%
31-Mar-1997	54,154	6,562	294	61,010	2,225	21,658	902	22,560	85,795	4.94%
31-Mar-2002	62,131	11,163	1,135	74,429	2,720	26,269	1,628	27,897	105,046	4.49%
31-Mar-2007	71,121	13,692	1,202	86,015	3,900	34,654	7,760	42,414	132,329	5.19%
31-Mar-2012	112,022	18,381	1,200	131,603	4,780	38,990	24,503	63,493	199,877	9.00%
31 Mar 2015	169,118	23,062	1,200	188,898	5,780	41,267	35,777	77,044	271,722	10.8%
31 Mar 2016	185,172	24,508	993	210,675	5,780	42,783	@ 42,727	85,510	301,965	11.13%

The sector wise breakup of the power generation in India is given in Table 3.2

Table 3.2 Sector wise breakup of Power Generation in India

Sector	Coal	Gas	Diesel	Total	Nuclear	Hydro	RES	Grand Total (MW)
Central	48,130.00	7,519.73	0	55,649.73	5,780.00	11,091.43	0	72,521.16
State	58,100.50	6,974.42	602.61	65,677.53	0	27,482.00	3,803.67	96,963.20
Private	58,405.38	8,568.00	597.14	67,570.52	0	2,694.00	31,973.29	102,237.81
All India	164,635.88	23,062.15	1,199.75	188,897.78	5,780.00	41,267.43	35,776.96	271,722.17

Various techniques have been developed to analyze the performance of power plants. Most of these techniques are based on the First Law of Thermodynamics. However, these techniques pose a major problem that they are not able to analyze the performance of a power plant effectively. Further it is almost impossible to assign monetary values to energy which is the most important parameter in any technique based on the First Law of Thermodynamics. Attempts to assign monetary values to energy have led to grave miscalculations.

Hence there has been a growing interest among the scientific community to use concepts of exergy and exergy destruction in order to evaluate the performance of a power plant. The exergy balance equations allow us to locate the irreversibilities within a thermal system and also allow us to ascertain the reason for their effects on the overall efficiency of the power plant. The main purpose of the exergy analysis is to detect and evaluate quantitatively the causes of thermodynamic imperfections in a thermal system. Exergy based analysis allows us to quantify the loss of efficiency in a process due to loss of quality of energy. This allows us to identify the critical components within a thermal system which provide maximum scope for improvement.

Exergy analysis provides information about the components within a thermal system where improvement is possible. However it does not say anything about whether it is practical to achieve that improvement. This problem is solved by combining economics with the usual exergy analysis.

3.2 SYSTEM DESCRIPTION

The power plant considered in the present study consists of a single unit of 210 MW at full load. Figure 3.1 shows the schematic layout of a 210 MW power plant. Boiler for this unit received coal of 16765 KJ/ kg with ash content of 38% and 26% volatile matter. The plant has High Pressure (HP) and Low Pressure (LP) turbines with reheat. Steam after expansion in the LP is fed to the condenser. The condensed steam passes through a series of HP and LP regenerative feed water heaters.

The power generation cycle has been analyzed keeping the following assumptions:

1. Gross calorific value of fuel has been used in calculations.
2. Specific exergy of fuel has been calculated as per Bejan et. al. (1996).
3. Intensive properties of the environment have been taken as constant.

3.3 ENERGY AND EXERGY ANALYSIS OF COAL FIRED THERMAL POWER PLANT

The analysis of the 210 MW coal based thermal power plant has been done. Mass and energy conservation laws have been applied to each component and also for whole system. The energy and exergy analysis for the base condition of 210 MW has been performed to pinpoint the location and magnitude of process irreversibilities. Further, the deviation of first and second law efficiency has been estimated. The major streams entering and leaving the components of the plant are shown in the Figure 3.2.

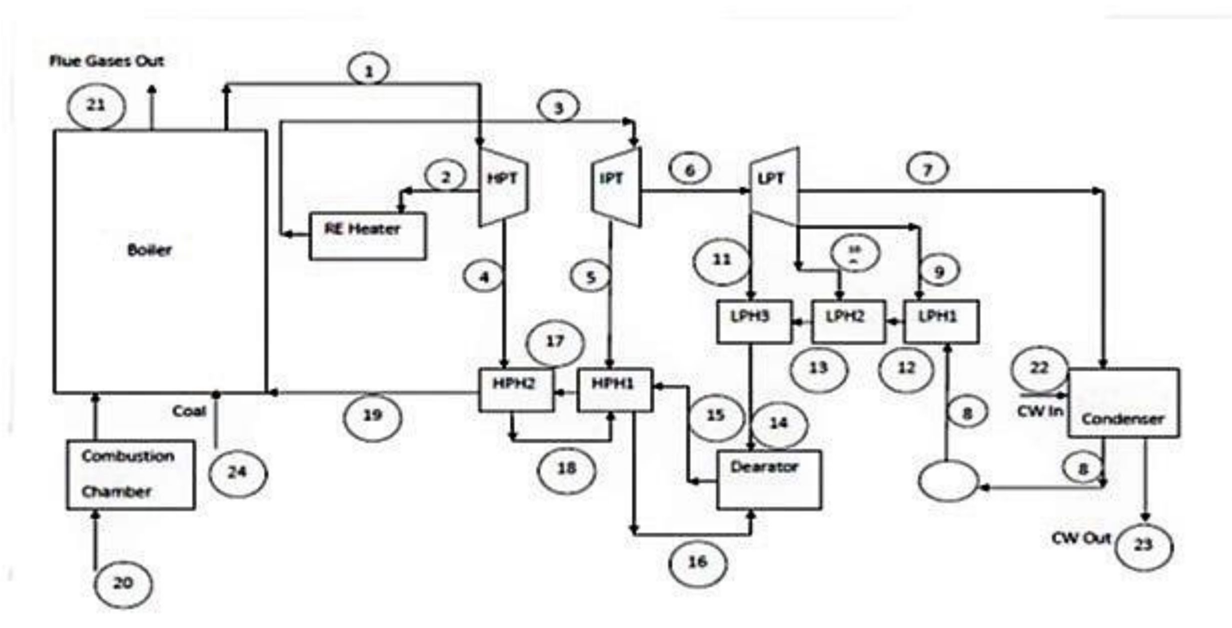


Figure 3.1 Schematic Layout of a 210 MW Coal Fired Thermal Power Plant

For analysis control volume technique has been employed which breaks the entire plant into separate control volumes with each component being considered separately along with its input and output streams. It allows us to ascertain the individual contribution of various components toward the gross irreversibility of the plant. A general energy and exergy-balance equations, applicable to any component of a thermal system can be formulated using the first and second law of thermodynamics. The specific thermo-mechanical exergy (neglecting kinetic and potential energy) is evaluated as per Bejan et. al. (1996) and is given by Equation (3.1).

$$e_j = (h - h_0) - T_0(s_j - s_0) \tag{3.1}$$

The total rate of exergy associated with any stream is calculated as per Equation (3.2).

$$\dot{E}_j = \dot{m}_j e_j \tag{3.2}$$

The total rate of energy associated with any stream is calculated as per Equation (3.3).

$$\dot{B}_j = \dot{m}_j h_j \tag{3.3}$$

where *m*, *j*, *h*, *s* and *0*, represent the mass flow rate of the stream, energy or exergy flow streams entering or leaving the system, specific enthalpy, specific entropy and thermodynamic properties at ambient conditions respectively.

The detailed exergy analysis of a thermal system includes the calculation of exergy destruction and the exergy loss in each component of the system. The exergy balance equation for any component is given by Equation (3.4).

$$\sum_e E_{e.k} + W_k = E_{q,k} + \sum_i E_{i,k} \tag{3.4}$$

where subscripts *e, i, k and q* represent the exit, inlet component and heat transfer respectively. *E* and *W* represent the exergy rate and work transfer rate.

The exergy destruction for a particular component is calculated from the exergy balance equation is given by Equation (3.5).

$$\dot{E}_D = \sum_i E_{i,k} - \sum_e E_{e.k} \tag{3.5}$$

The exergy destruction ratio for the kth component is calculated as per Equation (3.6).

$$y_{D,k} = \dot{E}_{D,k} / \sum \dot{E}_{D,k} \tag{3.6}$$

Thermodynamic analysis for the 210 MW coal fired thermal power plant has been done as per Figure 3.1. Energy and exergy calculations for various streams entering and leaving the system are given in Table 3.3.

Table 3.3 Pressure, temperature, mass flow rate, energy and exergy flow rates for different streams of the power plant

STREAM NO	P	T	m	h	s	ENERGY (KW)	EXERGY (MW)
1	139.45	536	178.33	3424.06	6.5211	610612.6	259.2943
2	38.34	364	174.1	3131.45	6.6629	545185.4	194.7165
3	35.2	535.5	174.1	3530.823	7.2444	614716.3	233.5566
4	36.34	226.8	4.2	2928.45	8.65	12299.49	1.314717
5	34.37	194.31	10.1	2885.75	8.69	29146.08	2.60784
6	7.09	332	163.94	3126.266	7.407	512520	145.5229
7	0.828	53.9	149.36	255.699	0.753	38191.2	5.115952
8	0.828	35.8	149.36	210.684	0.693	31467.76	1.109221
9	0.786	77.9	30.92	326.164	1.0503	10084.99	0.451154
10	3.971	85.01	10.73	356.25	1.134	3822.563	0.207125
11	6.7355	121.4	0.75	510.063	1.5424	382.5473	0.036982
12	18.04	56.7	149.166	314.979	1.01328	46984.16	2.182091
13	0.016	81.9	157.5	2654.271	9.2028	418047.7	20.27535
14	0.016	118.4	163.611	2723.34	9.388	445568.4	18.94726
15	179.265	165.7	164.04	710.3283	1.9788	116522.3	19.2387
16	179.265	172.2	51.08	738.649	2.4021	37730.19	0.882551
17	179.265	191.1	165.44	820.36	2.293	135720.4	21.84842
18	179.265	199	15.424	855.03	2.2966	13187.98	2.554849
19	179.265	223.3	170.3	963.37	2.5203	164061.9	35.11015
20	1.01325	146.3	164.77	7500	2.93	1235775	1090.546
21	1.01325	128	163.45	7200	2.76	1176840	1041.198
22	1.01325	30.9	44.8	129.59	0.4491	5805.632	0.012131
23	0.89	41.8	44.5	175.13	0.5693	7793.285	0.41706
24	1.01325	298.15	19.34			446367.2	454.5657

3.4 RESULTS AND DISCUSSION

In the current study, energy and exergy analysis has been done for a 210 MW coal fired thermal power plant with reheating. The schematic layout of the unit is shown in Figure 3.1. Table 3.3 shows the relevant thermodynamic data at various state points in the power plant.

At the base load condition under consideration, the boiler generates 178.33 kg/s of superheated steam at 139.45 bars. This superheated steam is fed into the HP turbine. From the HP turbine steam is fed to the re-heater. From the re- heater the steam passes on to the IP turbine from where it is fed to the LP turbine. Finally from the LP turbine, the steam is passed on to the condenser and reused in a closed cycle. For regenerative heating purpose, steam is extracted at various points from the HP and LP turbine as shown in Figure 3.1. The reference state of atmosphere has been taken at 25 ^{0}C.

The actual analysis of a coal fired thermal power plant and its various sub components is extremely complex owing to the huge amount of data associated with them, variable operating conditions and also the non linear behavior of steam properties. To make the analysis simpler programming in EXCEL has been used. This program allows us to make the calculations of various thermodynamic properties at different state points much easier. For analysis, the entire power plant has been sub divided into the following major sub systems: Boiler, Turbine, Condenser and Feed Water heaters. Control volume technique has been employed in the analysis wherein each component along with its different inlet and outlet streams has been considered as a single entity.

From the analysis, it has been observed that the boiler and steam turbine unit are the components where maximum exergy destruction takes place. These two components have been analyzed further to improve their performance on the basis of critical parameters.

3.4.1 ANALYSIS OF BOILER

There are many sources of irreversibilities in the boiler. The major among these is the chemical reaction owing to incomplete combustion of fuel and also the fact that high potential fuel is burnt in spontaneous combustion. The exergy destruction in the boiler increases at part load conditions due to improper heating of the inlet combustion air. The other major sources of exergy destruction are the excess air and inlet temperature of air. The effectiveness of the boiler can be improved significantly by pre heating the inlet air effectively and also by controlling the air to fuel ratio.

For the boiler, inlet hot air temperature, of the air coming from the air pre- heater, has been identified as the important parameter which affects its performance significantly. The hot air temperature has a direct impact on the exergetic efficiency and overall performance of the boiler. Analysis has been done to study the effect of variation in the hot air temperature on the exergetic efficiency of the boiler. Results of analysis are shown in Figure 3. 2.

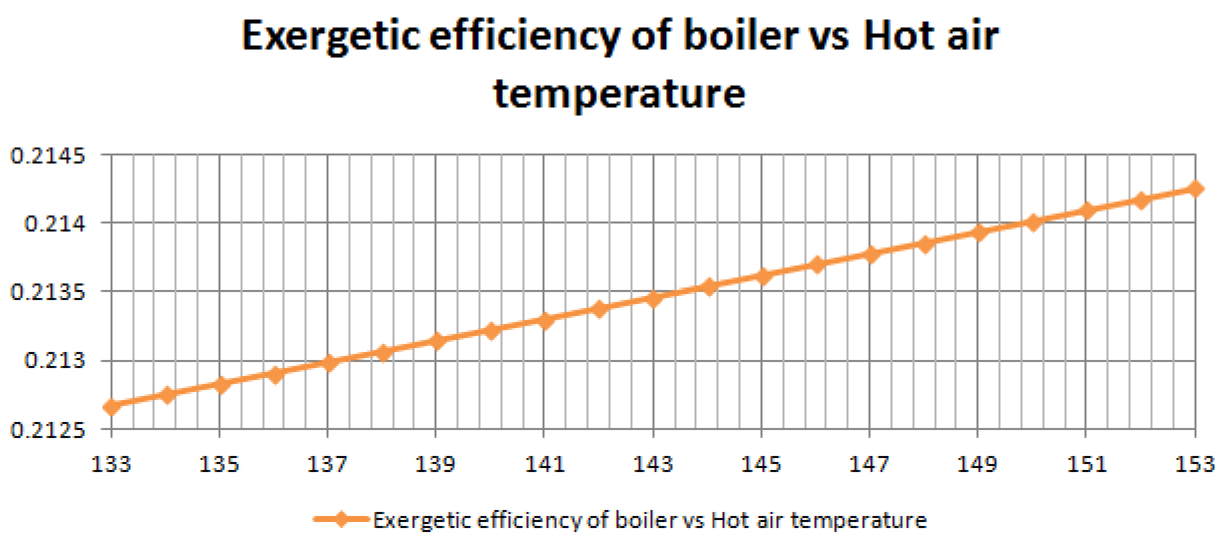


Figure 3.2 Analysis of exergetic efficiency of boiler as a function of hot air temperature

Analysis shows that the exergetic efficiency of the boiler increases with an increase in the hot air temperature. For the range of inlet hot air temperatures taken in the analysis, a steep rise in the exergetic efficiency is found to takes place. The main reason for this is that with an increase in the hot air temperature, for producing the same heating effect, less exergy is required from the fuel. Therefore, the inlet exergy reduces thereby increasing the overall exergetic efficiency of the boiler. Hence it is prudent to provide the hot air, coming from the air pre heater, at as high temperature as possible purely from exergetic efficiency point of view. However, the economics of the process which are discussed in later chapters point towards the fact that it becomes uneconomical to increase the temperature of hot air beyond a point.

Using the correlation techniques, a relation has been developed for exergetic efficiency of the boiler as a function of inlet hot air temperature as given below:

$$\varepsilon_B = 0.2841(1.0005\wedge T_{HA}) \tag{3.7}$$

where ε_B represents the exergetic efficiency of the boiler and T_{HA} represents the inlet hot air temperature.

3.4.2 ANALYSIS OF STEAM TURBINE

All the turbine units have been combined with a single inlet and multiple outlets. For the steam turbine, inlet steam temperature is the important parameter which affects the performance of the turbine significantly. The main reason for choosing inlet steam temperature as the critical parameter in the current study is that the temperature of steam entering the steam turbine has a direct impact on the performance of the steam turbine. Steam entering the turbine at higher temperature carries more ability to do work on the turbine blades. The major source of exergy destruction in the steam turbine is the large inertia of the turbine. When high energy steam makes impact with the turbine blade, lot of energy is wasted in overcoming the inertia of the turbine rotor. Further, friction between the turbine blades and steam is another factor, which leads to huge losses in exergy of the high energy steam. In the present study, analysis has been done to understand the effect of inlet steam temperature on the exergetic efficiency of the steam turbine unit. The results of the analysis are shown in Figure 3.3.

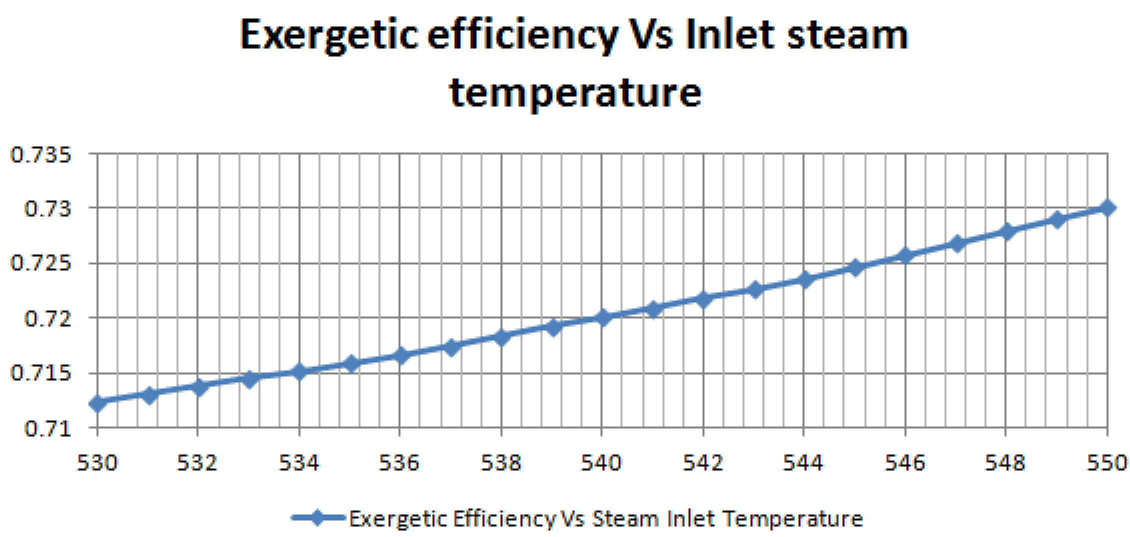


Figure 3.3 Analysis of exergetic efficiency of steam turbine unit as a function of inlet steam temperature

Results of the analysis show that exergetic efficiency increases with an increase in the inlet steam temperature. For the given range of inlet steam temperature, a steep rise in the exergetic efficiency of the steam turbine unit is found to take place. This happens because at higher steam inlet temperature, energy available to run the turbine is high, which in turn reduces the steam consumption in the turbine.

Using the correlation techniques, a relation has been developed for exergetic efficiency of the steam turbine unit as a function of inlet steam temperature as given below:

$$\varepsilon_T = 0.3724(1.0012^{T_s}) \tag{3.8}$$

Where ε_T represents the exergetic efficiency of the turbine and T_s represents the inlet steam temperature.

3.5 CONCLUSIONS

In the present work, energy and exergy analysis has been performed for a 210 MW coal fired thermal power plant to ascertain the losses taking place in the various components of the system. In the power plant under consideration, boiler and steam turbine have been found to be the components where maximum exergy destruction takes place.

These two components have been analyzed further to study the effect of important thermodynamic parameters on their respective exergetic efficiencies. Using correlation technique, relations have been developed between the exergetic efficiency of the boiler and steam turbine as functions of thermodynamic parameters.

Analysis shows that exergy based analysis provides a much better insight into the performance of a power plant as compared to the conventional approach based on the First Law of Thermodynamics and is a much more robust tool to analyze a thermal system.

CHAPTER- 4

EXERGOECONOMIC ANALYSIS AND OPTIMIZATION OF COAL FIRED THERMAL POWER PLANT

4.1 INTRODUCTION

In Chapter- 3 of the thesis, a detailed thermodynamic analysis of a 210 MW coal fired thermal power plant has been presented. Energy and exergy based analysis has been done for the power plant which allowed us to identify the boiler and steam turbine as the critical components where maximum exergy destruction takes place.

In this chapter, boiler and steam turbine have been analyzed by combining the principles of exergy with economics. As discussed earlier, it is much easier to assign monetary values to exergy. It allows us to estimate the unit product costs of various components and also allows us to quantify the monetary losses due to irreversibilities associated with various components. This analysis provides us with a strong tool to achieve the optimum design for complex thermodynamic systems.

Exergo-economics which integrates the exergy and economics principles involves the determination of:

a) Appropriate allocation of economic resources.
b) Understanding of economic feasibility and profitability of a thermal system.

All the techniques based on exergo-economics have certain common characteristics:

a) Combination of exergy and economic principles.
b) Exergy is the commodity of value in a thermal system and assigning of costs to various exergy related variables.

Any technique based on exergo-economics involves the calculation of exergoeconomic variables namely, average unit fuel cost, average unit product cost, cost of exergy destruction, relative cost difference and exergoeconomic factor. All the above mentioned factors are used to analyze the performance of a thermal system in monetary terms.

In the current study, exergoeconomic analysis and optimization has been done for a 210 MW coal fired thermal power plant. The critical components determined from the thermodynamic analysis i.e. boiler and steam turbine have been taken up for economic analysis.

In the present analysis, mass and energy conservation laws have been applied to these components. Quantitative balances of exergy and exergetic costs for the boiler and steam turbine have been considered.

4.2 FORMULATION OF EXERGO- ECONOMIC EQUATIONS

Exergy based analysis is a powerful tool to analyze the performance of any thermal system. It allows the analysis of both quantity and quality of energy utilization in the system. For a system operating at a steady state there may be a number of entering and exiting material streams as well as both heat and work interactions with the surroundings. Associated with these transfers of matters and energy are the exergy transfers into and out of the systems and the exergy destructions caused by the irreversibilities within the system. In exergy costing a cost is associated with each exergy stream.

The generalized exergy balance equation applicable to any thermal system has been formulated as per Bejan et. al. (1996) in the previous chapter and is given by Equation (4.1).

$$\sum_e \dot{E}_{e,k} + \dot{W}_k = \dot{E}_{q,k} + \sum_i \dot{E}_{i,k} \tag{4.1}$$

Unit exergetic cost can be assigned to every exergy stream and the cost balance equation corresponding to equation (4.1) is given by Equation (4.2).

$$\sum_e (c_e \dot{E}_e)_k + c_{w,k} \dot{W}_k = c_{q,k} \dot{E}_{q,k} + \sum_i (c_i \dot{E}_i)_k + \dot{Z}_k \tag{4.2}$$

where c denotes the unit exergetic cost associated with an exergy stream and the term $\dot{Z}_k$ represents all the charges associated with the operation and maintenance of the k_{th} component.

4.3 ECONOMIC ANALYSIS OF CRITICAL COMPONENTS OF THE POWER PLANT

The purpose of the analysis done in the economic study is to provide inputs which can further be used in the exergoeconomic analysis and optimization. During economic analysis following steps have been undertaken:

a) Estimation of purchased equipment cost. This has been done using the literature available.
b) Levelized costs have calculated keeping in mind that the costs of components change during the economic life cycle of the plant.

The amortization cost or present worth of a component is given by Equation (4.3).

$$PW = C_i - S_n PWF(i,n) \tag{4.3}$$

where C_i is the initial capital investment for the component. S_n and PWF denote the salvage value and present worth factor for the component, i and n denote the interest rate and component life in years.

The present worth of a component is converted into the annualized cost using the capital recovery factor *CRF(i,n)* as given by Equation (4.4).

$$\dot{C}(Rs./\,year) = PW * CRF(i,n) \tag{4.4}$$

We obtain the capital cost for the k_{th} component by dividing the levelized cost by 8760 annual operating hours as given by Equation (4.5).

$$\dot{Z}_k = \phi_k \, \dot{C}_k / (3600 * 8760) \tag{4.5}$$

The factor ϕ_k is used to take into account the maintenance cost by assuming the value as 1.06. However, in the present study the value has been taken as unity for each component.

4.4 RULES FOR FORMULATION OF AUXILLIARY EQUATIONS

The following rules have been used while developing the auxiliary equations for individual components of the power plant:

a) Costs of exergy streams associated with the entry of fuel and exhaust to the atmosphere have been taken as zero.

b) Costs of the exergy streams corresponding to the bleeding of steam from the high pressure turbines to the high pressure and low pressure heaters has been taken same as the cost of exergy stream related to the steam entering the high pressure turbine from the boiler.

4.5 COST BALANCE EQUATIONS FOR BOILER AND STEAM TURBINE

4.5.1 BOILER

The main purpose of the boiler is to raise the temperature and pressure of the feed water coming from the feed water heaters. The cost balance equation for the boiler is given by Bejan et. al. (1996s:

$$c_{24}\dot{E}_{CHE} + c_w \dot{E}_{baux} + c_{19}\dot{E}_{19} + \dot{Z}_b = c_1\dot{E}_1 + c_{21}\dot{E}_{21} \tag{4.6}$$

where c_{24}, c_{21}, c_w, and c_1 are the unit costs associated with streams 24, 21, auxiliary exergy consumption in the boiler and stream 1 respectively. $\dot{E}$, $\dot{E}_{CHE}$, $\dot{E}_{baux}$, and $\dot{Z}_b$ represent the exergy rates at various points, chemical exergy of fuel, auxiliary exergy consumption in the boiler and the capital investment and operation and maintenance cost of the boiler respectively. The auxiliary relations for the boiler are derived as per 4.4 (a) and is given by Equation (4.7).

$$c_{21} = c_{24} = 0 \tag{4.7}$$

4.5.2 STEAM TURBINE

The turbine system used in a coal fired thermal power plant comprises of a high pressure (HP) turbine, intermediate pressure (IP) turbine and a low pressure (LP) turbine. For analysis purpose, the entire turbine assembly has been taken as a single unit in which the steam is expanding from the turbine inlet pressure to condenser back pressure.

The exergetic cost balance equation for the turbine system is given by Equation (4.8).

$$c_1 \dot{E}_1 + c_w \dot{E}_{taux} + \dot{Z}_t = c_4 \dot{E}_4 + c_5 \dot{E}_5 + c_9 \dot{E}_9 + c_{10} \dot{E}_{10} + c_{11} \dot{E}_{11} + c_7 \dot{E}_7 + c_w \dot{E}_{turbine} \tag{4.8}$$

Where c denotes the unit exergetic cost for various streams. $\dot{E}_{taux}$ and $\dot{E}_{turbine}$ denote the auxiliary exergy to the turbine and exergy at the turbine outlet respectively. $\dot{Z}_t$ denotes the capital investment and operation and maintenance cost of the turbine.

Auxiliary relations used in the study are derived as per 4.4 (b) and is given by Equation (4.9).

$$c_1 = c_4 = c_5 = c_7 = c_9 = c_{10} = c_{11} \tag{4.9}$$

4.6 EXERGOECONOMIC ANALYSIS AND OPTIMIZATION OF BOILER AND STEAM TURBINE

4.6.1 BOILER

The boiler system comprises of the following sub systems:

1. Feed water system
2. Steam system
3. Fuel system
4. Air system

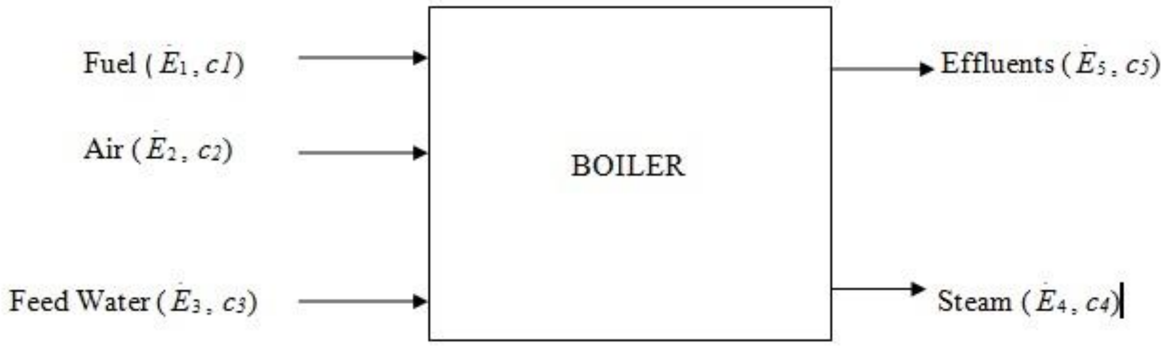


Fig. 4.1 Different Streams Entering and Leaving the Boiler System

Unit product cost of the boiler is calculated as per Equation (4.10).

$$c_4 = [c_1 \dot{E}_1 + c_2 \dot{E}_2 + c_3 \dot{E}_3 - c_5 \dot{E}_5] / \dot{E}_4 \tag{4.10}$$

The average costs per exergy unit of fuel and product for the boiler are given by Equations (4.11) and (4.12).

$$c_{F,B} = \frac{\dot{C}_{F,B}}{\dot{E}_{F,B}} \tag{4.11}$$

$$c_{P,B} = \frac{\dot{C}_{P,B}}{\dot{E}_{P,B}} \tag{4.12}$$

In the exergoeconomic analysis and optimization process, all the variables are taken as constant and the effect of hot air temperature, coming from the air pre- heater, on the unit product cost of boiler are analyzed. Optimum value of the inlet air temperature is obtained as a best balance between the unit product costs of the boiler and air pre-heater.

The optimization process for the boiler involves the following assumptions:

1. The exergy flow rate from the boiler is constant.
2. The unit cost rate for fuel is taken as constant.

4.6.2 STEAM TURBINE SYSTEM

Various streams entering and leaving the steam turbine system are shown in Figure 4.2.

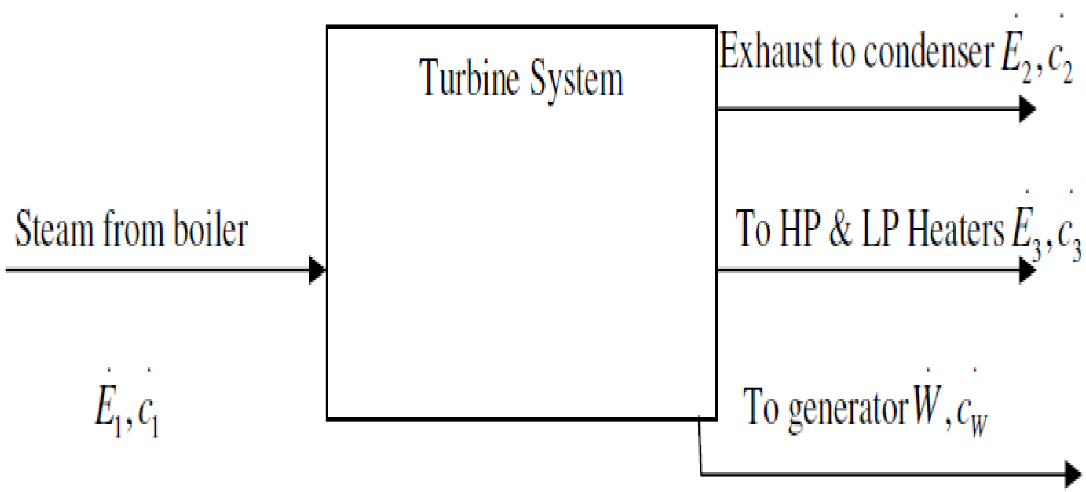


Figure 4.2 Different Streams Entering and Leaving the Steam Turbine System

For a component receiving a heat transfer and generating power we can write:

$$\sum C_{e,k} + C_{w,k} = C_{q,k} + \sum C_{i,k} + Z_k \tag{4.13}$$

Unit product cost of the turbine system is evaluated as per Equation (4.14).

$$c_w = [c_1 \dot{E}_1 - c_2 \dot{E}_2 - c_3 \dot{E}_3] / \dot{W} \tag{4.14}$$

Average cost per unit exergy of fuel and product, for the turbine, have been calculated as per Bejan et. al. and are given by Equations (4.15) and (4.16) [ ].

$$c_{F,T} = \frac{\dot{C}_{F,T}}{\dot{E}_{F,T}} \tag{4.15}$$

$$c_{P,T} = \frac{\dot{C}_{P,T}}{\dot{E}_{P,T}} \tag{4.16}$$

In the exergoeconomic analysis and optimization process all the variables are taken as constant and the effect of inlet steam temperature, coming from the boiler, on the unit product cost of steam turbine system are studied and analyzed. Optimum value of the inlet steam temperature is obtained as a best balance between the unit product costs of the steam turbine system and boiler.

The optimization process for the steam turbine system involves the assumption that unit exergetic costs of streams entering the feed water heaters and condenser is taken to be same as the input cost of fuel.

4.7 RESULTS AND DISCUSSION

4.7.1 BOILER

Table 3.3 shows the relevant thermodynamic data of a 210 MW coal fired thermal power plant. In the base case design, water from the feed water heaters enters the boiler system at 223.3 ^{0}C and hot air from the air pre heater enters at 146.3 ^{0}C. The unit product cost of the boiler and air pre heater at this configuration are found to be 0.19425 Rs/ MJ and 0.717 Rs/ MJ respectively.

Programming in Excel has been used to perform the analysis of unit product costs of boiler and air pre heater with variation in hot air temperature. Multi- objective optimization has been done considering unit product cost of boiler as the parameter to be optimized. Optimization has been achieved at single load condition. Time based analysis is not part of the current study. Exergoeconomic optimization for the boiler has been done considering hot air temperature as the variable.

The methodology adopted for exergo- economic analysis and optimization of the boiler system is depicted by the flow chart given in Figure 4.3. The flow chart depicts the step by step procedure followed during the analysis.

The first step in the exergo- economic analysis process is to analyze effect of hot air temperature on unit product cost of the air pre- heater. Unit product cost of air pre- heater is found to increase with an increase in the hot air temperature. This happens, since to provide air at a higher temperature, more money has to be spent in the operation of air pre- heater. Results of the analysis are shown in Figure 4.4.

Next step in the exergoeconomic process is to analyze the variation of unit product cost of boiler with hot air temperature. It has been found that increase in the hot air temperature results in lowering of the unit product cost of boiler because to produce the same heating effect less fuel is needed, resulting in lower fuel costs. Hence the overall unit product cost of the boiler decreases. Results of the analysis are given in Figure 4.5.

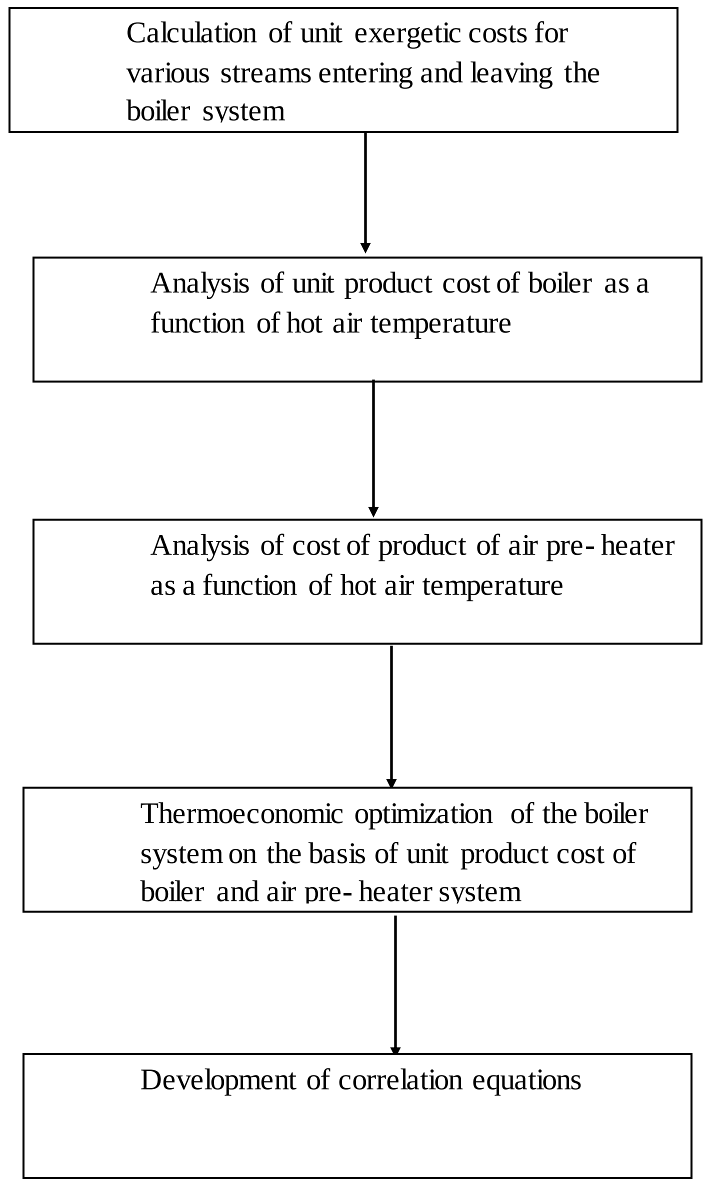


Figure 4.3 Flowchart depicting the methodology adopted in the Exergoeconomic Analysis and Optimization of Boiler

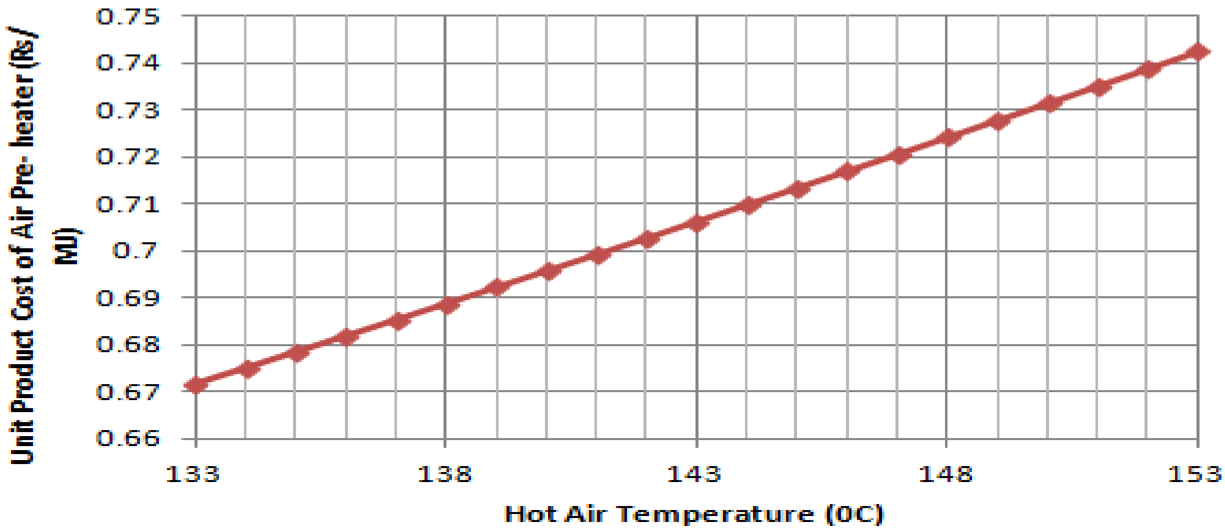


Figure 4.4 Variation in Unit Product Cost of Air Pre-heater with variation in Hot Air Temperature

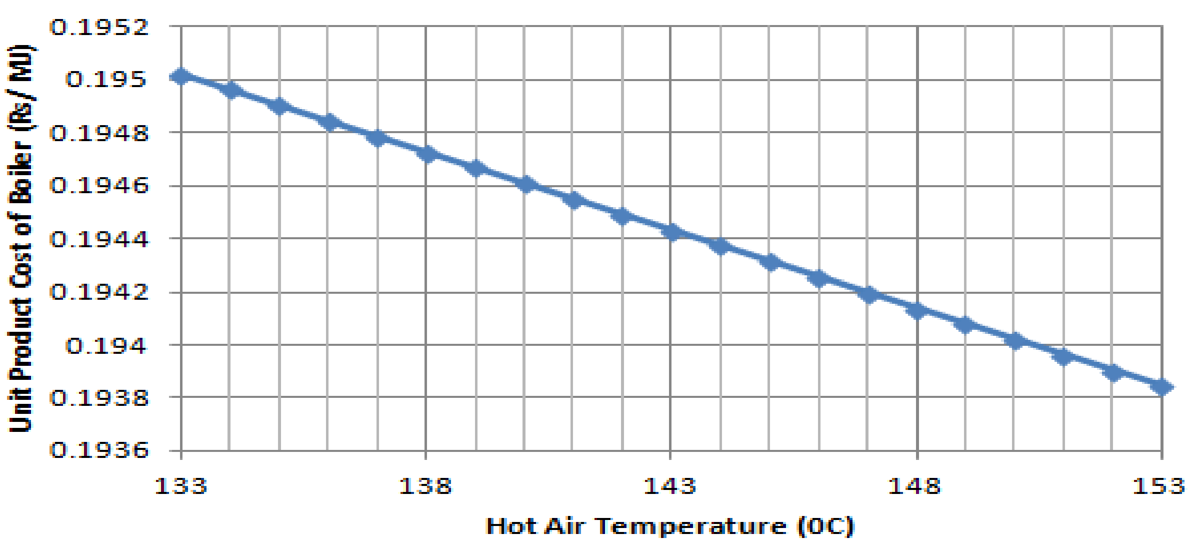


Figure 4.5 Variation in Unit Product Cost of Boiler with Hot Air Temperature

Combining the results of the previous two steps, it can be seen that the unit product cost of air pre- heater increases with an increase in the hot air temperature whereas the unit product cost of boiler decreases. The final step in the analysis is to find the optimal value of hot air temperature at which air should be supplied to the boiler. To arrive at the optimal solution, a best balance has been made between the unit product cost of the air pre- heater and the unit product cost of the boiler. Results of the optimization process are shown in Figure 4.6.

The optimal state appears at the point where the two curves, corresponding to the unit product cost of boiler and unit product cost of air pre- heater, intersect. At this point, the hot air temperature is found to be 143 ^{0}C. The unit product costs for the boiler and the air pre- heater are found to be 0.708 Rs/MJ and 0.1944 Rs/ MJ at this point. For achieving further reduction in unit product cost of the boiler more money would have to be spent to provide air at higher temperatures thereby increasing the overall unit product cost of the boiler system.

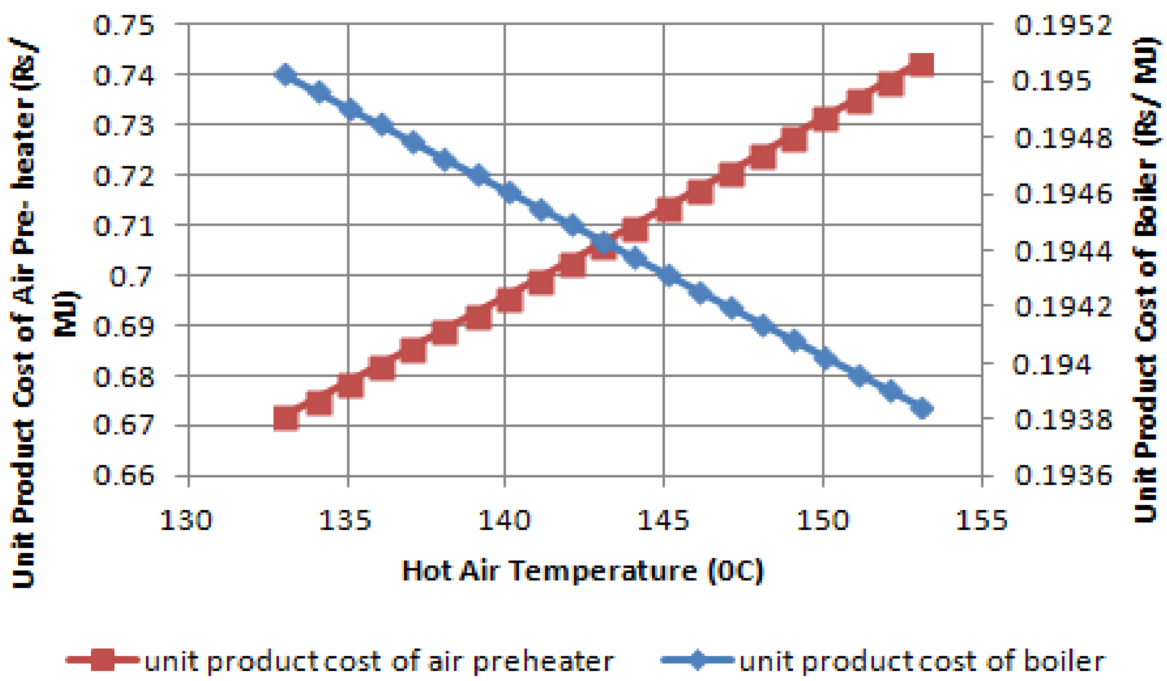


Figure 4.6 Optimization for Unit Product Cost of Boiler as a Function of Hot Air Temperature

From the analysis, the following relations have been developed to illustrate the relationship between the unit product cost of the boiler and air pre-heater as functions of hot air temperature:

$$c_{aph} = 0.0104(1.345\text{^}T_{HA}) \quad (4.17)$$

$$c_b = 0.3936(0.9996\text{^}T_{HA}) \quad (4.18)$$

4.7.2 STEAM TURBINE

The methodology adopted in the exergo- economic analysis and optimization of the steam turbine system is depicted by the flow chart given in Figure 4.7.

In the base case design, steam from the boiler enters the high pressure turbine at temperature and pressure of 536 ^{0}C and 139.45 bars. As discussed earlier, in the current analysis, the entire turbine system comprising of a High Pressure turbine, Intermediate Pressure turbine and Low Pressure turbine has been taken as a single unit. At this configuration, the unit product cost of the boiler and turbine are found to be 0.19425 Rs/ MJ and 1.1907 Rs/ MJ.

Programming in Excel has been used to perform the analysis of unit product cost of boiler and steam turbine with variation in inlet steam temperature. Multi- objective optimization has been done considering unit product cost of turbine as the parameter to be optimized. Optimization has been achieved at single load condition. Time based analysis is not part of the current study. Exergo- economic optimization for the steam turbine has been done considering inlet steam temperature as the variable.

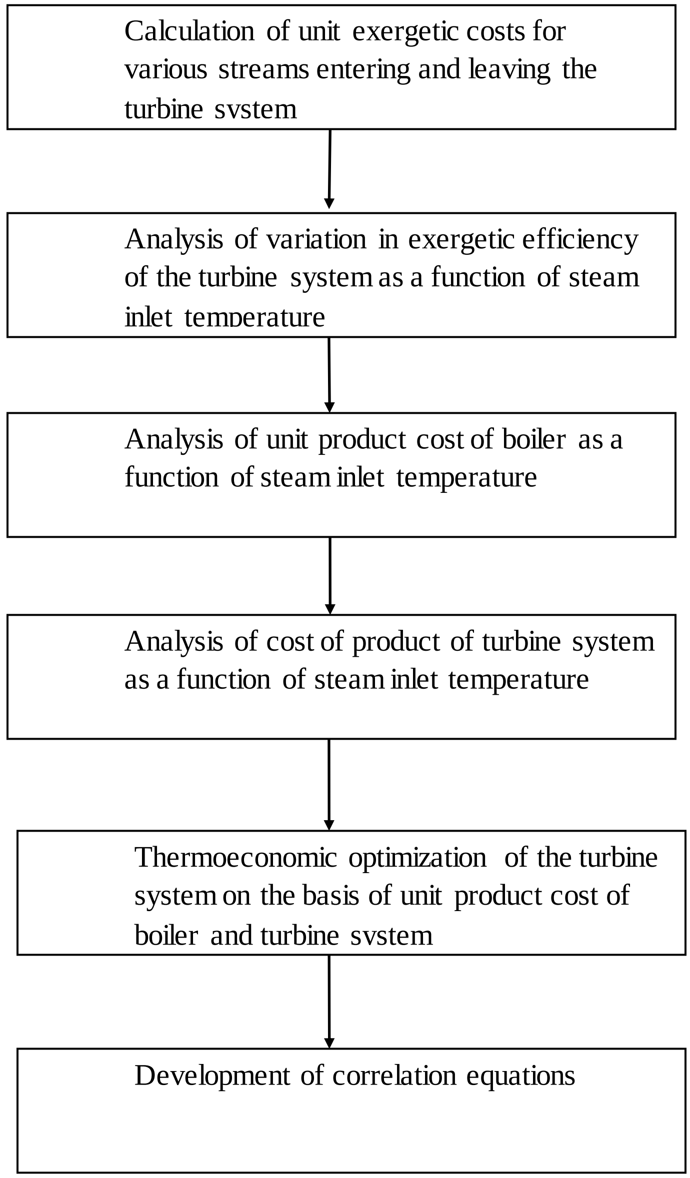


Figure 4.7 Flowchart depicting the methodology adopted in the Exergo- economic Analysis and Optimization of Steam Turbine

The first step in the exergo- economic analysis is to analyze the effect of inlet steam temperature on the unit product cost of the boiler. Unit product cost of the boiler is found to increases with an increase in the inlet steam temperature. This happens, since to provide steam at a higher temperature, more money has to be spent in the operation and maintenance of the boiler. Results of the analysis are shown in Figure 4.8.

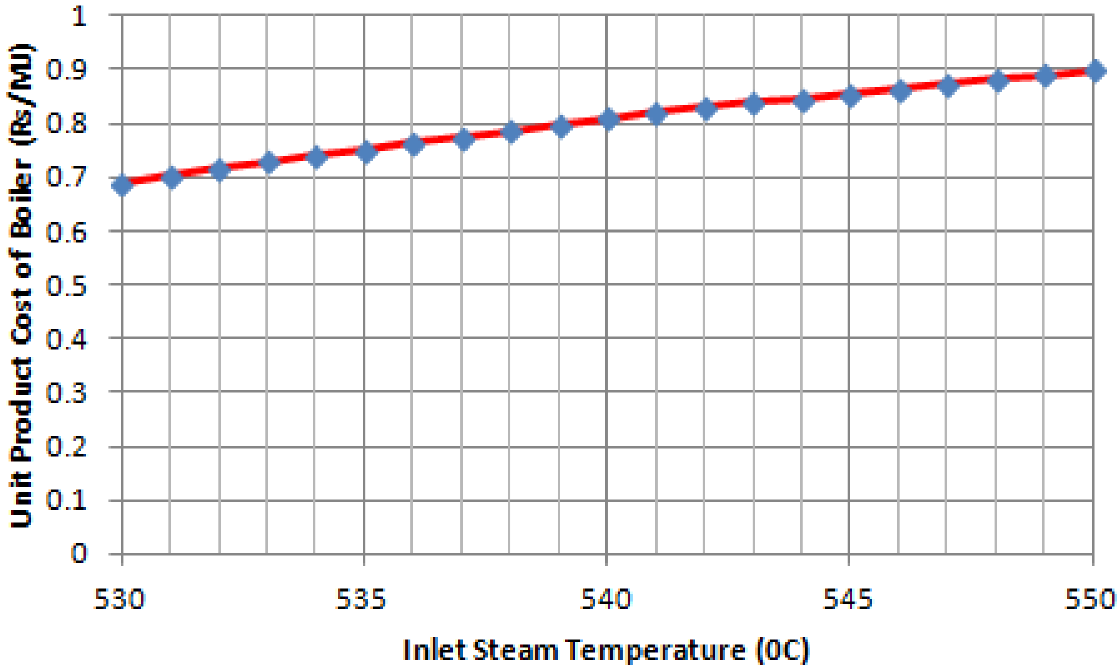


Figure 4.8 Variation in Unit Product Cost of Boiler with Inlet Steam Temperature

Next step in the exergoeconomic analysis is to analyze the variation of unit product cost of steam turbine with variation in inlet steam temperature. It has been found that increase in the inlet steam temperature results in lowering of the unit product cost of steam turbine. This happens because, with an increase in the inlet steam temperature, higher inlet pressure energy is available to run the turbine which in turn reduces the consumption of steam in the turbine. Hence, the overall unit product cost of the steam turbine decreases. Results of the analysis are given in Figure 4.9.

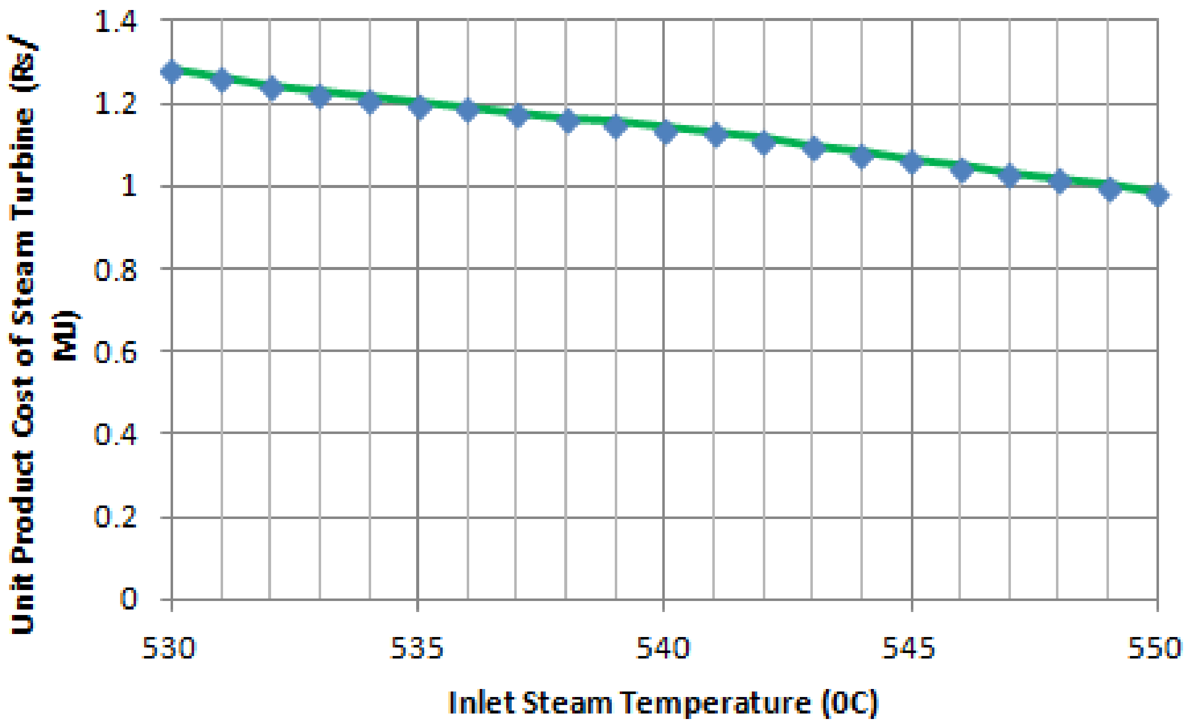


Figure 4.9 Variation in Unit Product cost of Steam turbine with Inlet Steam Temperature

Combining the results of the previous two steps, it can be seen that the unit product cost of boiler increases with an increase in the inlet steam temperature whereas the unit product cost of boiler decreases. The final step is to find the optimal value of inlet steam temperature at which steam should be supplied to the steam turbine. To arrive at the optimal solution, a best balance has been made between the unit product cost of the boiler and the unit product cost of the steam turbine. Results of the optimization process are shown in Figure 4.10.

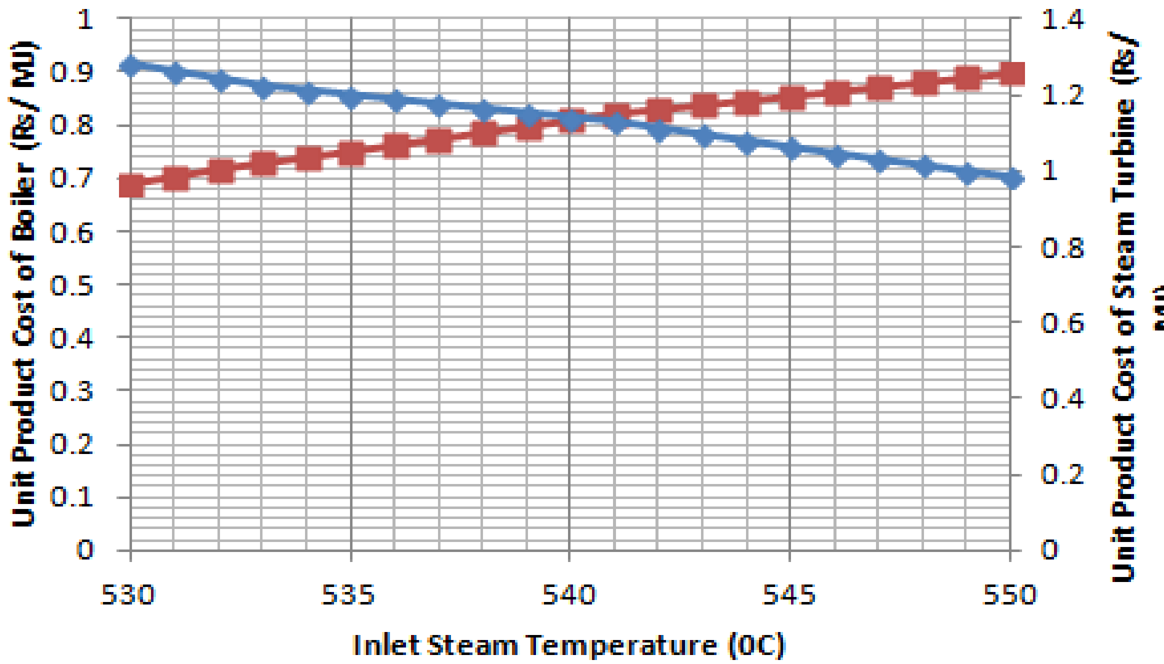


Figure 4.10 Optimization for Unit product Cost of Steam Turbine as a Function of Inlet Steam Temperature

The optimal state appears at the point where the two curves, corresponding to the unit product cost of boiler and unit product cost of steam turbine, intersect. At this point, the inlet steam temperature is found to be 540 ^{0}C and the unit product cost for the boiler and the steam turbine are found to be 0.80971Rs/MJ and 1.1437 Rs/ MJ. For achieving further reduction in unit product cost of the steam turbine, more money would have to be spent to provide steam at higher temperatures, thereby increasing the overall unit product cost of power generation.

From the analysis, the following relations have been developed to illustrate the relationship between the unit product cost of the boiler and steam turbine as functions of inlet steam temperature.

$$c_T = 822.925(0.9879 \wedge T_{IS}) \tag{4.19}$$

$$c_B = 0.0004(1.0141 \wedge T_{IS}) \tag{4.20}$$

4.8 CONCLUSIONS

The exergoeconomic technique presented in the current study is a powerful tool to analyze the performance of a coal fired thermal power plant. It combines the thermodynamic and economic principles. This technique is able to identify all the cost sources in a thermal system. It provides a better analysis of the cost consumption in any component of the thermal system by assigning unit costs to all the streams and leaving that component.

From the current study, many conclusions can be drawn. Boiler and Steam Turbine are the components with maximum exergy destruction. Hence maximum attention needs to be paid on these two components to achieve maximum cost savings. Critical parameters have been identified for both boiler and steam turbine which play a significant role in affecting their unit product cost.

Hot air temperature has been identified as the critical parameter which affects the unit product cost of the boiler significantly. Optimization has been done to arrive at the optimal value of hot air temperature of 143 ^{0}C at which air should be supplied to the boiler from the air pre- heater. At this temperature, the unit product cost of the boiler and turbine are found to be 0.19425 Rs/ MJ and 1.1907 Rs/ MJ.

Inlet steam temperature has been identified as the critical parameter which affects the unit product cost of the steam turbine significantly. Optimization has been done to arrive at the optimal value of inlet steam temperature of 540 ^{0}C at which steam should be supplied to the turbine from the boiler. At this temperature, the unit product cost of the boiler and the steam turbine are found to be 0.80971Rs/MJ and 1.1437 Rs/ MJ.

Finally, correlation equations have been developed for unit product cost of boiler and air pre-heater as functions of hot air temperature and also for unit product cost of boiler and steam turbine as functions of inlet steam temperature. These correlation equations quantify the unit product cost of various components in terms of thermodynamic variables under consideration.

CHAPTER- 5

ENERGY AND EXERGY ANALYSIS OF OPEN CYCLE GAS TURBINE POWER PLANT

5.1 INTRODUCTION

Open cycle gas turbine power plants play a significant role in the power generation industry. In India alone, 10.5% of the total thermal power capacity is generated by gas turbine power plants. Hence, analysis of these power plants is of significant interest from thermodynamic point of view. Over the years, many researchers have analyzed the performance of open cycle gas turbine power plants using the approach based on first law of thermodynamics. This approach uses energy as the criterion for defining the performance of the power plants. As pointed out earlier, use of this approach has its limitations, as it is unable to take into account the irreversibilities which are inherent part of the system. To take into account these irreversibilities, a better approach has been developed based on the second law of thermodynamics. In this approach, exergy is the criterion for defining the performance of a thermal system. This approach allows us to take into consideration the irreversibilities associated with the various components of the system.

In the current study, an exergy based approach has been illustrated for an open cycle gas turbine power plant. For formulation, a 25 MW open cycle gas turbine power plant has been considered as an example. Detailed exergy analysis has been done for the various plant components. Exergy destruction has been calculated for various components and the effect of thermodynamic variables on the exergy destruction in various components has been analyzed.

Finally, equations have been developed which provide a correlation between the exergy destruction in different components as functions of the thermodynamic variables under consideration. The current study provides a robust method which can be used to analyze open cycle gas turbines of different capacities.

5.2 SYSTEM DESCRIPTION

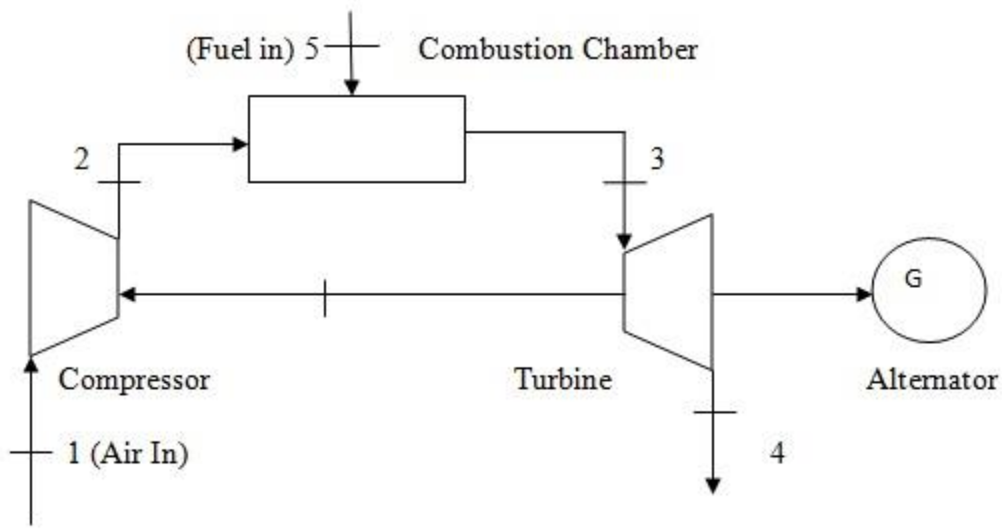


Figure 5.1 Schematic Layout of a 25 MW Open Cycle Gas Turbine Power Plant

The schematic layout of a 25 MW open cycle gas turbine power plant is given in Figure 5.1. It shows the major exergy and work flows in the system and also the state points which have been considered in the present analysis. The mass flow rate of air is 212.95 kg/ s and air enters the compressor at a temperature of 200 ^{0}C and a pressure of 0.981 bars. The pressure increases to 4.81 bars through the compressor whose isentropic efficiency has been taken as 80%. The inlet temperature to the gas turbine is 1123 ^{0}C and a pressure is 1.01325 bars. The isentropic efficiency of the turbine has been taken as 80%. The exhaust gases from the turbine are at 817 ^{0}C and 1.10 bars. The fuel (natural gas) is injected at 200 ^{0}C and 22 bars.

5.3 ENERGY AND EXERGY BALANCE FOR AN OPEN CYCLE GAS TURBINE POWER PLANT

Steady flow conditions can be closely approximated by devices which are considered for continuous operation such as compressor, combustion chamber and gas turbine for the power plant. The conservation of mass principle for a general steady flow system with multiple inlets and outlets is given in Equation 5.1.

$$\sum \dot{m}_i = \sum \dot{m}_e \tag{5.1}$$

where *m* denotes the mass flow rate and subscript *i* and *e* refer to inlet and exit respectively.

The general exergy balance equation applicable to the *kth* component of any thermal system is given in Equation 5.2.

$$\sum_e E_{e,k} + W_k = E_{q,k} + \sum_i E_{i,k} \tag{5.2}$$

The thermo- mechanical exergy of any stream can be decomposed into its thermal and mechanical components and is given by Equation 5.3 and Equation 5.4.

$$\dot{E}^T = \dot{m} c_p [T - T_0 - T_0 (\ln \frac{T}{T_0})] \tag{5.3}$$

where $\dot{E}^T$ is the thermal exergy of the component.

$$\dot{E}^{M} = mRT_0 \ln \frac{p}{p_0} \tag{5.4}$$

where $\dot{E}^{M}$ is the mechanical exergy of the component.

The physical exergy flow rate at any point is given by Equation 5.5.

$$\dot{E}^{PH} = \dot{E}^{T} + \dot{E}^{M} \tag{5.5}$$

The detailed exergy analysis includes calculation of exergy destruction in each component. The exergy balance equation for the kth component is given by Equation 5.6.

$$\sum_e \dot{E}_{e,k} + \dot{W}_k = \dot{E}_{q,k} + \sum_i \dot{E}_{i,k} \tag{5.6}$$

The exergy destruction for the kth component is calculated by Equation 5.7.

$$\dot{E}_{D,k} = \sum_i \dot{E}_{i,k} - \sum_e \dot{E}_{e,k} \tag{5.7}$$

5.4 RESULTS AND DISCUSSION

For the 25 MW open cycle gas turbine power plant under consideration, the values for different thermal properties and thermal, mechanical, chemical exergies and exergy flow rates have been calculated and are given in Table 5.1.

Table 5.1 Property values and thermal, mechanical, chemical and net exergy flow rates at various state points in the gas turbine power plant

State	$\dot{m}$ (kg/s)	P (bar)	T (K)	$\dot{E}^T$ (MW)	$\dot{E}^M$ (MW)	$\dot{E}^C$ (MW)	$\dot{E}$ (MW)
1	212.95	0.981	293.00	0.00	0.00	0.00	0.00
2	212.95	4.2	481.60	10.52	26.146	0.00	36.916
3	216.66	1.01325	1123.00	108.768	23.89	0.7488	133.406
4	216.66	1.1	817.60	55.80	1.5	0.7488	58.04
5	3.71	22	293.00	0.00	0.8240	190.53	191.39

The net flow rates of different exergies crossing the boundary at each component, at the base load condition, along with the exergy destruction in each component are shown in Table 5.2. Positive values indicate the exergy flow rates of product and negative values represent the exergy flow rates of fuel. In the current study, product of a component indicates the added exergy while the fuel indicates the exergy consumed by the component.

Table 5.2 Exergy balance for each component in the gas turbine power plant

Component	$\dot{E}^W$ (MW)	$\dot{E}^C$ (MW)	$\dot{E}^T$ (MW)	$\dot{E}^M$ (MW)	$\dot{E}^D$
Compressor	-40.363	0.00	10.52	25.146	4.697
Combustion chamber	0.00	-189.78	98.248	-3.083	94.615
Gas Turbine	66.498	0.00	-52.698	-22.39	8.59
Overall Plant	26.135	-189.78	56.07	0.673	106.902

The sum of the exergies of various exergy flow rates of fuel, product and destruction for each component turns out to be zero. This zero sum indicates that the exergy balance equations have been satisfied for each component. The net exergy flow rate to the compressor is 40.363 MW, out of which 26% belongs to thermal exergy and 65% belongs to mechanical exergy. The net exergy flow rate to the combustion chamber is 190 MW, out of which 52% belongs to the thermal exergy and a very small portion of 1.6% belongs to the mechanical exergy. Remaining exergy equivalent to approximately 46% is destroyed in the combustion chamber. The net exergy flow rate to the turbine is 66 MW, out of which 79% belongs to the thermal exergy and 33% belongs to mechanical exergy. Remaining exergy equivalent of approximately 13% is destroyed in the turbine.

The values for exergy destruction calculated for each component and the overall plant are plotted as shown in Figure 5.2.

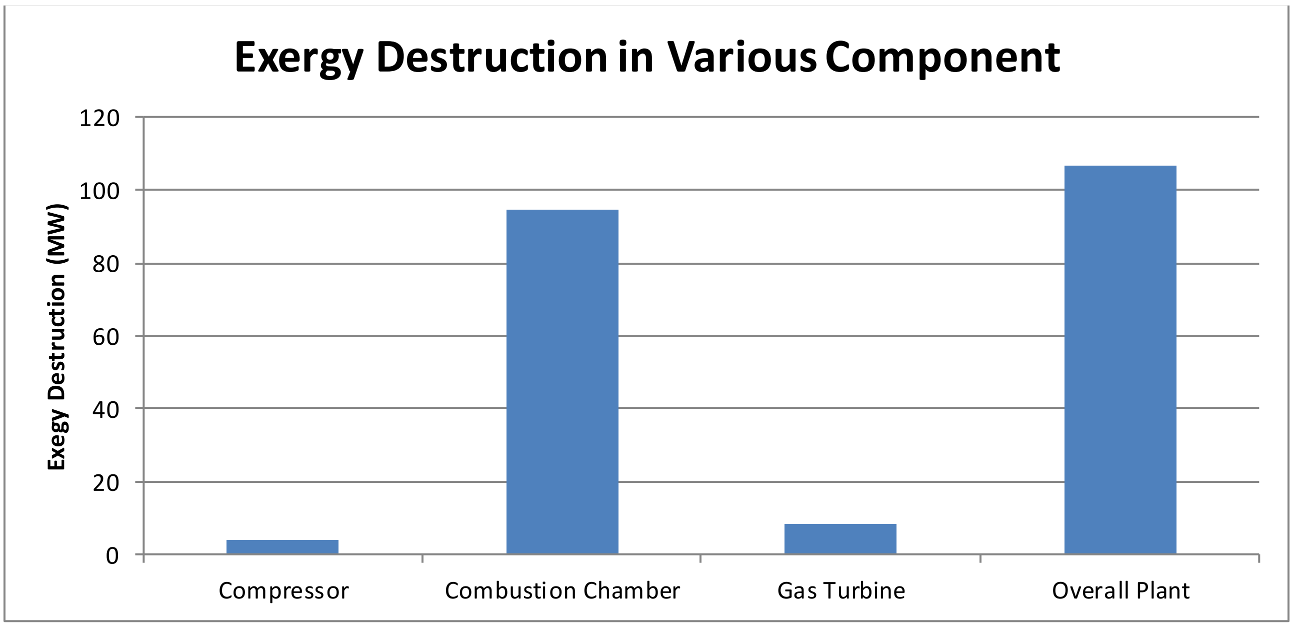


Figure 5.2 Exergy Destruction in Various Components of the Open Cycle Gas Turbine Power Plant

It can be seen that the maximum exergy destruction takes place in the combustion chamber followed by the gas turbine and the air compressor. The next step in the analysis is to study the effect of thermodynamic variables on the performance of the gas turbine power plant. For this the following two thermodynamic variables have been considered:

1. Compressor pressure ratio (r_p)

2. Inlet Air temperature

The effects of these two thermodynamic variables have been analyzed on the exergy destruction values for various components.

5.4.1 EFFECT OF COMPRESSOR PRESSURE RATIO (r_p)

The effects of variation of the compressor pressure ratio have been analyzed, in detail, on the exergy destruction taking place in the compressor and combustion chamber for the chosen range of compressor pressure ratios. The results of this analysis are illustrated in Figures 5.3 and 5.4.

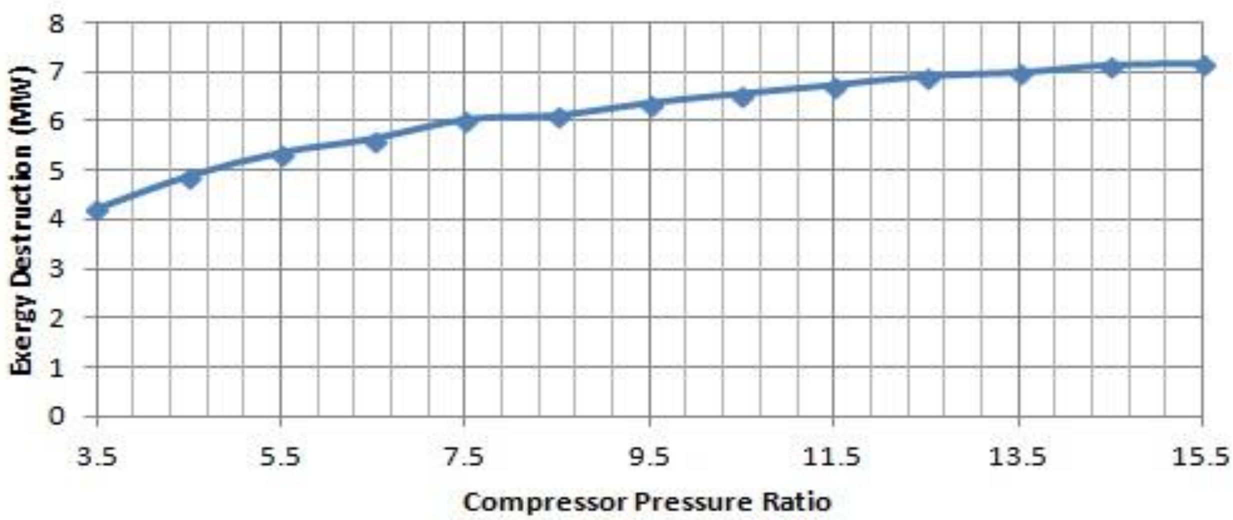


Fig. 5.3 Exergy Destruction in Compressor Vs. Compressor Pressure Ratio

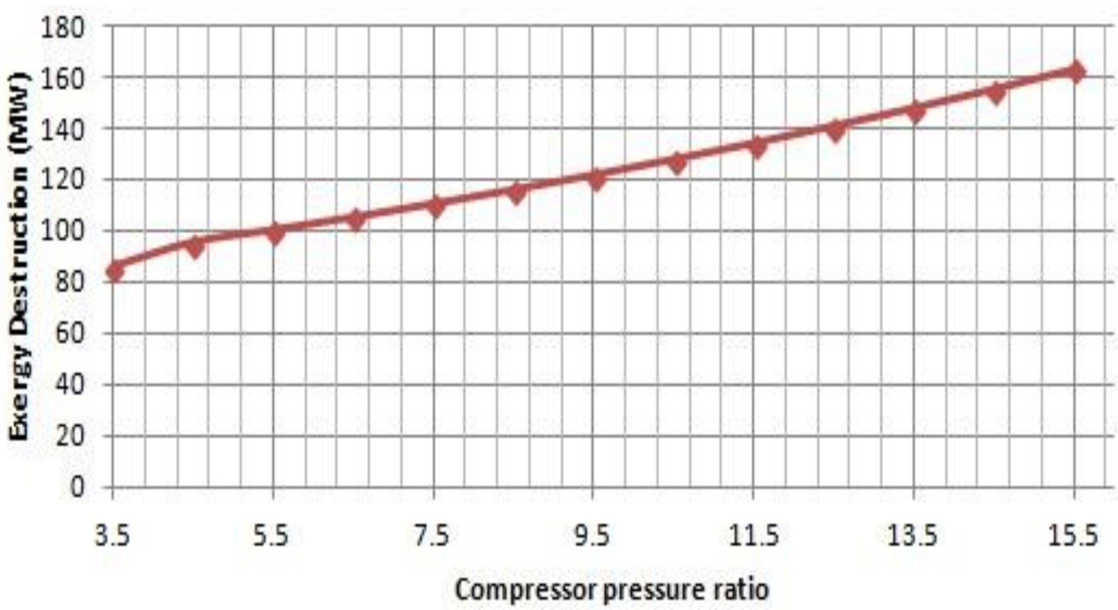


Fig. 5.4 Exergy Destruction in Combustion Chamber Vs. Compressor Pressure Ratio

From Figures 5.3 and 5.4 it is clear that with an increase in the compressor pressure ratio, the exergy destruction increases for both air compressor and combustion chamber. For the compressor, the exergy destruction increases from 4.12 MW at a compressor pressure ratio of 3.5 to 7.09 MW at a compressor pressure ratio of 15.5. The primary reason for the increase in exergy destruction, in the compressor, is that for the same exergetic efficiency the losses because of heat transfer tend to increase significantly. The overall increase in the exergy destruction for the chosen range of compressor pressure ratios is 70%. For the combustion chamber, the exergy destruction increases from 80.2 MW at a compressor pressure ratio of 3.5 to 160 MW at a compressor pressure ratio of 15.5. The main reason for the increase in exergy destruction in the combustion chamber is that with an increase in the compressor pressure ratio, the overall performance of the combustion system reduces and unless regeneration is used, the performance of the combustion chamber goes on deteriorating. The overall increase in the exergy destruction of the combustion chamber for the chosen range of compressor pressure ratios is 90%.

The effect of increase in compressor pressure ratio on the exergy destruction in various components is found to be different and its effect is felt more in the case of combustion chamber. For the given range of compressor pressure ratios, the increase in exergy destruction for various components, in terms of percentage is given in Table 5.3.

Table 5.3 Maximum exergy destruction variation in various components

Component	Maximum exergy destruction variation (%)
Compressor	70.002
Combustion chamber	90.03

Based on the analysis done, following equations have been developed which express the exergy destruction in compressor, combustion chamber and gas turbine as functions of compressor pressure ratio. These equations have been checked for different ranges of compressor pressure ratio and the results have been fairly satisfactory. These are represented as Equations 5.8 and 5.9.

$$\dot{E}^{D}_{Comp} = 1.9308 + 1.9505\ln(r_p) \tag{5.8}$$

$$\dot{E}^{D}_{CC} = 15.5271 + 49.8326\ln(r_p) \tag{5.9}$$

where $\dot{E}^{D}_{Comp}$ and $\dot{E}^{D}_{CC}$ represent the exergy destruction in compressor and combustion chamber.

5.4.2 EFFECT OF INLET AIR TEMPERATURE

The effects of variation of the ambient air temperature have been analyzed, in detail, on the exergy destruction taking place in the compressor, combustion chamber and gas turbine for the chosen range of turbine inlet temperatures. The results of this analysis are illustrated in Figures 5.5, 5.6 and 5.7.

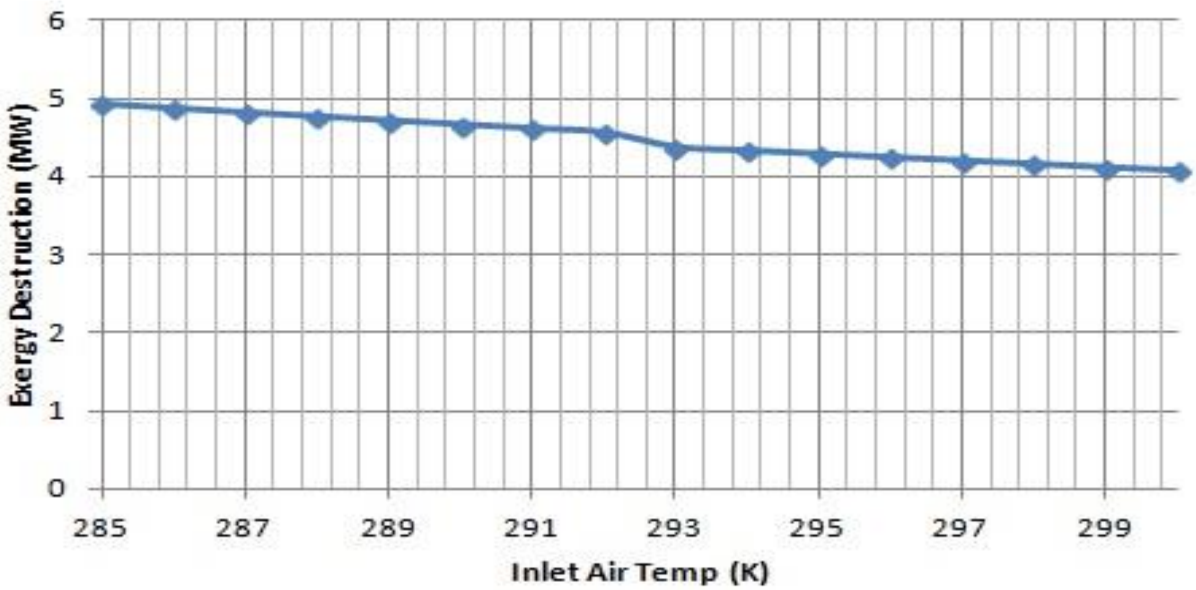


Figure 5.5 Exergy Destruction in Compressor Vs. Inlet Air Temperature

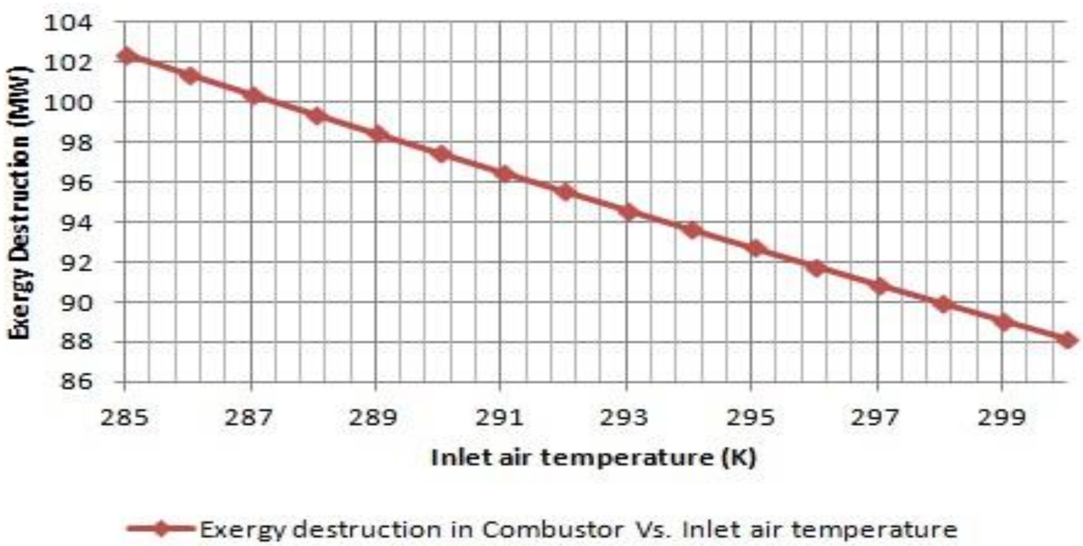


Figure 5.6 Exergy Destruction in Combustion Chamber Vs. Inlet Air Temperature

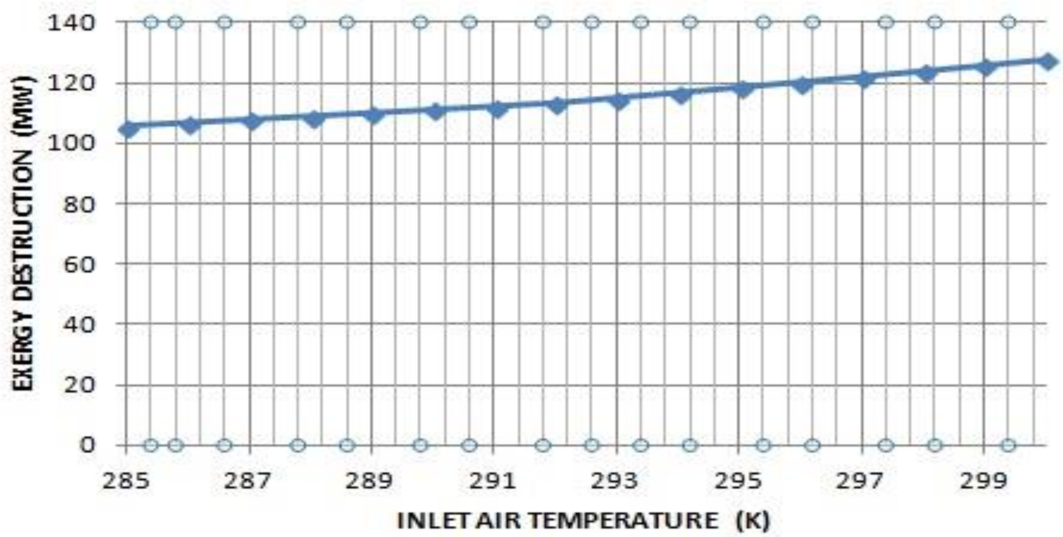


Figure 5.7 Exergy destruction in gas turbine Vs. Inlet Air Temperature

From Figures 5.5, 5.6 and 5.7, it is clear that the exergy destruction decreases for air compressor and combustion chamber and increases for gas turbine with increase in inlet air temperature. For the compressor, the exergy destruction decreases from 4.92 MW at 285 K to 4.18 MW at 299 K. This happens because air enters the air compressor at a higher temperature and for the same compression ratio it carries more heat. The overall decrease in the exergy destruction in the compressor is found to be 83% for the chosen range of inlet air temperatures. For the combustion chamber, the exergy destruction increases from 102 MW at 285 K to 86.9 MW at 299 K. This happens because air from the compressor enters the combustion chamber at a higher temperature and less quantity of fuel is required to achieve the same temperature for turbine inlet. The overall decrease in the exergy destruction for the combustion chamber is found to be 86% for the chosen range of inlet air temperatures. For the gas turbine, the exergy destruction increases from 103 MW at 285 K to 127 MW at 299 K. The primary reason for increase in the exergy destruction in the gas turbine is that with an increase in the inlet air temperature, exhaust temperature from the turbine increases, thereby increasing the exergy destruction in the gas turbine. The overall increase in the exergy destruction for the gas turbine is found to be 23% for the chosen range of inlet air temperatures.

The effect of increase in inlet air temperature on the exergy destruction in various components is found to be different and its effect is felt most in the case of combustion chamber. For the given range of inlet air temperatures, the increase in exergy destruction, for various components in terms of percentage is given in Table 5.4.

Table 5.4 Maximum exergy destruction variation in various components

Component	Maximum exergy destruction variation (%)
Compressor	82.9
Combustion chamber	86.07
Gas turbine	23.27

Based on the analysis done, following equations have been developed which express the exergy destruction in compressor, combustion chamber and gas turbine as a function of inlet air temperature. These equations have been checked for different ranges of inlet air temperatures and the results have been fairly satisfactory. These are represented as Equations 5.11, 5.12 and 5.13.

$$\dot{E}^{D}_{Comp} = 5.4154 - 0.1596\ln(T_a) \tag{5.11}$$

$$\dot{E}^{D}_{CC} = 110.1955 - 2.5805\ln(T_a) \tag{5.12}$$

$$\dot{E}^{D}_{GT} = 117.5092 + 3.9577\ln(T_a) \tag{5.13}$$

5.5 CONCLUSIONS

In the current study a 25 MW open cycle gas turbine power plant has been analyzed by using the energy and exergy balances for the individual components of the plant and the whole plant as well. Detailed exergy analysis has been done to calculate the values for various thermodynamic properties and net exergy flow rates have been calculated for all the components and exergy balances have been confirmed for individual components and the plant as a whole. Exergy destruction has been calculated for each component and various components have been ranked on the basis of exergy destruction which is taking place in them.

Effects of variation of thermodynamic variables such as compressor pressure ratio and inlet air temperature on the exergy destruction of various components has been analyzed. It has been found that increase in the compressor pressure ratio and inlet air temperature increases the exergy destruction in different components of the plant. However, this increase is different for various components.

A mathematical model has been developed in the form of equations which represent the exergy destruction taking place in various components of the plant as function of compressor pressure ratio and inlet air temperature. This mathematical model provides a robust tool to analyze the performance of various components in terms of thermodynamic properties.

The detailed thermodynamic analysis presented here along with the mathematical model developed is well suited to achieve the goal of effective energy resource use. It allows us to identify the location of energy wastage in the power plant. This information can be of great importance in the design of energy systems.

CHAPTER- 6

EXERGOECONOMIC ANALYSIS AND OPTIMIZATION OF OPEN CYCLE GAS TURBINE POWER PLANT

6.1 INTRODUCTION

The performance of a thermal system tends to deteriorate with the passage of time. There may be number of reasons which lead to the deterioration in the system performance and most of the time it is not possible to account for all of them. The analysis and diagnosis of the cause of performance deterioration in a complex thermal system is extremely difficult. One way to deal with this situation is to combine the principles of thermodynamics with economics to analyze the performance of thermal systems. During the last two to three decades many techniques based on the principles of exergo- economics have been developed to analyze the performance of different thermal systems.

Most of these techniques are cost accounting methods which use the average cost of exergy in terms of exergy [Lozano et. al. (1993), Valero et. al. (1994), Erlarch B. et. al. (1999), Torres C. et. al. (2002)] and average cost per unit of exergy [Kotas T. J. et al (1995), Bejan A. et. al. (1996) and Kwak H. et. al. (2003)].

In the current study, theory of exergetic cost has been used to analyze the performance of an open cycle gas turbine power plant. The theory of exergetic cost has been applied with the use of various mathematical tools. This technique involves two steps: 1) The first step involves the exergy based analysis of the open cycle gas turbine to identify the critical components where maximum exergy destruction takes place. This step has already been completed in the previous Chapter 5. 2) The second step involves the analysis of critical components, identified in the previous step, from exergetic cost point of view.

6.2 FORMULATION OF EXERGO- ECONOMIC EQUATIONS

Exergy based analysis is a powerful tool to analyze the performance of any thermal system. It allows the analysis of both quantity and quality of energy utilization in the system.

For a system operating at a steady state there may be a number of entering and exiting material streams as well as both heat and work interactions with the surroundings. Associated with these transfers of matters and energy are the exergy transfers into and out of the systems and the exergy destructions caused by the irreversibilities within the system. In exergy costing a cost is associated with each exergy stream.

The generalized exergy balance equation applicable to any thermal system has been formulated in the previous chapter as per Equation (5.2).

Unit exergetic cost can be assigned to every exergy stream and the cost balance equation corresponding to equation (5.2) can be written as:

$$\sum_e (c_e \dot{E}_e)_k + c_{w,k} \dot{W}_k = c_{q,k} \dot{E}_{q,k} + \sum_i (c_i \dot{E}_i)_k + \dot{Z}_k \tag{6.1}$$

where c denotes the unit exergetic cost associated with an exergy stream and the term $\dot{Z}_k$ represents all the charges associated with the operation and maintenance of the k_{th} component.

Unit costs per exergy unit of fuel and product for various components of the open cycle gas turbine are calculated as per Bejan et. al. (1996) and are given by Equations (6.2) and (6.3).

$$c_{F,k} = \frac{\dot{C}_{F,k}}{\dot{E}_{F,k}} \tag{6.2}$$

$$c_{P,k} = \frac{\dot{C}_{P,k}}{\dot{E}_{P,k}} \tag{6.3}$$

The purpose of the analysis done in the economic study is to provide inputs which can further be used in the exergoeconomic analysis and optimization. All calculations of exergetic costs have been done for the open cycle gas turbine power plant given in Figure 5.1.

6.3 COST BALANCE EQUATIONS FOR INDIVIDUAL COMPONENTS

6.3.1 AIR COMPRESSOR

The main purpose of the air compressor is to compress the air and raise its temperature. The cost balance equation for the air compressor is given by Equation (6.4).

$$c_1 \dot{E}_1 + c_w \dot{W}_{ac} + \dot{Z}_{ac} = c_2 \dot{E}_2 \tag{6.4}$$

where c_1, c_w, c_2 represent the unit exergetic costs for the inlet, work and exit streams for the air compressor. $\dot{Z}_{ac}$ denotes the capital investment and operation and maintenance cost of the air compressor. $\dot{E}_1$ and $\dot{E}_2$ represent the exergy rates for the inlet and exit streams. $\dot{W}_{ac}$ represents the exergy rate for work stream required to run the air compressor.

6.3.2 COMBUSTION CHAMBER

The main purpose of the combustion chamber is to provide combustion of fuel, natural gas in this case, to raise the temperature of compressed air coming from the air compressor. The hot air produced in the combustion chamber is further used to run the gas turbine. The cost balance equation for the combustion chamber is given by Equation (6.5).

$$c_2 \dot{E}_2 + c_5 \dot{E}_5 + \dot{Z}_{cc} = c_3 \dot{E}_3 \tag{6.5}$$

where c_2, c_3 and c_5 represent the unit exergetic costs for the inlet, exit and fuel streams of the combustion chamber. $\dot{Z}_{cc}$ denotes the capital investment and operation and maintenance cost of the combustion chamber. $\dot{E}_2$, $\dot{E}_3$ and $\dot{E}_5$ represent the exergy rates for the inlet, exit and fuel streams respectively.

6.3.3 GAS TURBINE

The main purpose of the gas turbine is to convert the thermal energy of hot air coming from the combustion chamber into mechanical energy which is further converted to electrical energy by means of an alternator. The cost balance equation for the gas turbine is given as:

$$c_3 \dot{E}_3 + \dot{Z}_{gt} = c_4 \dot{E}_4 + c_w \dot{W}_{ac} + c_w \dot{W}_{net} \tag{6.6}$$

where c_4 represents the unit exergetic cost of the exhaust from the gas turbine and $\dot{Z}_{gt}$ represents the capital investment and operation and maintenance cost of the gas turbine. $\dot{E}_4$ and $\dot{W}_{net}$ represent the exergy rate for the exhaust stream and the net work derived from the system.

6.3.4 OVERALL PLANT

In the exergo- economic analysis of the overall plant, two input and two exit streams have been considered: fuel and air entering the air compressor are taken as the input streams and exhaust from the gas turbine and output from gas turbine are taken as the exit streams. The exergetic cost balance for the overall plant is given by Equation (6.7).

$$c_1 \dot{E}_1 + c_5 \dot{E}_5 + \dot{Z}_{total} = c_4 \dot{E}_4 + c_w \dot{W}_{net} \tag{6.7}$$

where $\dot{Z}_{total}$ represents the capital investment and operation and maintenance cost for the overall plant.

6.4 RESULTS AND DISCUSSION

6.4.1 EXERGO- ECONOMIC ANALYSIS OF OPEN CYCLE GAS TURBINE POWER PLANT

In the current study, the first step is to calculate the costs associated with each stream entering and leaving the open cycle gas turbine system. Once these costs have been calculated, the next step involves analysis of variation in unit product cost of compressor and combustion chamber as functions of the compressor pressure ratio. Next we analyze the effect of turbine inlet temperature (TIT) on the unit product costs of the combustion chamber and gas turbine.

The optimization of the open cycle gas turbine power plant is achieved on the basis of a) compressor pressure ratio and b) turbine inlet temperature (TIT). To achieve the optimal values of the compressor pressure ratio, best balance is made between the unit product cost of the air compressor and the combustion chamber. To achieve the optimal value of the turbine inlet temperature, best balance is made between the unit product cost of the combustion chamber and the gas turbine.

The complete methodology adopted in the current study, for the exergo- economic analysis and optimization of an open cycle gas turbine power plant, is illustrated by means of a flow chart given in Figure 6.1. This flow chart explains in detail the step by step procedure used to analyze the open cycle gas turbine power plant from exergo- economic viewpoint.

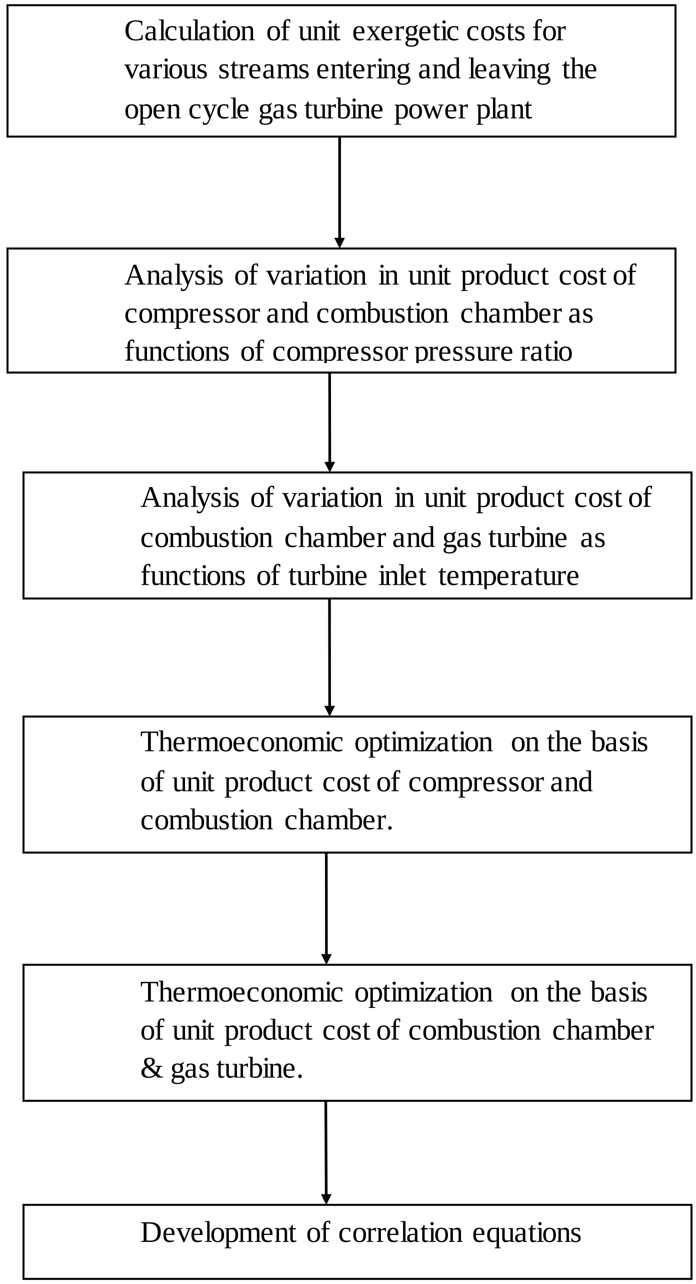


Figure 6.1 Flowchart depicting the methodology adopted in the Exergo- economic Analysis and Optimization of Open Cycle Gas Turbine Power Plant

The operating conditions for the base case design of 25 MW open cycle gas turbine power plant have been specified in Chapter- 5. For these conditions, the unit exergetic costs of various streams are given in Table 6.1.

Table 6.1 Unit Exegetic costs of Different Streams of a 25MW Open Cycle Gas Turbine Power Plant

Stream No.	Unit Cost *c* (US $/ MW)
1	0
2	5.146
3	4.28
4	4.34
5	4.34
c_w	4.46

These unit exergetic costs have been estimated based on the relevant cost data pertaining to gas turbine power plants given by Pauschert (2009).

In the current study, all the other variables are taken as constant and only the effects of compressor pressure ratio and turbine inlet temperature (TIT) on the economic performance of the open cycle gas turbine power plant have been analyzed.

The first step in the exergoeconomic process is to analyze the effect of compressor pressure ratio on the unit product cost of the compressor and combustion chamber. The range of compressor pressure ratios taken in the current study is from 3 to 9. Using programming in Excel, values for the unit product cost of air compressor and combustion chamber have been calculated at different values of compressor pressure ratios. The results of this analysis are given in Figures 6.2 and 6.3.

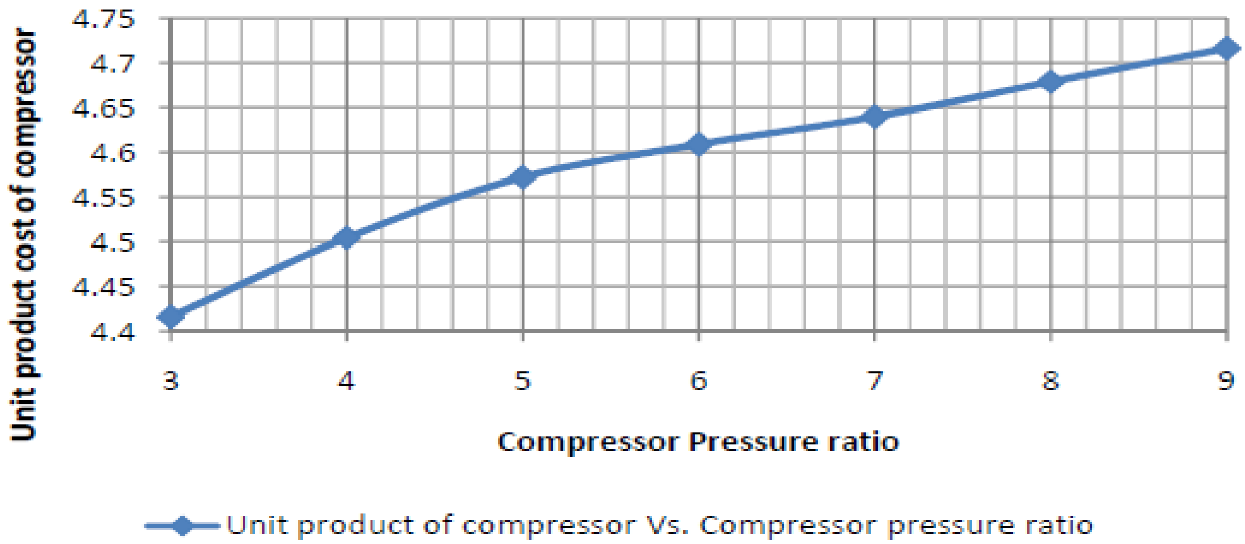


Figure 6.2 Variation in Unit Product cost of Air Compressor with variation in Compressor Pressure Ratio

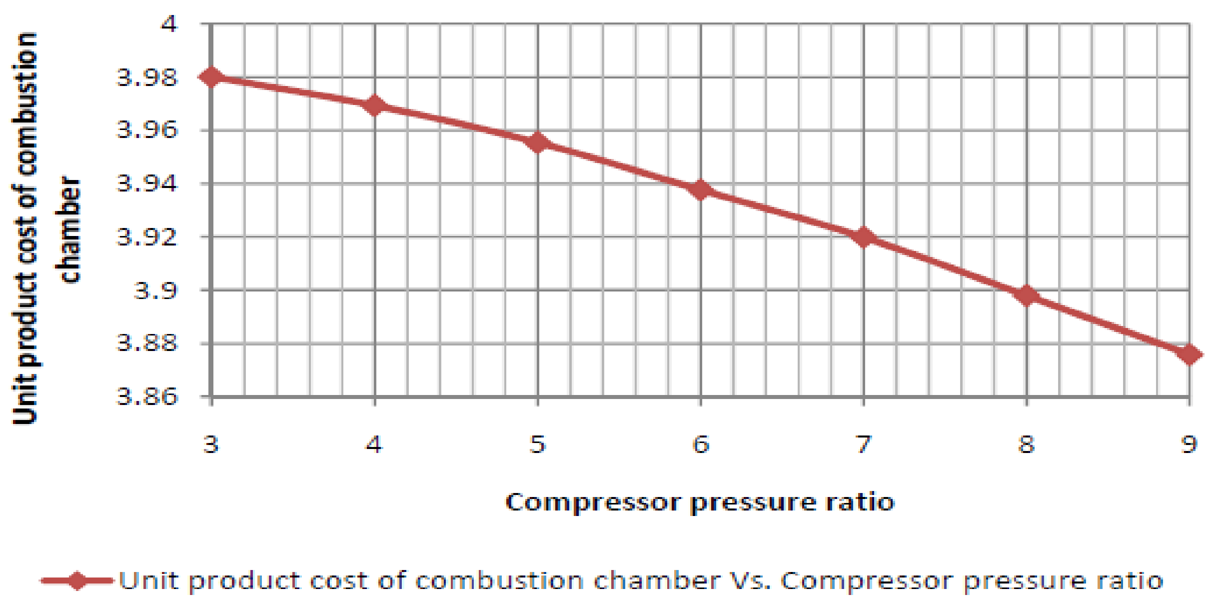


Figure 6.3 Variation in Unit Product cost of Combustion Chamber with variation in Compressor Pressure Ratio

From the analysis, it has been found that the unit product cost of air compressor increases with an increase in the compressor pressure ratio as illustrated in Figure 6.2. The primary reason for increase in the unit product cost of the compressor is that to provide air at higher compressor ratios, the compressor would have to be run at higher loads, thereby increasing the operation and maintenance cost of the compressor. Unit product cost of the air compressor increases from 4.43 US $/ MW at a compressor pressure ratio of 3 to 4.72 US $/ MW at a compressor pressure ratio of 9. A significant rise has been observed in values of unit product cost of the air compressor for the given range of compressor pressure ratios.

As illustrated in Figure 6.3, the unit product cost of combustion chamber decreases with an increase in the compressor pressure ratio. The main reason is that an increase in the compressor pressure ratio leads to the air, from the air compressor, entering the combustion chamber at a higher temperature. This results in lower fuel costs for the combustion chamber, as less fuel is required to generate the same amount of heat in the combustion chamber. Unit product cost of the combustion chamber decreases from 3.98 US $/ MW at a compressor pressure ratio of 3 to 3.88 US $/ MW at a compressor pressure ratio of 9. Results show that significant cost savings can be achieved in the combustion chamber by using higher compressor pressure ratios for the air compressor.

The next step in the exergo- economic analysis is to analyze the effect of turbine inlet temperature (TIT) on the unit product costs of the combustion chamber and gas turbine. The range of turbine inlet temperatures (TIT) taken in the current study is from 900 ^{0}C to 1500 ^{0}C. Programming in Excel has been used to calculate the values for unit product cost of combustion chamber and gas turbine at different values of turbine inlet temperatures (TIT). The results obtained from the analysis are given in Figures 6.4 and 6.5.

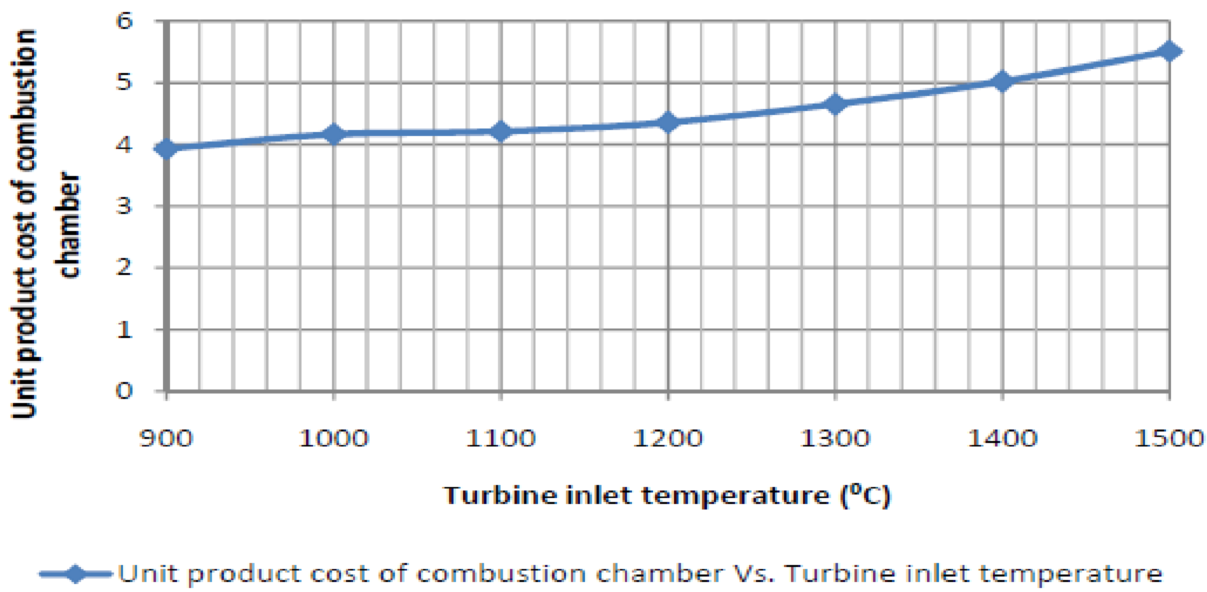


Figure 6.4 Variation in Unit Product Cost of Combustion Chamber with Variation in Turbine Inlet Temperature (TIT)

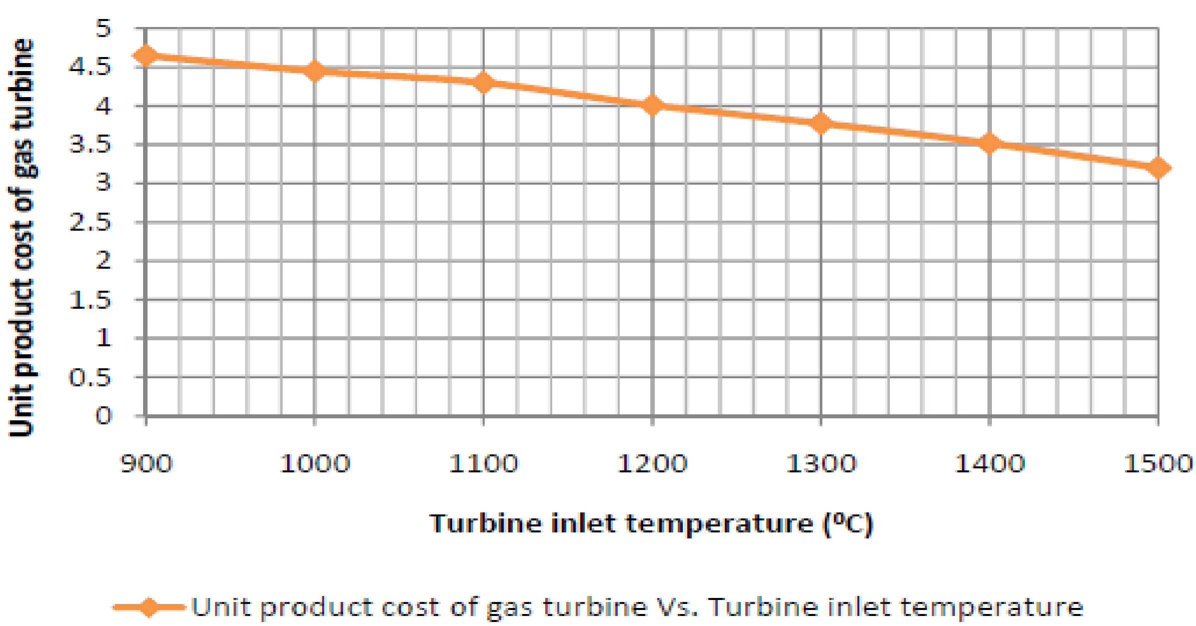


Figure 6.5 Variation in Unit Product Cost of Gas Turbine with Variation in Turbine Inlet Temperature (TIT)

From the analysis it has been observed that with an increase in the turbine inlet temperature, unit product cost of the combustion chamber increases. This happens because higher quantities of fuel are needed to generate products of combustion at higher temperatures. This results in higher fuel costs, thereby increasing the unit product costs for the combustion chamber. The unit product cost of the combustion chamber increases from 3.98 US $/ MW at turbine inlet temperature of 900 ^{0}C to 5.57 US $/ MW at turbine inlet temperature of 1500 ^{0}C. A significant rise is observed in values of unit product cost of the combustion chamber for the given range of turbine inlet temperatures.

As illustrated in Figure 6.5, the unit product cost of gas turbine decreases with an increase in the turbine inlet temperature (TIT). This happens because an increase in the turbine inlet temperature results in the gas turbine operating at higher exergetic efficiency, thereby leading to reduction in unit product costs. Unit product cost of the combustion chamber decreases from 4.62 US $/ MW at turbine inlet temperature of 900 ^{0}C to 3.28 US $/ MW for turbine inlet temperature of 1500 ^{0}C. Results show that significant cost savings can be achieved in the combustion chamber by using higher turbine inlet temperatures for the gas turbine.

6.4.2 OPTIMIZATION OF OPEN CYCLE GAS TURBINE POWER PLANT

For arriving at the optimal solution, results of exergo- economic analysis are used. Optimal solution has been achieved by combining the results of previous steps carried out in the exergo-economic analysis. Optimization has been achieved on the basis of two thermodynamic parameters: (a) Compressor pressure ratio and (b) Turbine inlet temperature (TIT). From the optimization process optimal values of these two thermodynamic parameters have been ascertained to achieve maximum savings in unit product costs.

The first optimal solution is achieved on the basis of compressor pressure ratio *(r)*. As shown earlier, the unit product cost of the air compressor increases with an increase in the compressor ratio and the unit product cost of the combustion chamber decreases with an increase in the compressor pressure ratio. These two results have been combined to arrive at the optimal value of the compressor pressure ratio at which the air compressor should be operated. The results of the optimization process are illustrated in Figure 6.6.

For the chosen range of compressor pressure ratios, the optimal value has been found to be 5.8. At this value, the unit product costs of the air compressor and combustion chamber are 4.6 US $/MW and 3.94 US $/MW. If further reduction is to be achieved in the cost for combustion chamber, more money would be needed to run the compressor at higher compressor pressure ratios. This would increase the unit product cost of the air compressor and hence the overall cost of power generation would increase.

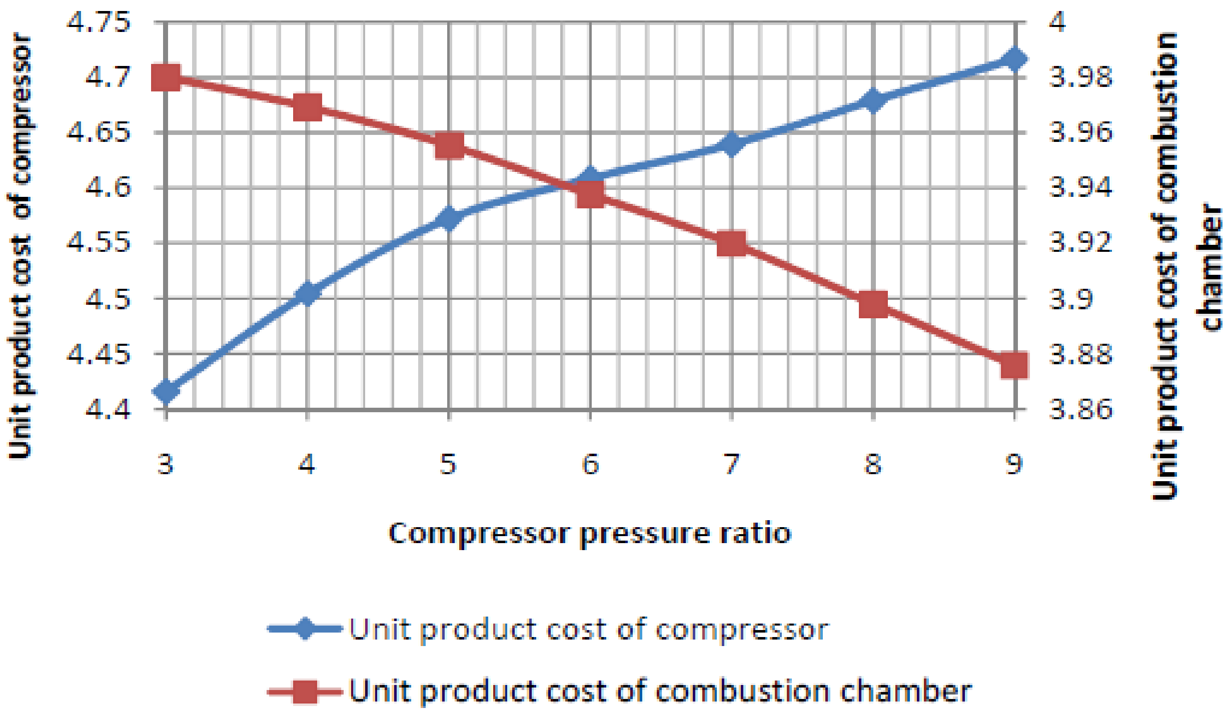


Figure 6.6 Combined effect of Compressor Pressure Ratio on Unit Product Cost of Air Compressor and Combustion Chamber

The next step in the analysis is to develop correlation equations for unit product cost of air compressor and combustion chamber as functions of compressor pressure ratio. The following relations have been derived, which put forward the unit product cost of compressor and unit product cost of the combustion chamber as a function of compressor pressure ratio *(r)*:

$$c_{ac} = 4.3151(1.0103)^{r} \tag{6.8}$$

$$c_{cc} = 4.0402(0.9955)^{r} \tag{6.9}$$

The second optimal solution is achieved on the basis of turbine inlet temperature (TIT). As shown earlier, the unit product cost of the combustion chamber increases with an increase in the turbine inlet temperature and the unit product cost of the gas turbine decreases with an increase in the turbine inlet temperature. These two results have been combined to arrive at the optimal value of the turbine inlet temperature, at which the products of combustion from the combustion chamber should be supplied to the gas turbine. The results of the optimization process are illustrated in Figure 6.7.

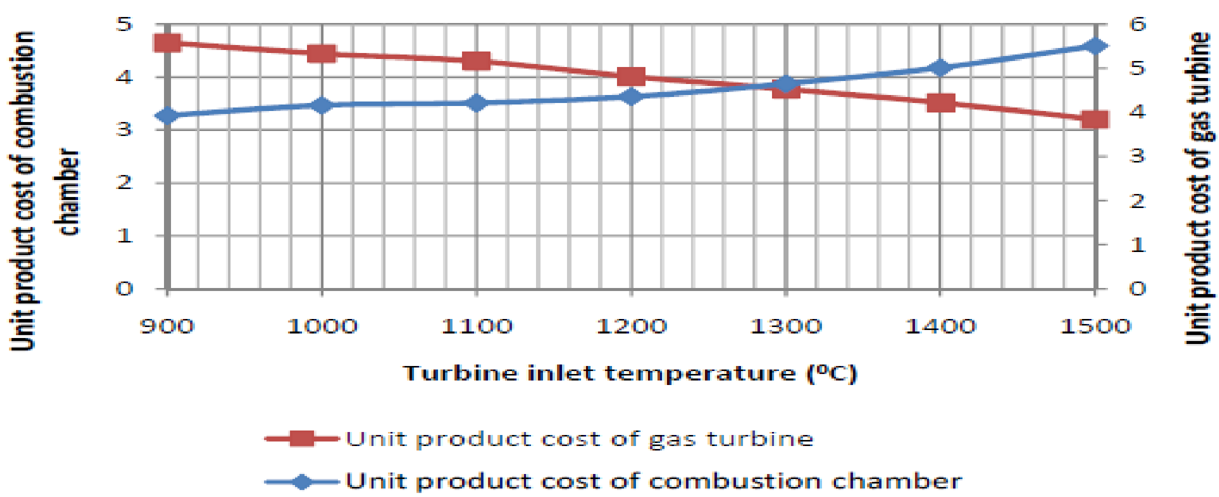


Figure 6.7 Combined Effect of Turbine Inlet Temperature (TIT) on Unit Product Cost of Combustion Chamber and Gas Turbine

For the chosen range of turbine inlet temperatures, the optimal value has been found to be 1280 ^{0}C. At this value, the unit product cost of the combustion chamber and gas turbine are 3.9 US $/MW and 4.7 US $/MW. If further reduction is to be achieved in the cost for gas turbine, more money would be needed, to provide products of combustion entering the gas turbine at a higher temperature from the combustion chamber. This would increase the unit product cost of the combustion chamber and hence the overall cost of power generation would increase.

The next step in the analysis is to develop correlation equations for unit product cost of combustion chamber and gas turbine as functions of turbine inlet temperature (TIT). The following relations have been derived, which put forward the unit product cost of unit product cost of the combustion chamber and gas turbine as a function of turbine inlet temperature:

$$c_{cc} = 2.3863(1.0005)^{(TIT)} \tag{6.10}$$

$$c_{gt} = 8.2648(0.9994)^{(TIT)} \tag{6.11}$$

The correlation equations between the unit product cost of different components and thermodynamic parameters have been checked for different ranges of thermodynamic parameters and the results obtained have been satisfactory.

6.5 CONCLUSIONS

The exergoeconomic technique presented in the current study is a powerful tool to analyze the performance of an open cycle gas turbine power plant. It combines the principles of thermodynamic and economics. This technique is able to identify all the cost sources in any thermal system. It provides a better analysis of the cost consumption in any component of the thermal system by assigning unit costs to all the streams entering and leaving that component.

From the current study, many conclusions can be drawn. Combustion chamber and gas turbine are the components with maximum exergy destruction. Hence they need to be carefully analyzed to achieve maximum cost savings. Critical thermodynamic parameters have been identified which play a significant role in affecting the unit product costs of these two components.

Compressor pressure ratio has been identified as the critical parameter which provides good scope to achieve cost savings in the combustion chamber. Optimization has been done to arrive at the optimal value of compressor pressure ratio of 5.8 at which air should be supplied to the combustion chamber from the air compressor. At this compressor pressure ratio, the unit product cost of the air compressor and combustion chamber are found to be 4.6 US $/MW and 3.94 US $/MW.

Turbine inlet temperature (TIT) has been identified as the critical parameter which affects the unit product cost of the combustion chamber and gas turbine significantly. Optimization has been done to arrive at the optimal value of turbine inlet temperature of 1280 ^{0}C at which products of combustion from the combustion chamber should be supplied to the gas turbine. At this temperature the unit product costs for the combustion chamber and gas turbine are found to be 3.9 US $/MW and 4.7 US $/MW.

Further, correlation equations have been developed for unit product cost of air compressor and combustion chamber as functions of compressor pressure ratio and also for unit product cost of combustion chamber and gas turbine as functions of turbine inlet temperature (TIT). These correlation equations quantify the unit product costs of components in terms of thermodynamic variables under consideration.

CHAPTER- 7

COMPARISON OF COAL FIRED AND OPEN CYCLE GAS TURBINE POWER PLANTS

7.1 INTRODUCTION

The coal fired thermal power plants came into existence in the year 1882. Since then, the process of generating electricity by using coal as the primary fuel has undergone many changes. The capacity of these plants to generate electricity has also increased manifolds. In India alone, 60% of the total power generation is done with coal fired thermal power plants. With advancements in technology over the years, the performance of coal fired thermal power plants has improved significantly. Typically, a coal fired thermal power plant works in the efficiency range of 33-48%. Many researchers have tried to improve the efficiency of a coal fired thermal power plant beyond this range. However, all such efforts have proven to be useless from economic considerations.

Even today, the coal fired thermal power plants account for the maximum amount of electricity generation among all the different types of power plants. However, in the recent past, it has been observed that these power plants are no longer a viable option to continue producing electricity. There are many reasons for the strategic shift in the power generation sector away from the coal fired thermal power plants. Some of these reasons for this shift include scarcity of coal, environmental impact of coal fired thermal power plants with huge emissions of carbon dioxide, high installation cost, depleting water resources etc.

As mentioned above, in the recent years there has been an understanding among the scientific community to look beyond the coal fired thermal power plant and look for better alternatives. One of these alternatives is the gas turbine power plant which takes care of all the problems which are associated with a coal fired thermal power plant. Hence it is a much more viable option to generate electricity as compared to a coal fired thermal power plant. In India alone, 10.5% of the total thermal power capacity is generated by gas turbine power plants and more emphasis is being laid on developing better technologies by which electricity can be generated using gas turbine power plants. Over the years, lots of modifications have been made to the conventional gas turbine power plants to improve their performance. These include the use of heat recovery steam generator (HRSG) and cogeneration system etc.

In the current study, both coal fired and open cycle gas turbine power plants have been analyzed from thermodynamic and economic viewpoint. In this chapter, a brief comparison is presented between the two from thermodynamic and economic considerations.

7.2 COMPARISON OF COAL FIRED AND OPEN CYCLE GAS TURBINE POWER PLANT FROM THERMODYNAMIC CONSIDERATIONS

A detailed energy and exergy analysis has been presented in the current study for both coal fired and open cycle gas turbine power plants. One of the major advantages of performing the detailed exergy analysis is that it is able to pinpoint and identify the major sources of exergy destruction within the power plant. For the coal fired thermal power plant, boiler and steam turbine have been identified as the components where maximum exergy destruction takes place. Boiler is the component where combustion of fuel, coal in this case, takes place to generate the required amount of energy to convert water into steam. Steam turbine is the component which is supplied high temperature steam from the boiler for conversion of thermal energy into mechanical energy. This mechanical energy is further converted into electrical energy by using an alternator which is coupled to the shaft of the steam turbine.

Specific thermodynamic variables have been identified which affect the performance of these two components significantly. For the boiler, hot air temperature has been identified as the important parameter and for the steam turbine inlet steam temperature has been identified as the critical parameter. Performance analysis has been done for the boiler and steam turbine in terms of the variation in their exergetic efficiency as functions of hot air temperature and inlet steam temperature respectively. From the analysis, it has been observed that the exergetic efficiency of the boiler increases with an increase in the hot air temperature. Also, the exergetic efficiency of the steam turbine increases with an increase in the inlet steam temperature. The reasons for the same have already been mentioned in Chapter- 3.

For the open cycle gas turbine power plant, combustion chamber and gas turbine have identified as the components where maximum exergy destruction takes place. In a gas turbine power plant, combustion chamber does the function similar to the boiler in a coal fired thermal power plant. The combustion chamber is fed compressed air from the air compressor and combustion of fuel (natural gas) takes place to produce the desired products of combustion. These products of combustion are used to run the gas turbine which converts the thermal energy into mechanical energy. The function of a gas turbine in an open cycle gas turbine is similar to that of a steam turbine in a coal fired thermal power plant. It is used to convert the thermal energy of products of combustion, obtained from the combustion chamber, into electrical energy by using an alternator which is coupled to the shaft of the gas turbine.

Specific thermodynamic variables have been identified which affect the performance of the combustion chamber and gas turbine significantly. For the combustion chamber, compressor pressure ratio has been identified as the critical parameter and for the gas turbine, turbine inlet temperature (TIT) has been identified as the important parameter. Performance analysis has been done for the combustion chamber and gas turbine in terms of exergy destruction taking place in them as functions of compressor pressure ratio and turbine inlet temperature respectively. From the analysis, it has been observed that the exergy destruction in the combustion chamber increases with an increase in the compressor pressure ratio and the exergy destruction in the gas turbine increases with an increase in the turbine inlet temperature. The reasons for the same have already been mentioned in Chapter- 5.

Comparison between the coal fired and open cycle gas turbine power plant shows that even though they are completely different from each other in their setup and working, certain similarities can be drawn between the two in terms of components performing similar functions. Both hold important place in the current electricity generation scenario and are extremely important.

7.3 COMPARISON OF COAL FIRED AND OPEN CYCLE GAS TURBINE POWER PLANTS FROM ECONOMIC CONSIDERATIONS

From the exergy analysis, for a coal fired thermal power plant, boiler and steam turbine have been identified as the components with maximum exergy destruction. These are the components which provide maximum scope for cost savings. In the exergo- economic analysis, unit costs of product and fuel have been calculated for the boiler and steam turbine for the base case design.

From the base case design, optimization has been achieved to arrive at the optimal values of various thermodynamic parameters. For the boiler, hot air temperature has been considered as the critical parameter and the optimal value of hot air temperature has been achieved as a best balance between the unit product cost of the boiler and the air pre- heater. This value of hot air temperature provides maximum cost saving and if air is to be supplied at higher temperature, unit product cost of boiler decreases but the unit product cost of air pre- heater shows a significant rise. Reasons for the same have been discussed in detail in Chapter- 4.

For the steam turbine, inlet temperature has been considered as the critical parameter and the optimal value of inlet steam temperature has been achieved as a best balance between the unit product cost of the boiler and the steam turbine. This value of inlet steam temperature provides maximum cost saving and if steam is to be supplied at higher temperature, unit product cost of steam turbine decreases but the unit product cost of boiler shows a significant rise.

From the exergy analysis, for an open cycle gas turbine power plant, combustion chamber and gas turbine have been identified as the components with maximum exergy destruction. These are the components which provide maximum scope for cost savings. In the exergo- economic analysis, unit costs of product and fuel have been calculated for the combustion chamber and gas turbine for the base case design.

From the base case design, optimization has been achieved to arrive at the optimal values of various thermodynamic parameters. For the combustion chamber, compressor pressure ratio has been considered as the critical parameter and the optimal value of compressor pressure ratio has been achieved as a best balance between the unit product cost of the combustion chamber and the air compressor. This value of compressor pressure ratio provides maximum cost saving and if air is to be supplied at higher compressor pressure ratio, unit product cost of boiler decreases but the unit product cost of air compressor shows a significant rise.

For the gas turbine, turbine inlet temperature (TIT) has been considered as the critical parameter and the optimal value of turbine inlet steam temperature has been achieved as a best balance between the unit product cost of the combustion chamber and the gas turbine. This value of turbine inlet temperature provides maximum cost saving and if products of combustion are to be supplied at higher temperature, unit product cost of gas turbine decreases but the unit product cost of combustion chamber shows a significant rise. Reasons for the same have been discussed in detail in Chapter- 6.

7.4 CONCLUSIONS

As can be seen from the current study, major differences exist in the setup and working cycles of a coal fired and an open cycle thermal power plant. However, some similarities can also be drawn between the two, in terms of different components performing the same functions in both.

In both the coal fired and open cycle gas turbine power plant, combustion process is carried out with the help of boiler and combustion chamber respectively. Also, conversion of thermal energy into mechanical energy is done with the help of a steam turbine and a gas turbine respectively. Optimization for both the plants has been achieved in similar manner by obtaining the optimal values of various thermodynamic parameters.

Both types of plants, owing to their extensive usage in today’s electricity generation scenario, are extremely important and offer tremendous scope for improvement in thermodynamic efficiency and cost reduction. The main aim of the present study is to achieve higher thermodynamic efficiency and reduction in cost for both types of power plants. For both the plants, components with maximum exergy destruction have been identified and cost reduction has been achieved for these components in terms of different thermodynamic parameters.

CHAPTER- 8

CONCLUSIONS AND FUTURE SCOPE OF WORK

8.1 CONCLUSIONS

The current energy crisis which the world is facing at present has put tremendous stress on the researchers to improve the efficiencies of the existent power generation systems. This has led to the development of various analysis techniques using which the performance of power generation systems can be improved. A new technique which combines the principles of thermodynamics and economics has been developed to analyze and improve the performance of thermal systems.

This technique is based on the concept of exergy and it predicts the performance of a power generation system by quantifying the exergy destruction in various components. Unlike energy, exergy is not conserved in a thermodynamic process but is destroyed in the system. Many authors and researchers have used the concept of exergy and combined it with the concepts of economics to analyze the performance of different types of thermal systems.

In the current study, the concept of exergy has been applied to analyze the performance of a 210 MW coal fired thermal power plant and a 25 MW open cycle gas turbine power plant. For both the plants, components with maximum exergy destruction have been identified. Based on the theory of exergetic cost, thermo- economic evaluations have been done for components with maximum exergy destruction to reduce their unit product cost. The major conclusions emerging from the study are as follows:

- It has been observed that, in a coal fired thermal power plant, boiler and steam turbine are the components where maximum exergy destruction takes place. These two components have been analyzed further to determine the critical thermodynamic parameters which affect their performance significantly.

- For the boiler, temperature of hot air, coming from the air pre- heater, has been identified as the critical parameter which affects the performance of a boiler significantly. Analysis has been done to study the effect of hot air temperature on the exergetic efficiency of the boiler. It has been found that, with an increase in the hot air temperature, the exergetic efficiency of the boiler increases. The exergetic efficiency of the boiler increases from 21.27% to 21.42% for a 20 ^{0}C rise in the hot air temperature.

- For the steam turbine, inlet steam temperature, coming from the boiler, has been identified as the critical parameter. It has a major impact on the performance of the steam turbine. Analysis has been done to study the effect of inlet steam temperature on the exergetic efficiency of the steam turbine. It has been found that exergetic efficiency of the steam turbine increases with an increase in the inlet steam temperature. For a 20 ^{0}C rise in the inlet steam temperature, the exergetic efficiency of the steam turbine is found to increase from 71.1% to 73%.

- In the exergo- economic analysis for the boiler, variations in the unit product cost for the boiler and air pre- heater have been analyzed as functions of the hot air temperature. Optimization has been done, to achieve the optimal value of hot air temperature of 143.6 ^{0}C, at which air should be supplied to the boiler. This value of hot air temperature provides maximum reduction in the overall cost of power generation. If higher values of hot air temperatures are employed, even though the unit product cost of boiler reduces, the overall cost of power generation increases significantly.

- In the exergo- economic analysis of the steam turbine, variations in the unit product cost for the boiler and steam turbine have been analyzed as functions of the inlet steam temperature. Optimization has been done to achieve the optimal value of inlet steam temperature of 540 ^{0}C, at which steam should be supplied to the steam turbine. This value of inlet steam temperature provides maximum reduction in the overall cost of power generation. If higher values of inlet steam temperatures are employed, even though the unit product cost of steam turbine reduces, the overall cost of power generation increases significantly.

- Detailed exergy analysis has been done for an open cycle gas turbine power plant which shows that maximum exergy destruction takes place in the combustion chamber followed by the gas turbine and the air compressor. Effects of critical thermodynamic variables have been analyzed, on the performance of various components of the power plant, in terms of variations in their exergy destructions. Compressor pressure ratio and inlet air temperature are the thermodynamic variables whose effect has been analyzed on the performance of various components.

- It has been observed from the analysis that an increase in the compressor pressure ratio leads to significant increase in the exergy destruction in the compressor and combustion chamber. The exergy destruction in the combustion chamber increases from 4.12 MW at a compressor pressure ratio of 3.5 to 7.09 MW at a compressor pressure ratio of 15.5. The overall increase in the exergy destruction for the chosen range of compressor pressure ratios is 70%. For the combustion chamber, the exergy destruction increases from 80.2 MW at a compressor pressure ratio of 3.5 to 160 MW at a compressor pressure ratio of 15.5. The overall increase in the exergy destruction of the combustion chamber for the chosen range of compressor pressure ratios is 90%. For the same turbine inlet temperature, compressor pressure ratio has practically no effect on the exergy destruction in the gas turbine.

- From the analysis, it has been observed that an increase in the inlet air temperature tends to decrease the exergy destruction in the air compressor and combustion chamber but the exergy destruction in the gas turbine increases. For the compressor, the exergy destruction decreases from 4.92 MW at 285 K to 4.18 MW at 299 K. The overall decrease in the exergy destruction in the compressor is found to be 83% for the chosen range of inlet air temperatures. For the combustion chamber, the exergy destruction increases from 102 MW at 285 K to 86.9 MW at 299 K. The overall decrease in the exergy destruction for the combustion chamber is found to be 86% for the chosen range of inlet air temperatures. For the gas turbine, the exergy destruction increases from 103

MW at 285 K to 127 MW at 299 K. The overall increase in the exergy destruction for the gas turbine is found to be 23% for the chosen range of inlet air temperatures.

- A mathematical model has been developed, in the form of equations, which represent the exergy destruction taking place in various components of the plant as functions of compressor pressure ratio and inlet air temperature. This mathematical model provides a robust tool to analyze the performance of various components of the plant in terms of critical thermodynamic parameters.

- Exergo- economic analysis and optimization has been done for the open cycle gas turbine power plant. During the exergo- economic analysis, unit product cost has been calculated for each component of the plant. The optimization of the open cycle gas turbine power plant is achieved on the basis of a) compressor pressure ratio and b) turbine inlet temperature (TIT).

- To achieve the optimal values of the compressor pressure ratio a best balance has been made between the unit product cost of the air compressor and the combustion chamber. To achieve the optimal value of the turbine inlet temperature a best balance has been made between the unit product cost of the combustion chamber and the gas turbine.

- From the analysis, it has been observed that the unit product cost of air compressor increases with an increase in the compressor pressure ratio. Unit product cost of the air compressor increases from 4.43 US $/ MW at a compressor pressure ratio of 3 to 4.72 US $/ MW at a compressor pressure ratio of 9. The unit product cost of combustion chamber decreases with an increase in the compressor pressure ratio. The unit product cost of the combustion chamber decreases from 3.98 US $/ MW at a compressor pressure ratio of 3 to 3.88 US $/ MW at a compressor pressure ratio of 9.

  These two results have been combined to arrive at the optimal value of the compressor pressure ratio, at which the air compressor should be operated. For the chosen range of compressor pressure ratios, the optimal value has been found to be 5.8. At this value, the unit product costs of the air compressor and combustion chamber are 4.6 US $/MW and 3.94 US $/MW.

- From the analysis, it has been observed that with an increase in the turbine inlet temperature, unit product cost of the combustion chamber increases. The unit product cost of the combustion chamber increases from 3.98 US $/ MW at turbine inlet temperature of 900 ^{0}C to 5.57 US $/ MW at turbine inlet temperature of 1500 ^{0}C. The unit product cost of gas turbine decreases with an increase in the turbine inlet temperature (TIT). Unit product cost of the combustion chamber decreases from 4.62 US $/ MW at turbine inlet temperature of 900 ^{0}C to 3.28 US $/ MW for turbine inlet temperature of 1500 ^{0}C.

  These two results have been combined to arrive at the optimal value of the turbine inlet temperature (TIT), at which the products of combustion from the combustion chamber should be supplied to the gas turbine. The optimal value has been found to be 1280 ^{0}C. At this value, the unit product cost of the combustion chamber and gas turbine are 3.9 US $/MW and 4.7 US $/MW.

  A mathematical model has been developed in terms of correlation equations for the unit product cost of air compressor and combustion chamber as functions of compressor pressure ratio and also for unit product cost of combustion chamber and gas turbine as functions of turbine inlet temperature (TIT). These correlation equations quantify the unit product costs of components in terms of thermodynamic variables under consideration.

- In the current study, a brief comparison has been made between the coal fired and open cycle gas turbine power plants. It can be seen that major differences exist between the two in terms of their set up and working cycles. However, some similarities can also be drawn between them in terms of different components performing similar functions.

8.2 CONTRIBUTION OF CURRENT STUDY

The current study encompasses the performance analysis of coal fired and open cycle gas turbine power plant using the concept of exergoeconomics. As has been found during the literature review, very little work has been done in the analysis of performance of power plants using specific thermodynamic variables and their effect on the economic performance of the power plants. Current study tries to bridge that gap by specifically analyzing the economic performance of coal fired and open cycle gas turbine power plants in terms of thermodynamic variables.

8.3 FUTURE SCOPE OF WORK

The detailed thermodynamic analysis, parametric study and optimization of coal fired and open cycle gas turbine power plant which have been presented here, are well suited to fulfill the goal of achieving maximum efficiency at minimum cost. This information is very useful and the work can be further extended to other thermal energy conversion systems. Following studies may further be undertaken:

1. There are more thermodynamic factors which affect the performance of a power plant. Further studies may involve analysis based on those parameters.
2. Multi-objective optimization of power plants can be extended using more than two objectives simultaneously and using other evolutionary techniques.
3. Analysis at different operating loads for a power plant can be done.
4. Time based analysis can be taken up for future studies.

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APPENDIX- A

Sample calculations for 210 MW Steam Turbine Power Plant

STREAM NO	P	T	m	h	s	ENERGY (KW)	EXERGY (MW)
1	139.45	536	178.33	3424.06	6.5211	610612.6	259.2943
2	38.34	364	174.1	3131.45	6.6629	545185.4	194.7165
3	35.2	535.5	174.1	3530.823	7.2444	614716.3	233.5566
4	36.34	226.8	4.2	2928.45	8.65	12299.49	1.314717
5	34.37	194.31	10.1	2885.75	8.69	29146.08	2.60784
6	7.09	332	163.94	3126.266	7.407	512520	145.5229
7	0.828	53.9	149.36	255.699	0.753	38191.2	5.115952
8	0.828	35.8	149.36	210.684	0.693	31467.76	1.109221
9	0.786	77.9	30.92	326.164	1.0503	10084.99	0.451154
10	3.971	85.01	10.73	356.25	1.134	3822.563	0.207125
11	6.7355	121.4	0.75	510.063	1.5424	382.5473	0.036982
12	18.04	56.7	149.166	314.979	1.01328	46984.16	2.182091
13	0.016	81.9	157.5	2654.271	9.2028	418047.7	20.27535
14	0.016	118.4	163.611	2723.34	9.388	445568.4	18.94726
15	179.265	165.7	164.04	710.3283	1.9788	116522.3	19.2387
16	179.265	172.2	51.08	738.649	2.4021	37730.19	0.882551
17	179.265	191.1	165.44	820.36	2.293	135720.4	21.84842
18	179.265	199	15.424	855.03	2.2966	13187.98	2.554849
19	179.265	223.3	170.3	963.37	2.5203	164061.9	35.11015
20	1.01325	146.3	164.77	7500	2.93	1235775	1090.546
21	1.01325	128	163.45	7200	2.76	1176840	1041.198
22	1.01325	30.9	44.8	129.59	0.4491	5805.632	0.012131
23	0.89	41.8	44.5	175.13	0.5693	7793.285	0.41706
24	1.01325	298.15	19.34			446367.2	454.5657

APPENDIX- B

Sample Calculation for Optimization of Boiler for Steam Turbine Power Plant

EEFECT OF HOT AIR TEMPERATURE ON COST OF PRODUCT OF BOILER							
Temp	**Pressure**	c_4	ε_b		$E_{F,B}$	m_{coal}	c_{ap}
133	1.058	0.195024	0.212677	494128.8	448519	19.43323	0.671768
134	1.058	0.194965	0.212755	493946	448353.1	19.42604	0.675144
135	1.058	0.194906	0.212834	493763.3	448187.3	19.41886	0.678536
136	1.058	0.194847	0.212913	493580.7	448021.5	19.41168	0.681946
137	1.058	0.194788	0.212991	493398.2	447855.8	19.4045	0.685373
138	1.058	0.194729	0.21307	493215.7	447690.2	19.39732	0.688817
139	1.058	0.19467	0.213149	493033.2	447524.6	19.39015	0.692278
140	1.058	0.194611	0.213228	492850.9	447359.1	19.38297	0.695757
141	1.058	0.194552	0.213307	492668.6	447193.6	19.37581	0.699253
142	1.058	0.194493	0.213386	492486.4	447028.2	19.36864	0.702767
143	1.058	0.194434	0.213465	492304.2	446862.9	19.36148	0.706299
144	1.058	0.194375	0.213544	492122.1	446697.6	19.35431	0.709848
145	1.058	0.194316	0.213623	491940.1	446532.4	19.34716	0.713415
146	1.058	0.194257	0.213702	491758.2	446367.2	19.34	0.717
147	1.058	0.194198	0.213781	491576.2	446202	19.33284	0.720585
148	1.058	0.194139	0.21386	491394.3	446036.9	19.32569	0.724188
149	1.058	0.19408	0.213939	491212.5	445871.9	19.31854	0.727809
150	1.058	0.194021	0.214018	491030.8	445706.9	19.31139	0.731448
151	1.058	0.193962	0.214098	490849.1	445542	19.30425	0.735105
152	1.058	0.193904	0.214177	490667.5	445377.2	19.2971	0.738781
153	1.058	0.193845	0.214256	490485.9	445212.4	19.28996	0.742475

APPENDIX- C

Sample calculation for optimization of steam turbine

Effect of Inlet Steam Temperature on Turbine Cost and boiler cost					
STEAM TEMP.	cw	c1	Exergetic Efficiency		
530	1.28281669	0.689707087	0.71234		
531	1.263858808	0.703782741	0.71305234		
532	1.245181092	0.718145654	0.713765392		
533	1.226779401	0.729081883	0.714479158		
534	1.21463307	0.740184653	0.715193637		
535	1.202607	0.7514565	0.715908831		
536	1.1907	0.7629	0.716624739		
537	1.178793	0.7743435	0.717484689		
538	1.16700507	0.785958653	0.718345671		
539	1.155335019	0.797748032	0.719207685		
540	1.143781669	0.809714253	0.720070735		
541	1.132343852	0.821859967	0.72093482		
542	1.115358695	0.830078566	0.721799941		
543	1.098628314	0.838379352	0.722666101		
544	1.082148889	0.846763145	0.723533301		
545	1.065916656	0.855230777	0.724618601		
546	1.049927906	0.863783085	0.725705528		
547	1.034178988	0.872420915	0.726794087		
548	1.018666303	0.881145125	0.727884278		
549	1.003386308	0.889956576	0.728976104		
550	0.988335514	0.898856142	0.730069568		

APPENDIX- D

Sample calculation for an Open Cycle Gas Turbine Power Plant

State	$\dot{m}$ (kg/s)	P (bar)	T (K)	$\dot{E}^T$	$\dot{E}^M$	$\dot{E}^C$	$\dot{E}$
1	212.95	0.981	293	0	0	0	0
2	212.95	4.2	481.6	10.52	26.146	0	36.916
3	216.66	1.01325	1123	108.768	23.89	0.7488	133.406
4	216.66	1.1	817.6	55.8	1.5	0.7488	58.04
5	3.71	22	293	0	0.824	190.53	191.39
6	...	...	...	...	...	...	40.63
7	...	...	...	...	...	...	25

Component	$\dot{E}^W$	$\dot{E}^C$	$\dot{E}^T$	$\dot{E}^M$	$\dot{E}^D$	**Monetary Flow rates** $\dot{Z}$
Compressor	-40.363	0	10.52	26.146	3.697	**9.92**
Combustion chamber	0	-189.78	98.248	-3.083	94.615	**0.51**
Gas Turbine	66.498	0	-52.698	-22.39	8.59	**71.38**
Other equipment						**33.53**
Overall Plant	26.135	-189.78	56.07	0.673	106.902	**135.89**

APPENDIX- E

Sample calculation for optimization of Open Cycle Gas Turbine Power Plant

Effetc of Compressor pressure ratio on unit product cost of each component								
Compressor Pressure Ratio	$\dot{c}$ **Compressor**	$\dot{C}$ **CC**	$\dot{C}$ **GT**	$\dot{C}$ **OP**				
3	4.416	3.98		4.14				
4	4.50432	3.969254		4.2021				
5	4.5718848	3.955362		4.483641				
6	4.608459878	3.937562		4.836055				
7	4.63933656	3.919843		5.308054				
8	4.67877092	3.897892		5.94502				
9	4.716201088	3.875791		6.836773				
Effetc of Turbine Inlet Temperature on unit product cost of each component								
900		3.926549	4.648947					
1000		4.15948	4.4462					
1100		4.21	4.3					
1200		4.35314	4.00846					
1300		4.648283	3.772362					
1400		5.015032	3.516218					
1500		5.510016	3.200814					

LIST OF PUBLICATIONS OUT OF THESIS

a) International Journals

1. "Thermoeconomic Optimization of a Boiler Used in a Coal Fired Thermal Power Plant Based on Hot Air Temperature", International Journal of Recent advances in Mechanical Engineering (IJMECH) Vol.4, No.2, May 2015.

2. "Optimization of an open cycle gas turbine power plant using exergoeconomics", International Journal of Recent advances in Mechanical Engineering (IJMECH) Vol.4, No.4, and November 2015.

3. "Exergy Based Analysis of An Open Cycle Gas Turbine Power Plant", Canadian Journal of Basic and Applied Sciences, Vol. 03(10), 273-282, October 2015

4. "Optimization of a turbine used in coal fired thermal power plants based on inlet steam temperature using thermoeconomics", International Journal of Recent advances in Mechanical Engineering (IJMECH) Vol.4, No.4, November 2015.

5. "Exergoeconomic Analysis of a Boiler for a Coal Fired Thermal Power Plant", American Journal of Mechanical Engineering, 2014, Vol. 2, No. 5, 143-146.

6. "Exergy Based Evaluation of Coal Based Thermal Power Plants: A Review", International Journal of Emerging Technology and Advanced Engineering, Volume 3, Special Issue 2, January 2013.

b) International Conferences

1. “Graph Theoretic Approach for Performance Evaluation of a Boiler”, International Conference on Machine Intelligence and Research, SMVDU, Katra, Jammu, 2013.
2. “Diagraph & Matrix Method Using Exergy for Evaluation of Coal Based Thermal Power Plants in India”, International Conference on Machine Intelligence and Research, SMVDU, Katra, Jammu, 2013.

3. “Energy Analysis: An Efficient Tool for Evaluation of Power Plants in India”, International Conference on Advanced Technologies for Research & Product Development, 2012.

c) National Conferences

1. “Energy Method for Performance Evaluation of a Boiler in a Coal Fired Thermal Power Plant”, Trends and Advances in Mechanical Engineering, YMCA University of Science & Technology, Faridabad, 2012.