# Exergetic Evaluation of single pressure HRSG based Combined Cycle Gas Power plant

*A Major Thesis Submitted in Partial Fulfilment of the requirements for the award of the degree of*

**MASTER OF TECHNOLOGY**

**IN**

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*Submitted by:*

**ASHUTOSH MISHRA**

**(2K15/THE/04)**

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*Under the Guidance of*

**Dr. B. B. ARORA**

**Department of Mechanical Engineering,**
**DELHI TECHNOLOGICAL UNIVERSITY,**
Shahbad Daulatpur, Main Bawana Road, Delhi-110042

# DECLARATION

I hereby declare that the work which being presented in the major thesis entitled **"Exergetic Evaluation of single pressure HRSG based Combined cycle Gas Power plant"** in the partial fulfilment for the award of the degree of Master of Technology in **"Thermal Engineering"** submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of **Dr. B. B. ARORA,** Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever.

Place: Delhi **ASHUTOSH MISHRA**

Date**:** **Roll No. 2K15/THE/04**

# CERTIFICATE

This is to certify that **ASHUTOSH MISHRA**, (Roll no. **2K15/THE/04**), student of M.Tech., THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled **"Exergetic Evaluation of single pressure HRSG based Combined Cycle Gas Power plant"** under my guidance towards the partial fulfilment of the requirements for the award of the degree of Master of Technology under my guidance and supervision.

**DR. B. B. ARORA**
Department of Mechanical Engineering
Delhi Technological University
Delhi-110042

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It is said that gratitude is a virtue. This part is dedicated to special thanks that I would like to deliver to the people who helped me in making the fulfilment of this thesis project possible.

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**ASHUTOSH MISHRA**
**(2K15/THE/04)**

# ABSTRACT

The increasing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on first law of thermodynamics only. The real useful energy loss cannot be justified by the fist law of thermodynamics, because it does not differentiate between the quality and quantity of energy. The exergy analysis (second law analysis) is used for providing information about the losses qualitatively as well as quantitatively along with their locations. Exergetic (thermodynamic) optimization improves the performance of a system by reducing the exergetic inefficiencies (exergy destruction and exergy losses) and increasing exergetic efficiency. In this analysis shows exergy efficiency is less at each and every point of unit equipment's. Also presents major losses of available energy at combustor, HRSG and gas turbine. The primary objectives of this work is to analyse the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load.

A program code is established using MATLAB software to perform the calculations required for the exergy plant analysis considering real variation ranges of the main operating parameters such as pressure ratio, air fuel ratio and inlet temperature. The effects of theses parameters on the system performances are investigated.

**Key words:** combined cycle, heat recovery steam generator, exergy, thermodynamic optimization, gas turbine

**Scientific field:**

Technical science, Mechanical engineering, Thermal engineering

**Narrow scientific field:**

Thermal engineering

# CONTENTS

# LIST OF FIGURES



# LIST OF TABLES

# Nomenclature

| | | |
|---|---|---|
| $C_p$ | *Heat capacity at constant pressure* | *kJ/kg·K* |
| $X$ | *Exergy* | *kJ* |
| $X_{Dest.}$ | *Exergy destruction* | *kJ* |
| $X_P$ | *Exergy Production* | *kJ* |
| $h$ | *Specific Enthalpy* | *kJ/kg* |
| $s$ | *Specific Entropy* | *kJ/kg.K* |
| *LCV* | *Low Calorific Value* | *kJ/kg* |
| $\dot{m}_a$ | *Mass Flow Rate of air* | *kg/s* |
| $\dot{m}_f$ | *Mass Flow Rate of fuel* | *kg/s* |
| $P$ | *Pressure* | *bar* |
| $T$ | *Temperature* | *K* |
| $S_{gen}$ | *Entropy Generation* | *kJ/K* |
| $v$ | *Specific Volume* | $m^3/kg$ |
| $V$ | *Velocity* | *m/s* |
| $W_C$ | *Compressor Work rate* | *kW* |
| $W_p$ | *Pump Work rate* | *kW* |
| $W_{ST}$ | *Steam Turbine Work rate* | *kW* |
| $W_{GT}$ | *Gas Turbine Work rate* | *kW* |
| $w$ | *specific work output* | kJ/kg |
| $z$ | *height* | m |
| $r_p$ | *Pressure* Ratio | - |
| $\dot{Q}$ | *Heat Transfer Rate* | kW |
| $q$ | *specific heat supplied or rejected* | kJ/kg |
| $g$ | *gravitational acceleration* | $m/s^2$ |

## Greek Symbols

| | | |
|---|---|---|
| $\gamma$ | *heat capacity ratio* | - |
| $\Psi$ | *specific exergy* | kJ/kg |
| $\eta$ | *efficiency* | - |
| $\rho$ | *density* | $Kg/m^3$ |

## Subscripts

| | |
|---|---|
| *CCGT* | Combined cycle gas turbine |
| *CH* | Chemical |
| *CV* | Control Volume |
| *out* | Outlet |
| *f* | Fuel |
| *i,in* | Inlet |
| *net* | Net |
| *e,out* | Outlet |
| *PH* | Physical |
| *sys* | System |
| *th* | Thermal |
| HRSG | Heat Recovery Steam generator |
| Dest. | Destruction |
| Comb. | Combustion Chamber |
| Cond. | Condenser |
| GT | Gas Turbine |
| ST | Steam Turbine |
| AFR | Air Fuel Ratio |
| PP | Pinch Point |

# CHAPTER 1

## 1.Introduction

Energy systems contain a colossal number and a few sorts of coordinated efforts with the world outside their physical limitations. Thus, architects must address numerous wide issues, particularly energy, economy and the environment. Combined cycle power plants (CCPPs) have as of late gotten significant consideration because of their nearly high energy efficiencies, low poisonous waste and ozone depleting substance releases, and operational suppleness. A typical Combined cycle power plant is the cycle, which is comprised of a gas cycle (topping cycle) and a steam turbine cycle (bottoming cycle) coupled through a Heat recovery steam generator (HRSG).

To streamline the efficiency, cost viability and ecological effect of such plants, it is critical to decide the areas, sorts and genuine extents of wasteful aspects (irreversibility's). Exergy investigation is a helpful tool for such examinations, and allow measurement of the thermodynamic wasteful aspects of the procedure.

Advancement of energy change frameworks turns out to be more vital because of impediments of non-renewable energy sources and the ecological effect amid their utilization. The utilization of energy is discovered wherever in an assortment of uses from warming and cooling to atomic power plants. For a considerable length of time, the reaction to the constantly developing requirement for electric era limit was to manufacture another steam power plant, one not altogether different from the past one.

The energy transformation engineer is confronted with an assortment of issues today: rising advances, changing the social and technological climate in which a differing qualities of methodologies is probably going to be acknowledged.

Some critical qualities of new power activities are low capital and working costs, capacity to work with an assortment of energizes and with high resistance to fuel fluctuation, short development time, low emanation of toxins, attractive or possibly dormant and effortlessly expendable waste items, and high efficiency, maintainability, finance ability, and reliability.

Another main problem faced by the energy conversion engineer is the finiteness of natural resources critically important for human beings (such as natural gas and oil) in the world and ever-increasing energy demands by developing countries. Maybe future power plants ought to use coal and nuclear energy to spare the flammable gas and oil for mechanical sustain stocks and other more basic future needs. Then again, major issues exist concerning use of regular assets. A great part of the promptly accessible coal has unsuitably high sulphur, which fundamentally corrupts the earth when discharged from power plant stacks in untreated ignition items. The notable issue of corrosive rain has been ascribed to discharges from coal-copying power plants.

Thus, just progressive changes in efficiency can be normal. Then again, critical enhancements in efficiency can here and there be gotten by joining regular cycles in fitting ways. Such power plants are alluded to as combined cycle plants. It is obvious from the investigation of the Rankine and Brayton cycles, and truth be told, all heat engines, that the dismissal of a lot of thermal energy to the surroundings goes with the generation of valuable power. This heat rejection can't be disposed of, yet it can be decreased by enhancing the thermal efficiency of the cycle.

Established thermodynamics gives the idea of energy, energy transfer by heat and work, energy balance, entropy and entropy adjust and estimations of thermodynamic properties at balance. The second law of thermodynamics upgrades an energy balance by ascertaining the genuine thermodynamic estimation of an energy balance and genuine thermodynamic inefficiencies and losses from the procedure and framework. Exergy is the greatest valuable work achievable from a energy carrier under the given natural conditions. The exergy of a energy carrier is a thermodynamic property that relies on upon both the condition of the carrier being considered and the condition of nature. It communicates the greatest ability of the energy balance to bring about changes. Along these lines, energy is firmly identified with the financial estimation of the balance since clients pay the capability of energy to bring about changes. At the point when expenses are doled out to energy balance, exergy ought to fill in as a premise in the costing procedure.

Ordinarily, first law analysis gives just energy usage situation regarding preservation of energy. In any case, it can't give the data with respect to the losses both subjectively and quantitatively, and can't discover the area of these losses. These constraints compel us to perform exergy examination in view of second law of thermodynamics. Exergy is not a conserved property but rather some of it is demolished in the genuine process. Exergy investigation gives a uniform base for correlation of different thermodynamic procedures. This analysis demonstrates the data with respect to losses that incorporate their area subjectively and quantitatively. This data can be utilized for further change in the outline and operation of the framework. By finding the exergy destruction, the system performance can be enhanced by enhancing the exergetic proficiency of the component and the system.

## 1.1 The Aim of the Thesis

The scope and purpose of this research is to develop effective methodology to achieve exergetic optimizations of CCGT power plants. Therefore, the aim of the work is to improve the performance of the power plant by means of proposing an exergy optimization method. With the help of this method, it would be possible to:

a) Provide information about the exergy destruction and exergy losses along with their location.

b) Predict the highest exergy destructor components of the system.

c) Suggest ways of improving the exergetic efficiency.

d) Find the optimal realistic values of operating parameters, which gives the maximum possible power output and efficiency. Additionally, is would be possible to calculate minimum possible exergy destructions.

## 1.2 Outline of the Thesis

The thesis is divided into seven chapters. In this chapter, the general concept of exergetic evaluation is introduced. The importance of the optimization and the objective of the study are briefly discussed.

Chapter 2 presents an extensive review of literature covering topics related to this study, about exergy analysis and optimization.

Chapter 3 is an overview of combined cycle gas turbine power plant system. The main emphasis is given to combined cycle thermodynamic and its main components.

Chapter 4 deals with the theoretical background of exergy, exergy analysis, with the expressions.

Chapter 5 represents component wise mathematical modelling of combined cycle gas turbine plant system.

Chapter 6 presents the results of each step of the methodology. In addition, it discusses the effect of the operating parameter on the performance parameters.

Chapter 7 concludes the study results.

# CHAPTER 2

## 2.Literature Survey and Status of Investigation

In order to have an idea of the present methodology development in the area of performance and optimization of combined cycle gas turbine power plant, a brief survey of available literature was made. However, this chapter is concerned with a review of literature on optimization performed on various thermal systems. In general, some authors focus on the gas turbine operating parameters (topping cycle), others optimize the steam plant (bottoming cycle) on the basis of a given gas turbine, whereas others propose appropriate optimization methods for the whole combined cycle power plant. Furthermore, the optimization can be analysed from a thermodynamic point of view, according to the first and/or second law analysis, or using a thermoeconomic or environmental-economic strategy (Kaviri et al [1], Ahmadi and Dincer [2], Boyano et al [3] and Petrakopoulou et al [4]). From the point of view of optimization methodology, there are many types of analyses. In this work, the review will highlight most common methodology: the exergy destruction method, and the exergoeconomic method.

## 2.1 Review of Analysis and Optimization of Topping Cycle

### 2.1.1 Thermodynamic Analysis and Optimization

The gas turbine operating parameters which influence the combined cycle gas turbine performance are; ambient conditions, compressor pressure ratio, and turbine inlet temperature.

#### 2.1.1.1 The Effect of Ambient Conditions

One of the factors that affect gas turbine performance is the ambient conditions, mainly ambient temperature, atmospheric pressure, and the relative humidity of air. These parameters affect the generated electric power and the heat-rate during operation. The location of power plant plays a major role on its performance. The atmospheric air, which enters the compressor, becomes hotter after compression and it is directed to a combustion chamber. Several authors reported the effect of ambient temperature: Ibrahim et al [5], Ameri and Hejazi [6], Boonnasa et al [7] and Hosseini et al [8].

**Ameri and Hejazi [6]** observed that the variation in the ambient temperature causes a loss of 20% of the rated capacity of the 170 gas turbine units in Iran. They studied five gas turbines, where the difference between the ambient temperature and the ISO conditions was on average 11.8 °C. They found that for each 1 °C increase in ambient temperature, the power output was decreased by 0.74%, and they suggested cooling the compressor's intake-air temperature to improve the gas turbine cycle efficiency.

**Hosseini et al** [8] indicated that the gas turbine compressor is designed for constant air volume flow, which makes the electric power output dependent on the ambient temperature through the specific mass flow rate. They added that the increase in the ambient temperature also decreases the compressor's output pressure, which reduces the gas turbine cycle efficiency, while the increase in the air density reduces the gas turbine's heat rate and increases its specific fuel consumption. They stated that for each 1°C increase in the ambient air temperature, the electric power output of the gas turbine decreases by 0.5% to 0.9%, and by 0.27% for a combined cycle.

#### 2.1.1.2 Effect of Compressor Pressure Ratio

The properties of air entering combustion chamber depend upon the compressor pressure ratio studied by: Ibrahim et al [5], Ibrahim and Rahman [9], and Khaliq and Kaushik [10].

**Ibrahim and Rahman [9]** performed a parametric thermodynamic analysis of a combined cycle gas turbine. They investigated the effect of operating parameters, compression ratio, gas-turbine peak temperature ratio, isentropic compressor and efficiency and air fuel ratio on the overall plant performance. Their results show that the compression ratios, air to fuel ratio as well as the isentropic efficiencies are strongly influenced by the overall thermal efficiency of the combined cycle gas turbine power plant. The overall thermal efficiency increases with compression ratio as well as isentropic compressor and turbine efficiency. However, the variation of overall thermal efficiency is minor at the lower compression ratio while it is very significant at the higher compression ratio for both isentropic compressor and turbine efficiency. The overall efficiencies for combined cycle gas turbine are much higher than the efficiencies of gas turbine plants. Efficiency quoted range is about 61%. In addition, the overall thermal efficiency increases and total power output decreases linearly with the increase of the compression ratio with constant turbine inlet temperature. The peak overall efficiency occurs at the higher compression ratio with the higher cycle peak temperature ratio as well as higher isentropic compressor and turbine efficiencies.

#### 2.1.1.3 Effect of Turbine Inlet Temperature

The turbine inlet temperature (TIT) plays an important role on the performance of combined cycle. The maximum value of TIT is fixed due to the metallurgical problem of turbine blade cooling. Research in this area was done by:

**Sanjay** [11] stated that the parameter that affects cycle performance most is the turbine inlet temperature TIT. The TIT should be kept on the higher side, because at lower values, the exergy destruction is higher.

## 2.2 Review of Analysis and Optimization of Bottoming Cycle

### 2.2.1 Thermodynamic Analysis and Optimization

The efficiency of steam power plants can be improved by increasing the live steam and reheat-steam parameters, and by introducing high-efficiency, low-loss turbine blade geometries. The first goal, to increase the steam parameters, is primarily achieved by choosing appropriate materials for the components operating under live-steam and reheat-steam conditions while retaining the proven designs. Collaborative European programs have led to the development and qualification of steels with much improved creep properties at temperatures of up to 600 °C, appropriate for the manufacture of key components. At the same time, optimization of the blade profiles and geometries allowed further major improvements in operating efficiency. The achievable improvements in efficiency is about 0.5% per 10 °C live steam and reheat (RH) temperature increase, and 0.2 % per 10 bar pressure increase. Second important part of the bottoming cycle is the heat recovery steam turbine (HRSG), its design and optimization affects to a large extent influence the efficiency and the cost of the whole plant.

**Mohagheghi and Shayegan [12]** performed the thermodynamic optimization of design variables and heat exchangers layout in a heat recovery steam generator HRSG for combined cycle gas turbine CCGT using a genetic algorithm. Their method was introduced for modelling the steam cycle in advanced combined cycles by organizing the non-linear equations and their simultaneous used solutions with numerical methods.

In addition to the optimization of design variables of the recovery boiler, they performed the distribution of heat exchangers among different sections and optimized their layouts in HRSGs. A standard gas turbine was assumed, and then outlet gas stream conditions (mass flow rate, temperature, and chemical composition of gas stream) were considered as the inlet parameters for the recovery boiler model. From the optimization process maximum output power from a steam cycle for different HRSGs was then analysed.

**Bracco and Silvia [13]** studied a combined cycle power plant with a single level heat recovery steam generator HRSG. They developed a mathematical model to determine the optimal steam pressure values in the HRSG according to different objective functions (in the HRSG for a given gas turbine). Their work reports numerical results for the combined cycle power plant considering four different gas turbines. The optimization approach was focused on the study of the heat transfer between the steam and the exhaust gas in the HRSG, based on an exergetic analysis. They present the comparison among different objective functions that refer to the HRSG specifically or to the whole bottoming cycle. In their mathematical model, they considered the presence of specific constraints for the operating parameters of the power plant, the most important constraints that were considered refer to the steam quality at the turbine outlet, the HRSG outlet exhaust gas temperature and the steam turbine blade height. In their work, a parametric analysis was also performed to evaluate the influence of the gas temperature at the HRSG inlet and the pinch point temperature difference on the considered objective functions.

**Woudstra et al [14]** performed the thermodynamic evaluation of combined cycle plants with the same gas turbine and different steam bottoming cycles. The evaluation showed that the increasing the number of pressure levels of steam generation will reduce the losses due to heat transfer in the HRSG, but also the exergy loss due to the exhaust of flue gas to the stack. Among the investigated configurations for bottoming cycle, triple pressure reheat was the best option from exergy point of view.

**Mansouri et al [15]** investigated the effect of pressure levels of steam generation at heat recovery steam generator HRSG on the energetic and exergetic efficiency of HRSG, bottoming cycle and combined cycle power plants, as well as the effect of

HRSG (heat recovery steam generator) pressure levels on exergy destruction at HRSG and other main components of the bottoming cycle. Their result show that an increase in pressure levels of steam generation at HRSG leads to an increase in the exergy efficiency of HRSG and CCPP increase respectively. In addition, an increase in pressure levels at HRSG decreases the exergy destruction due to heat transfer in HRSG: the exergetic efficiency of HRSG increases with an increase in pressure levels of steam generation and adding reheat to the cycle.

**Xiang and Chen [16]** considered a combined cycle with three-pressure HRSG, equipped with the GE PG9351FA gas turbine. They maximized the combined cycle efficiency through the optimization of the HRSG operating parameters by minimizing exergy losses. Moreover, they highlighted the influence of the HRSG inlet gas temperature on the bottoming cycle efficiency. They studied the influence of HRSG inlet gas temperature on the steam bottoming cycle efficiency. Their result shows that increasing the HRSG inlet temperature has less improvement to steam cycle efficiency when it is over 590ºC.

## 2.3 Exergy Analysis

In order to have an idea of the present methodology development in the area of performance and optimization of combined cycle gas turbine power plant, a brief survey of available literature was made. However, this chapter is concerned with a review of literature on optimization performed on various thermal systems. In general, some authors focus on the gas turbine operating parameters (topping cycle), others optimize the steam plant (bottoming cycle) on the basis of a given gas turbine, whereas others propose appropriate optimization methods for the whole combined cycle power plant.

Furthermore, the optimization can be analysed from a thermodynamic point of view, according to the first and/or second law analysis, or using a thermoeconomic or environmental-economic strategy (Kaviri et al [1], Ahmadi and Dincer [2], Boyano et al [3] and Petrakopoulou et al [4]). From the point of view of optimization methodology, there are many types of analyses. In this work, the review will highlight most common methodology: the exergy destruction method, and the exergoeconomic method.

**Sayed A. Abdel-Moneim et al. [17]** shows the energy and exergy analysis of a combined cycle with a supercharged boiler was carried out. A combination of a basic gas turbine and steam cycle with both a supercharged boiler (SB) and a heat recovery boiler (HRB) was investigated. The effects of the inlet temperature of the gas turbine, the excess air factor, and the compressor pressure ratio on the performance of the supercharged boiler combined cycle (SBCC) were studied. Comparisons between the SBCC and the conventional combined cycle were performed. The results indicated that the SBCC gives output power up to 2.1 times of that of the conventional combined cycle when compared at the same values of the operating parameters. However, the SBCC efficiency was found to be lower than the conventional combined cycle. The exergy analysis showed an advantage of SBCC over the conventional combined cycle.

**Fiaschi and Giampaolo [18]** investigated an exergy analysis of the semi-closed gas turbine combined cycle. They concluded that combustion, heat recovery steam generator, water injection/mixing, and water recovery system are the main sources of the losses, representing globally more than 80% of the overall exergy destruction.

**Cihan et al [19]** carried out energy and exergy analyses for a combined cycle located in Turkey, and suggested modifications to decrease the exergy destruction in CCPPs. Their results showed that combustion chambers, gas turbines, and HRSGs are the main sources of irreversibilities, representing over 85% of the overall exergy losses.

**Mousafarash and Ameri [20]** their study consists of exergy analysis of a typical GT power plant, analysis of system performance at different ambient temperatures and partial loads, and exergo-economic analysis of the gas turbine power plant. The results of their study reveal that the highest exergy destruction occurs in the combustion chamber, where the large temperature difference is the major source of the irreversibility. In addition, the effects of the gas turbine load variations and ambient temperature were investigated to see how system performance changes: the gas turbine was significantly affected by the ambient temperature, which led to a decrease in net power output. The results of the load variation of the gas turbine showed that a reduction in gas turbine load resulted in a decrease in the exergy efficiency of the cycle as well as all the components. They conducted an exergo-economic analysis to determine the cost of exergy destruction in each component and to determine the cost of fuel. The results show that combustion chamber has the largest cost of exergy destruction.

**Butcher and Reddy [21]** carried out exergy analysis for waste heat recovery based power generation system. The performance of the waste heat recovery power generation systems based on second law analysis was investigated for various operating conditions. The temperature profiles across the heat recovery steam generator (HRSG), network output, second law efficiency, and entropy generation number were simulated for various operating conditions. The variation in specific heat with exhaust gas composition and temperature were accounted in the analysis and results. The effect of pinch point on the performance of HRSG, entropy generation rate and second law efficiency were also investigated. The researchers found that the second law efficiency Chapter 2 Literature Survey and Status of Investigation 18 of the HRSG and power generation system decreases with increasing pinch point. The first and second law efficiency of the power generation system varies with exhaust gas composition and with oxygen content in the gas. The results contribute further information about the role of gas composition, specific heat and pinch point influence on the performance of a waste heat recovery based power generation system (based on first and second law of thermodynamics).

**Rashad et al. [22]** In this study, the energy and exergy analysis of Shobra El-Khima power plant in Cairo, Egypt is presented. The primary objectives of this paper are to analyse the system components separately and to identify and quantify the sites having largest energy and exergy losses at different load. The performance of the plant was estimated by a component-wise modelling and a detailed break-up of energy and exergy losses for the considered plant has been presented at different loads (Maximum load, 75% load and, 50 % load). Energy losses mainly occurred in the condenser where (404.653 MW at Max load, 306.747 MW at 75% load and 278.849 MW at 50% load) is lost to the environment. The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the Turbine system (42% at Max load, 59% at 75% load and 46.1 at 50% load) followed by the condenser (28% at Max load, 20.3% at 75% load) while at 50% load the feed water heaters showed more exergy destruction (27.7%) than condenser (23.8) and then the feed water heaters (20.8% at max load, 12.1% at 75% load). In addition, the calculated thermal efficiency based on the specific heat input to the steam was 43% while the exergy efficiency of the power cycle was (44% - 48%).

## 2.4 Exergy Destruction Method

**Kamate and Gangavati [23]** analysed cogeneration power plants in sugar industries through exergy destruction method for various steam inlet condition. The result shows that, at optimal steam inlet conditions of 61 bar and 475 C, the backpressure steam turbine cogeneration plant perform with energy and exergy efficiency of 0.863 and 0.307, while the condensing steam turbine plant perform with energy and exergy efficiency of 0.682 and 0.26, respectively. Boiler is the least efficient component and turbine is the most efficient component of the plant.

**Aljundi [24]** studied energy and exergy analysis of a steam power plant in Jordan using exergy destruction method. A component wise modelling and a detailed break-up of energy and exergy losses estimated the performance of the plant. The modelling shows that the thermal efficiency (26%) is low compared to modern power plants, because this efficiency was not based on the specific heat input to the steam; rather, it was based on the lower heating value of the fuel to incorporate the losses occurring in the furnace-boiler system due to energy lost with hot gases, incomplete combustion, etc. It was also observed that the maximum exergy destruction is in boiler and maximum exergy loss in condenser.

**Abusoglu and Kanoglu [25]** applied the exergy destruction method to the diesel engine powered cogeneration systems generating electricity and steam. They defined the fuel and product in terms of exergy flow for each component of the system and then calculated the exergetic efficiency of them. It was observed that the total exergy destruction in the engine was mostly due to the highly irreversible combustion process in the engine, heat losses from engine, and friction.

**Tyagi and Khan [26]** studied the effects of gas turbine exhaust temperature, stack temperature and ambient temperature on the overall efficiency of combine cycle power plant keeping the gas turbine efficiency as well as steam turbine efficiency constant. They concluded that the stack temperature should be minimum and gas turbine exhaust temperature should be maximum. Out of these three variables i.e. turbine exhaust temperature, stack temperature and ambient temperature, the dominating factor of increasing the overall efficiency of the combine cycle power plant is the stack temperature.

**Valdés et al [27]** showed a possible way to achieve a thermoeconomic optimization of combined cycle gas turbine power plants. The optimization was done by using a genetic algorithm, tuned by applying it to a single pressure CCGT power plant. Once tuned, the optimization algorithm was used to evaluate more complex plants, with two and three pressure levels in the heat recovery steam generator. The variables considered for the optimization were the thermodynamic parameters that established the configuration of the HRSG. Two different objective functions were proposed: one minimizes the cost of production per unit of output and the other maximizes the annual cash flow. The results obtained with both functions were compared in order to find the better optimization strategy. The results show that it is possible to find an optimum for each design parameter. This optimum depends on the selected optimization strategy.

**Bassily [28]** presented the effects of varying the inlet temperature of the gas turbine and PP on the performance of a dual pressure reheat combined cycle. He also modelled some feasible techniques to reduce the irreversibility of the HRSG of both cycles, and showed that optimizing or reducing the irreversibility of these cycles could increase their efficiencies by 2–3%. Applying gas reheat increases the generated power and average temperature at which heat is supplied, whereas applying gas recuperation takes advantage of the increased gas temperature at the outlet of the GT to enhance cycle efficiency. For gas-reheat gas-recuperated combined cycles, recuperated heat exchangers fabricated from stainless steel have to be used to withstand these conditions. He compared the optimized results with the regularly designed triple pressure reheat combined cycle Bassily. Boonnasa et al studied the performance improvement of an existing combined cycle power plant located in Bangkok that consisted of two gas turbines (110.76MW each), and one 115.14MW steam turbine in ISO conditions. The plant used an absorption chiller to cool one of the two gas turbine's intake-air to 15°C, in addition to having a thermal energy storage tank that stored the sensible heat of the chilled water to meet the varying daily cooling load. Low-pressure steam from a heat recovery steam generator was used to drive the absorption chiller needed to meet a maximum load of 7049.58kW with the help of the thermal heat storage. As a result, the power output of the cooled gas turbine increased by 10%, improving the CCPP total power output by 6.24%. Economically, the study found that due to the low initial investment cost of retrofitting the absorption chiller the internal rate of return was 40%, and the payback period was just 3.81 years. However, the authors also reported a reduction by 2.85% in the steam turbine power output, which was due to powering the absorption chiller directly from the HRSG unit steam that was powering the steam turbine. This reduction in the steam turbine power output could have been avoided if they had used a boiler that utilized the waste heat energy from the stack after the HRSG unit.

The efficiency of steam power plants can be improved by increasing the live steam and reheat-steam parameters, and by introducing high-efficiency, low-loss turbine blade geometries. The first goal, to increase the steam parameters, is primarily achieved by choosing appropriate materials for the components operating under live-steam and reheat-steam conditions while retaining the proven designs. Collaborative European programs have led to the development and qualification of steels with much improved creep properties at temperatures of up to 600 °C, appropriate for the manufacture of key components. At the same time, optimization of the blade profiles and geometries allowed further major improvements in operating efficiency. The achievable improvements in efficiency is about 0.5% per 10 °C live steam and reheat (RH) temperature increase, and 0.2 % per 10 bar pressure increase. Second important part of the bottoming cycle is the heat recovery steam turbine (HRSG), its design and optimization affects to a large extent influence the efficiency and the cost of the whole plant.

**Mohagheghi and Shayegan [29]** performed the thermodynamic optimization of design variables and heat exchangers layout in a heat recovery steam generator HRSG for combined cycle gas turbine CCGT using a genetic algorithm. Their method was introduced for modelling the steam cycle in advanced combined cycles by organizing the non-linear equations and their simultaneous used solutions with numerical methods. In addition to the optimization of design variables of the recovery boiler, they performed the distribution of heat exchangers among different sections and optimized their layouts in HRSGs. A standard gas turbine was assumed, and then outlet gas stream conditions (mass flow rate, temperature, and chemical composition of gas stream) were considered as the inlet parameters for the recovery boiler model. From the optimization process maximum output power from a steam cycle for different HRSGs was then analysed. Bracco and Silvia studied a combined cycle power plant with a single level heat recovery steam generator HRSG. They developed a mathematical model to determine the optimal steam pressure values in the HRSG according to different objective functions (in the HRSG for a given gas turbine). Their work reports numerical results for the combined cycle power plant considering four different gas turbines. The optimization approach was focused on the study of the heat transfer between the steam and the exhaust gas in the HRSG, based on an exergetic analysis. They present the comparison among different objective functions that refer to the HRSG specifically or to the whole bottoming cycle. In their mathematical model, they considered the presence of specific constraints for the operating parameters of the power plant, the most important constraints that were considered refer to the steam quality at the turbine outlet, the HRSG outlet exhaust gas temperature and the steam turbine blade height. In their work, a parametric analysis was also performed to evaluate the influence of the gas temperature at the HRSG inlet and the pinch point temperature difference on the considered objective functions.

**Woudstra et al [14]** performed the thermodynamic evaluation of combined cycle plants with the same gas turbine and different steam bottoming cycles. The evaluation showed that the increasing the number of pressure levels of steam generation will reduce the losses due to heat transfer in the HRSG, but also the exergy loss due to the exhaust of flue gas to the stack. Among the investigated configurations for bottoming cycle, triple pressure reheat was the best option from exergy point of view. Mansouri et al investigated the effect of pressure levels of steam generation at heat recovery steam generator HRSG on the energetic and exergetic efficiency of HRSG, bottoming cycle and combined cycle power plants, as well as the effect of HRSG (heat recovery steam generator) pressure levels on exergy destruction at HRSG and other main components of the bottoming cycle. Their result show that an increase in pressure levels of steam generation at HRSG leads to an increase in the exergy efficiency of HRSG and CCPP increase respectively. In addition, an increase in pressure levels at HRSG decreases the exergy destruction due to heat transfer in HRSG: the exergetic efficiency of HRSG increases with an increase in pressure levels of steam generation and adding reheat to the cycle.

**Arora and Rai [30**] shows The plant consists of a compressor, combustor, gas turbine, waste heat recovery boiler, steam turbine, and generator(s). The input temperature to a steam turbine is about 540°C and the exhaust can be maintained at the atmospheric pressure, due to design consideration the input temperature is limited and the efficiency of the about 40%. The input temperature of the gas turbine can be as high as 1100°C but the exhaust temperature can be lowered to about 500-600°C, the efficiency of a gas turbine is about 33%. It can be seen that to obtain higher efficiencies the exhaust of the gas turbine can used to drive the steam turbine giving efficiency up to 60%.

# CHAPTER 3

## 3 Description of CCPP

### 3.1 Introduction

The efficiency of combined cycle was around 5–6% higher than that of a comparative traditional steam plant. These systems could monetarily use uncovered tubes in the boiler in light of the high mean temperature contrast between the combustion products and the water/steam. The second era, which combined cycle system with finned tube boilers, entered service in1959. Amid the 1960s, the utilization of the heat recovery kind of combined cycle systems turned out to be more common. Its underlying application was in power and heat applications where its power to heat proportion was more positive. In addition, few of the heat recovery type combined cycles were introduced in utility power generation applications amid the 1960s. The use of these systems in the 1980s set up the heat recovery sustain water heating combined cycle as a developed technology for base load and mid-go benefit. [31].

By 1970, there were various plants in operation. All through the 1980s, the innovation created with bigger gas turbines and the presentation of pre-mixed combustion for low $NO_2$ discharges. Around 1990 the net plant productivity of joined cycles passed 50% (LHV). All through the 1990s, countless cycle power plants were assembled, and a considerable lot of them in the base load operation. Around 1995 another era of expensive gas turbines went to the market, bringing the piece size of combined cycle power plant yield to 350-400MW and proficiency up to 57-58% [32]. As of 2011, the power output was increased to about 570MW and efficiency close to 61% [33].

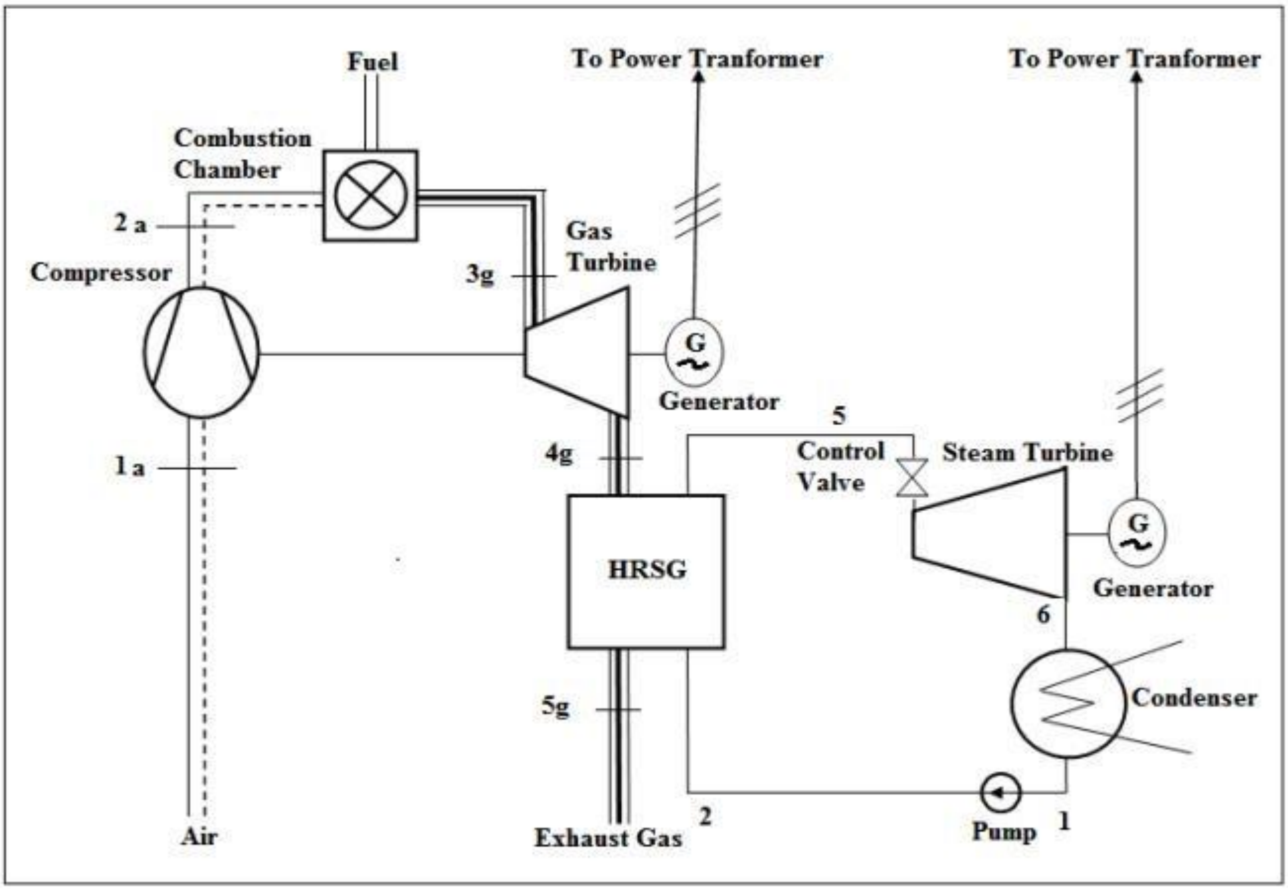


*Fig 3.1 Schematic combined cycle gas turbine topping cycle and bottoming cycle*

The HRSG catches around 66% of the gas turbine exhaust heat with the staying third being lost in the leave stack. At long last, 19% of the fuel info is changed over into power by means of the steam turbine with 1% lost in the turbine and 21% of the fuel energy lost in the spent steam, which is sent to the condenser. The combined gross power of gas and steam turbines compares to 55% (LHV) of the fuel energy. Plant assistant records for ~2% of the fuel input at long last leaving 53% as net output combined cycle efficiency. In this manner, the fundamental defence for using HRSGs inside utility power plants lays free advantage from the superposition of the gas turbine Brayton cycle over the steam turbine Rankin cycle Figure 3-1 which brings about an improved overall thermal efficiency. [34].

### 3.1.1 Compressor

The compressors in many gas turbine applications, particularly units more than 5MW, utilize axial flow compressors. An axial flow compressor is one in which the stream enters the compressor in a hub bearing (parallel with the axis of rotation) and exits from the gas turbine, also in an axial direction. The axial flow compressor packs its working liquid by first quickening the liquid and after that diffusing it to acquire a pressure increment. The liquid is accelerated by a line of rotating air foils (cutting edges) called the rotor, and afterward diffused consecutively of stationary blades (the stator). The diffusion in the stator changes over the velocity increment picked up in the rotor to a pressure increase. A compressor comprises of a few stages: 1) A combination of a rotor taken after by a stator make-up a phase in a compressor; 2) An additional row of stationary blades are regularly utilized at the compressor inlet and are known as Inlet Guide Vanes (IGV) to guarantee that air enters the first stage rotors at the desired flow angle, these vanes are additionally pitched variable in this manner can be changed in accordance with the shifting stream prerequisites of the engine; and 3) notwithstanding the stators, another diffuser at the exit of the compressor comprising of another arrangement of vanes further diffuses the liquid and controls its velocity entering the combustors and is frequently known as the Exit Guide Vanes (EGV).

In an axial flow compressor, air goes starting with one stage then onto the next, each stage raising the pressure somewhat. By delivering low-pressure increments on the request of 1.1:1 to 1.4:1, high efficiencies can be acquired. The use of different stages licenses overall pressure increases of up to 40:1 in some aviation applications and a pressure ratio of 30:1 in some Industrial applications.

The most recent twenty years has seen an extensive development in gas turbine innovation. The development is led by the expansion in compressor pressure proportion, advanced combustion techniques, the development of materials innovation, new coatings, and new cooling plans. The expansion in gas turbine productivity is reliant on two essential parameters:

1. Increment in Pressure Ratio
2. Increment in Firing Temperature

It additionally ought to be recollected that the Gas Turbine Axial Flow Compressor devours between 55%-65% of the power created by the Turbine area of the gas turbine.

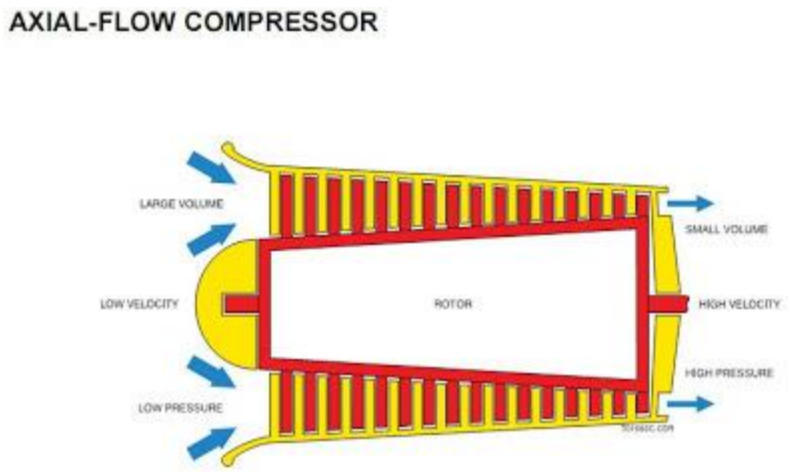


*Figure 3.2: Axial flow compressor*

### 3.1.2 Combustor

The combustion chamber is that turbo-engine part which fuel, provided by bolstering nozzles, is blended with air flow originating from the compressor and consumes discharging heat to get a gas stream to a temperature however much as could reasonably be expected uniform in condition requested for from the turbine.

This task must be completed with the conceivable least of pressure leaks and with the most extreme conceivable release of heat regarding the restricted space accessible. The measure of fuel provided to the air depends on asked for temperature raising.
Be that as it may, the most extreme temperature is restricted in 850 - 1700 C interim, depending on qualities of materials of which turbine vanes and blades are constituted and by cooling arrangement of same blades and vanes. The air enters in burning chamber with temperature involved in the vicinity of 200 and 550 °C in view of the heating brought on by pressure, so the required temperature bringing up in the ignition chamber is 650-1150 °C.

The ignition chamber should likewise have the capacity to keep up a steady and efficient burning in a wide sweep of engine working conditions.

An efficient burning has turned out to be increasingly imperative amid a most recent couple of years due to the quick addition of the business air traffic and subsequent increment of climatic contamination in exhaust smokes shape.

The air originating from the compressor enters the combustion chamber with speed that can achieve 150 m/s. This speed is excessively high for ignition, so first that must occur in a combustor is a spread, that is a deceleration of air with a resulting increment of the static pressure.

The fuel consuming velocity to the ordinary blending rate air-fuel is of little meters/second, any fire acquired in these conditions would be overwhelmed.

Accordingly, a low-speed axial region must be made in the chamber so that the fire stays touched off and tied down in all states of motor operation.
In ordinary conditions, the aggregate air-fuel rate in an ignition chamber can fluctuate in the vicinity of 60:1 and 130:1. The fuel just consumes in an effective approach to a rate close to 15:1, in this way the fuel must be scorched just with a piece of entering the air in that which essential zone is called. This is gotten by a fire tube that has a few gets to that direct the air dispersion along combustor.
Roughly 20% of the air limit enters in ignition chamber through the nose.
Instantly downstream to the nose, there are swirl vanes and a punctured plate through which the air goes in the essential zone of the combustor. The air that originates from the twirls initiates a movement of distribution in the essential zone.

### 3.1.3 Gas Turbine

A gas turbine is a machine conveying mechanical power or thrust. It does this utilizing a gaseous working fluid. The mechanical power created can be utilized by, for instance, a modern gadget. The active vaporous liquid can be utilized to produce push or to create power. In the gas turbine, there is a constant stream of the working liquid. This working liquid is at first packed in the compressor. It is then warmed in the combustion chamber. The turbine changes over the energy of the gas into mechanical work.

Some portion of this work is utilized to drive the compressor. The rest of, "useful power", is utilized as the yield shaft power to turn an energy conversion device, for example, an electrical generator [35]. In the power era field, the gas turbine can be utilized as a remain solitary unit or with combined cycle power plants. Power creating gas turbines are generally open cycle worked. The gas turbine execution relies on upon the execution of its segments i.e. compressor, combustion chamber, and turbine.

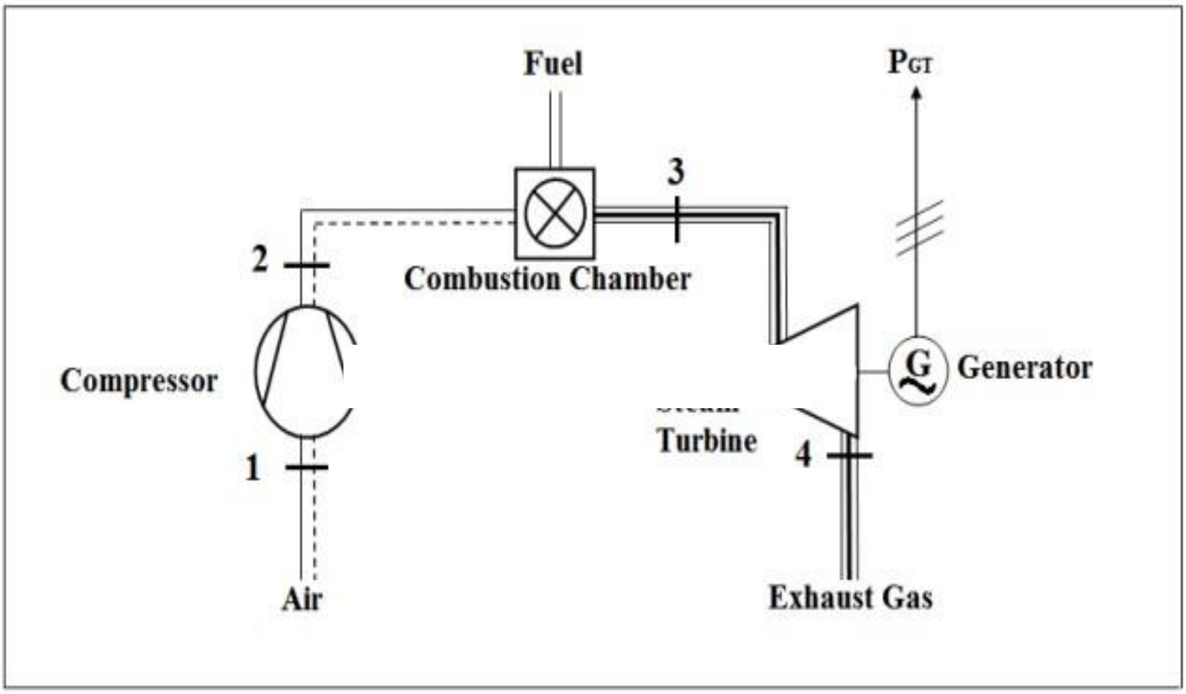


*Fig 3.3. Schematic diagram of Brayton cycle*

The thermodynamic cycle of a gas turbine is known as the Brayton cycle. Gas turbines usually operate on an open cycle. The air is first compressed in the compressor, the air drawn at ambient conditions into the compressor intake, where the compressor pressurizes the air up to $P_2$, increasing both pressure and temperature at the expenses of using compression work ($W_{comp}$ ) which is supplied by the turbine itself. The high-pressure air proceeds into the combustion chamber, where the fuel is burned at a constant pressure. This added heat ($Q_H$) raises the temperature from $T_2$ to the turbine inlet temperature (highest cycle temperature) $T_3$. The resulting high temperature gases then enter the turbine, where they expand to the atmospheric pressure while producing power ($W_{Turbine}$) enough to drive the compressor and produce net shaft work (W).

Finally, heat $Q_L$ is rejected to the environment. Most of the gas turbines in electricity generation use axial flow compressors [36].

The exhaust gases leaving the turbine are thrown out (not re-circulates), causing the cycle to be classified as an open cycle. The Compression ratio and turbine inlet temperature are important parameters.

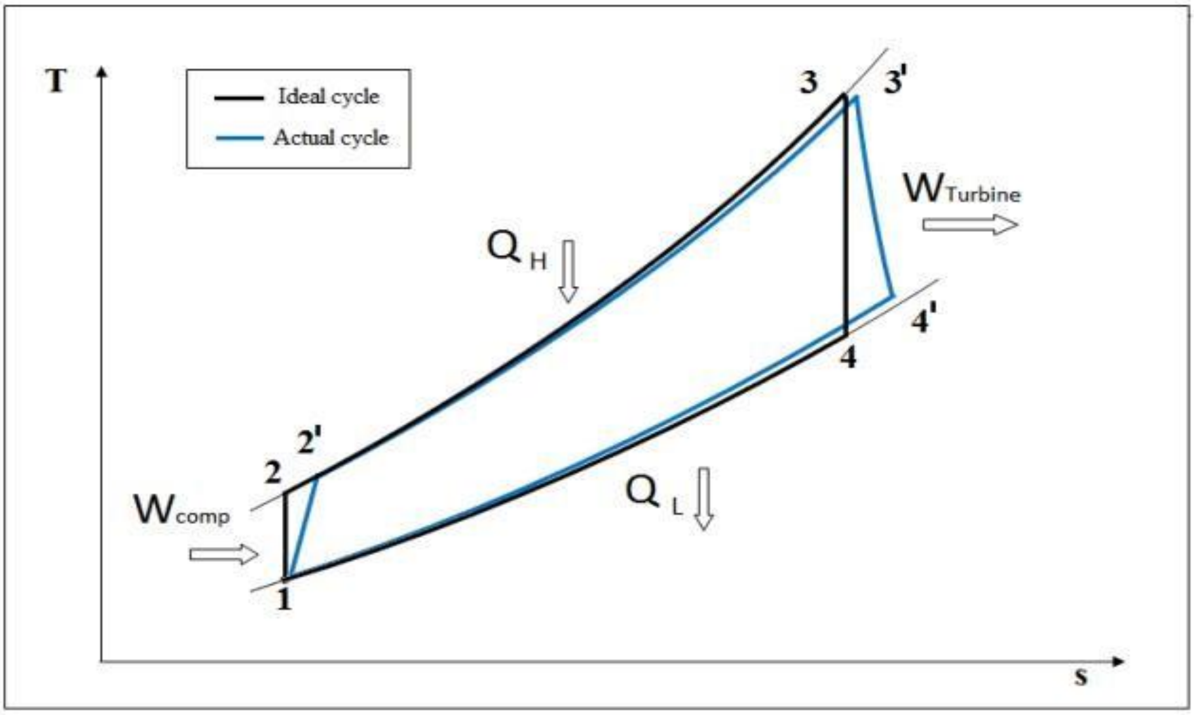


Figure 3.4: Brayton cycle

### 3.1.4 Steam Turbine

Steam turbine is a great prime mover to change over heat energy of steam to mechanical energy. It is one of the notable prime movers, for example, gasoline engines, diesel engines, gas turbines, jet engines, and so forth. All steam engines, regardless of whether turbines or not, are intended to concentrate energy from high-pressure steam and change it into movement by enabling the steam to extend. For the turbine designs, steam permitted to extend slowly through more than one arrangement of cutting edges, for accomplishing substantially higher efficiencies contrasted with a solitary stride extension [37]. The steam grows through progressive rings of moving blades on a pole and fixed blades in the casing, creating purely turning movement. At the point when combined with an electric generator, steam turbine is a standout amongst the most critical methods for creating mass electric power on the planet. The advanced steam turbine may have three stages. The high-pressure segment has small blades. They are little on the grounds that the approaching steam has high energy at high temperature. After the steam goes through the high-pressure area, it is sent back to the heater to be reheated. The steam is then sent to the following area of the turbine, called the middle-pressure section. The blades here are bigger than those in the high-pressure section. In the wake of going through this segment, the steam is sent to the low-pressure area of the turbine. Since the greater part of the energy was beforehand expelled from the steam, the blades here are the biggest in the turbine. The steam leaves the turbine through the base, where it is consolidated once again into the water. From that point it is sent back to the boiler, to be made into steam once more. The steam turbine is regularly utilized as a part of a combined heat and power generation handle where the turbine drives a machine in the meantime: steam separated from the machine is utilized to supply locale heating as well as process steam systems [36].

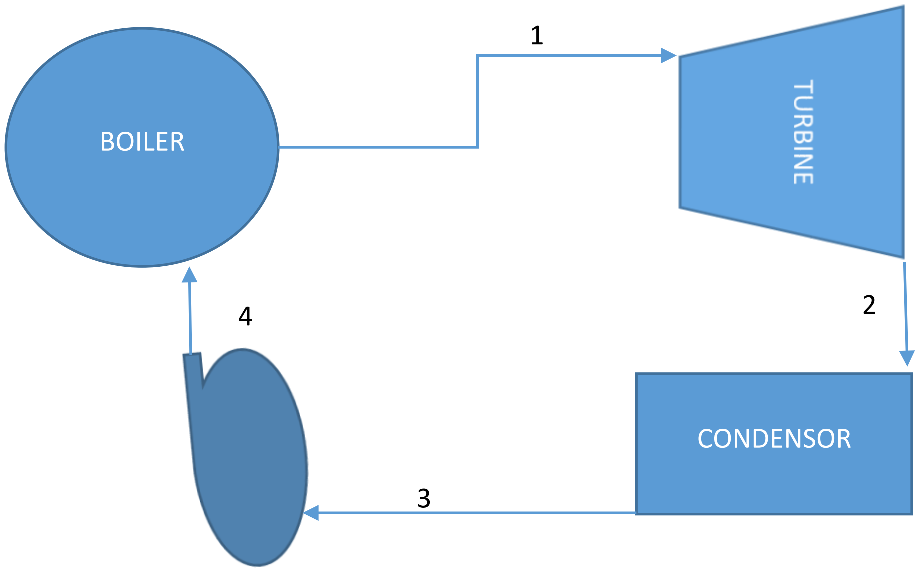


*Fig 3.5 Steam power plant cycle*

A single unit of steam turbine can create power extending from 1 MW to 1000 MW. The thermal efficiencies of the present day steam power plant over 120 MW are as high as 38% to 40% [38]. Water (steam) is the working liquid for most vapour power cycles. Water works over a wide scope of temperatures and pressures have an expansive heat limit, and it is steady, safe, and ecologically cordial. The energy sources used to produce steam incorporate gas, coal, oil, and nuclear sources.

Steam Turbine Capacity: the capacities of small turbines and coupled generators fluctuate from 500 to 7500 kW, while substantial turbo alternators have limit differing from 10 to 90 MW. Large size units have limits up to 500 MW. Producing units of 200 MW limit are ending up plainly very normal. The steam utilization by steam turbines relies on upon steam pressure and temperature at the inlet, deplete pressure number of draining stages and so on. The steam utilization of huge steam turbines is around 3.5 to 5 kg for every kWh [39].

**Steam Turbine Performance:** Turbine execution is communicated by the the most usually utilized vapour power cycle is the Rankine cycle. Despite the fact that a portrayal of the Rankin cycle can be found in any building thermodynamics course book, it is quickly secured here. The working liquid is pumped to a high pressure and circled through the heater. The fluid is bubbled at a consistent pressure in the boiler after which the high-pressure vapour created is extended through a turbine, in this manner extracting work from it. The vapour leaving the turbine is condensed in a condenser by dismissing heat to a cooling fluid. A few alterations to the Rankine cycle

accompanying components: a) The steam stream handle through the unit expansion line or condition curve

b) The steam stream rate through the unit

c) Thermal efficiencies

d) The losses, for example, deplete, mechanical, generator and radiation

Mechanical losses incorporate bearings losses, oil pump losses, and generator bearing losses. Generator losses incorporate electrical and mechanical losses. Debilitate losses incorporate the active energy of the steam as it leaves the last stage and the pressure drop from the exit of the last stage to the condenser organize. For effective operation of a steam turbine, it is alluring to supply steam at consistent pressure and temperature. Steam pressure can be effortlessly managed by methods for security valve fitted on the boiler.

are utilized to accomplish better efficiencies. These incorporate superheating, heating, and recovery. A hefty portion of the difficulties related to the Carnot cycle can be wiped out by superheating the steam in the boiler and gathering it totally in the condenser.

The cycle that outcomes is the Rankine cycle, which is the perfect cycle for vapor power plants. The perfect Rankine cycle does not include any internal irreversibility and comprises of the accompanying four procedures: isentropic compression in a pump, consistent pressure warm expansion in a boiler, isentropic expansion in a turbine, and steady pressure warm dismissal in a condenser. Every one of the four sections of the Rankine cycle is unfaltering state enduring stream devices. The potential and dynamic energy impacts can be ignored. The boiler and the condenser don't include any work, and the pump and the turbine are thought to be isentropic [37].

### 3.1.5 Condenser

The capacity of the condenser is to condense exhaust steam from the steam turbine by dismissing the heat of vaporization to the cooling water going through the condenser.
Once the steam has gone through the turbine, it enters the condenser where heat is evacuated until it consolidates over into liquid water. This is finished by passing the wet steam around a large number of little frosty water tubes.
The cold water is normally provided from a nearby ocean, lake, waterway, or from a cooling tower. The condensed steam is collected at the base of the condenser and come back to the heater utilizing bolster water pumps, to start the water-to-steam, steam-to-water cycle once more.
The temperature of the condensate decides the pressure in the steam/condensate side of the condenser. This pressure is known as the turbine backpressure and is typically a vacuum. Diminishing the condensate temperature will bring about a bringing down of the turbine backpressure and expanding turbine power output and efficiency.
The condenser also has the following secondary functions:

- The condensate is gathered in the condenser hot well, from which the condensate pumps take their suction;
- Provide here and now stockpiling of condensate;
- Provide a low-pressure gathering point for condensate channels from different systems in the plant; and
- Provide for de-air circulation of the gathered condensate.

Large power plant condensers are usually 'shell and tube' heat exchangers.
These types of condensers are also classified:

- As single pressure or multi-pressure, contingent upon whether the cooling water stream way makes at least one turbine backpressures;
- By the number of shells (which is reliant on quantity of low-pressure turbine housings); and
- As either single pass or two-pass, contingent upon the quantity of parallel water stream ways through each shell.

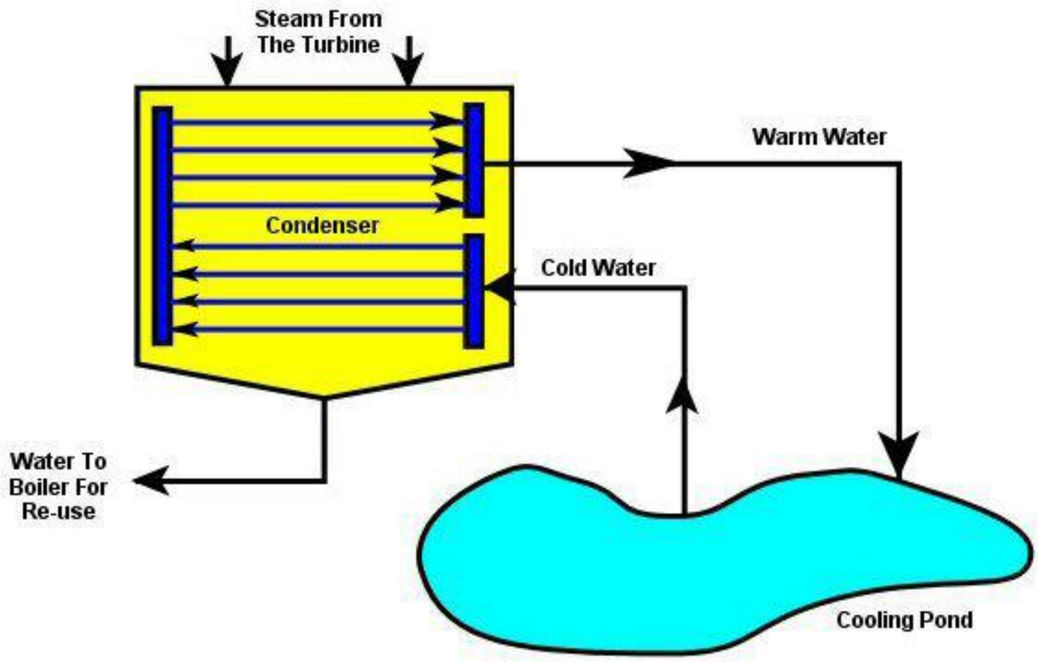


*Figure 3.6: Condenser used in steam power cycle*

### 3.1.6 Boiler feed Water Pump

Boiler Feed Pump as the name proposes, discovers application in heater or reactor vessels to feed water and furthermore control the measure of water fed to the boiler. It has got significant role in the operation of boilers.

Most common pump is a multistage centrifugal in power plants for feeding water to boilers. The primary parts of this pump are the impeller, stainless steel 2-piece twin volute, and stage pieces, mix of balancing drum and circle, shaft and gland packing. A substantial round and hollow fashioned steel barrel is a necessary part for the stages to be stacked and shut down toward one side by a dash on the head. The stages inside barrel casing are part radially. In few pumps, the stages are likewise part axially.

### 3.1.7 Heat Recovery Steam Generator

In the present innovation, the most efficient energy conversion systems to deliver electrical and thermal energy are the combined cycle power plants. In a run of the mill

CCPP, exhaust heat from the gas turbine GT is recuperated in a heat recovery steam generator to create steam in the steam cycle.  HRSG execution larger affects the general execution of a combined cycle power plant. Steam generated in HRSG with different pressure levels depend on the design. HRSG consist of three heat exchanger packages (economizer, evaporator, and superheated).

Combustion gases enter superheater, evaporator, and economizer package respectively. The heat recovery from gas side to the water-steam is achieved in three steps:

In the economizer, the feed water is heated to temperature close to its saturation temperature. In the evaporator, the water vanishes at a consistent temperature and pressure and ends up plainly immersed steam. In the superheater, the high esteem heat from the exhaust is utilized for superheating the steam created in the evaporator. Superheated steam is fed to the steam turbine.

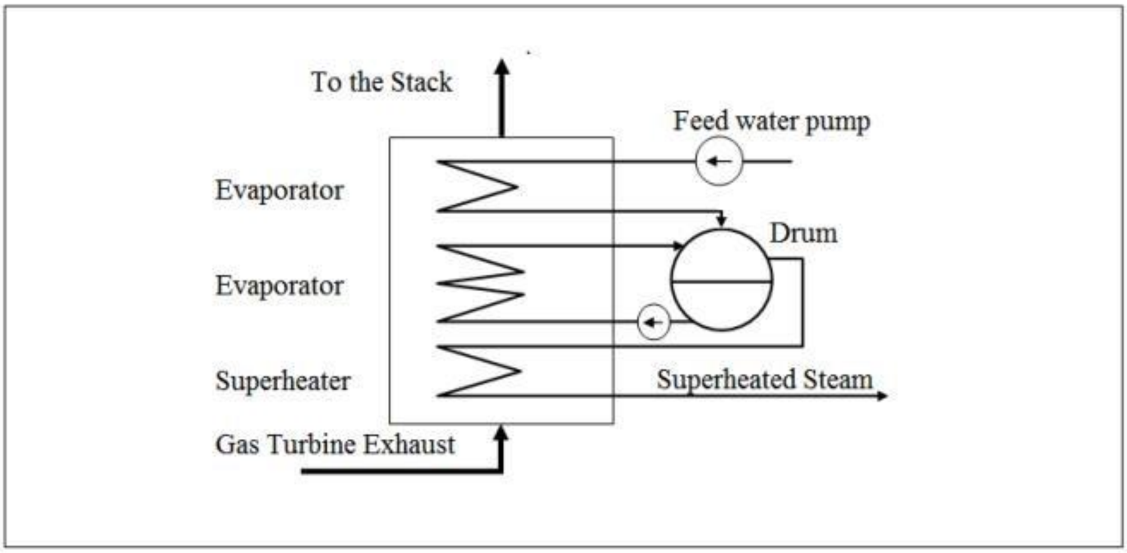


*Fig 3.7 Single pressure HRSG*

Natural circulation typically consists of vertical tubes and horizontal flow Arrangement.

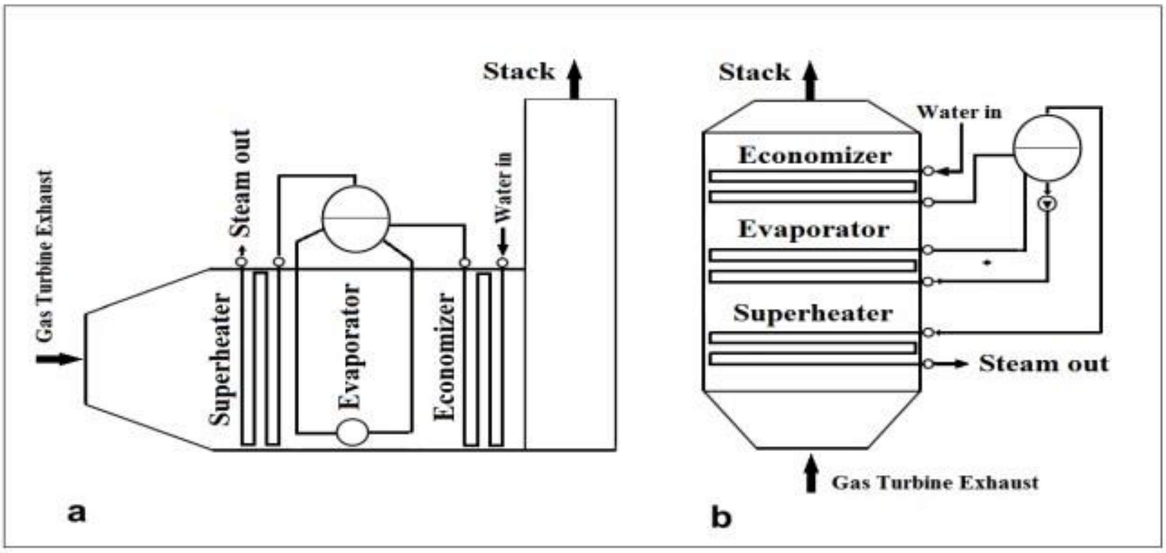


*Fig 3.8 HRSG (a) Natural circulation, (b) Forced circulation*

Circulation maintained up by the density contrasts between chilly water in lower chamber and hot steam-water mixture in evaporator tubes appear in Figure 3.8a.

Forced Circulation; HRSGs are portrayed by horizontal tubes with vertical gas stream and utilize pumps to circle steam-water blend (Figure 3.8b)

### 3.1.7.1 Important HRSG Performance and Design Parameters

### 3.1.7.1.1 HRSG Main Design Parameters

In general, when designing of any gas turbine HRSG the following parameters should be considered:

**a)** Pinch point temperature: Pinch point temperature is the contrast between the saturation temperature of water and the gas temperature of the gas leaving the evaporator (Figure 3.9).

**b)** Economizer approach temperatures (approach point): the approach point is the difference between the temperature of saturated steam and the temperature of the water entering the evaporator (Figure 3.9). Selection of these two variables also affects the size of the superheater, the evaporator, and the economizer. The smaller temperature difference means that the surface area required to produce the same heat transfer will be much greater. The direct consequence is that more material is used and hence capital cost is higher.

**c)** Steam pressure and temperature are the outlet parameters from HRSG supplied to the steam turbine. These parameters are selected to provide an economical design.

**d)** Superheater approach temperatures: the difference between the superheating steam temperature in the superheater and gas turbine outlet temperature.

**e)** Stack outlet temperatures: the temperature of the gases that leave the HRSG to the atmosphere (Figure 3.9).

**f)** Allowable backpressure: The HRSG cross-sectional zone fundamentally impacts the gas turbine backpressure. smaller, compact HRSGs require higher gas turbine backpressures to drive through the pipe gas, be that as it may, while the size decrease may diminish HRSG cost, the necessity to give a higher pressure at the turbine exit detrimentally affects gas turbine efficiency. The run of the mill estimations of gas turbine back-pressure are 2.5 to 3.7 KPa in many units [40].

#### 3.1.7.1.2 T-Q Diagram of HRSG

The T-Q diagram demonstrates profiles for the heat transfer prepare between fumes gas and water/steam, utilizing temperature on the ordinate axis and heat exchanged on the abscissa axis.

The use of T-Q diagram is crucial in understanding and designing combined cycles. Figure 3-10 shows the T-Q diagram for a single-pressure combined cycle. The smallest temperature difference in the HRSG is called the pinch-point, and it is located on the cold side of the evaporator. The upper line, with an almost constant slope, represents the temperature profile of the flue-gas, and the lower line represents the temperature of the water/steam. The HRSG of a single-pressure combined cycle consists of three different sections.

**First section HRSG**: starting at the lowest temperature, the first section is called the economizer, and is the place where liquid water is heated to the saturation temperature. To avoid evaporation, which could cause steam blockage that may result in "water hammering" in the economizer, the outlet temperature is always kept a few degrees

below the saturated state. This temperature difference is called the approach point.

**Second section:** is the evaporator, in which the water is evaporated at constant temperature.

**Third section HRSG** is the super heater where the evaporated steam is superheated.

The relation between temperature and heat can be described by:

$$\dot{Q} = \dot{m} c_p \Delta T \tag{3.1}$$

This equation (3.1) is valid when the working medium does not undergo a phase transition. In the HRSG, a phase transition from water to steam occurs in the evaporator [54], which means that equation (3-2) must be replaced by:

$$\dot{Q} = \dot{m} \Delta h_{evap} \tag{3.2}$$

where the energy is transferred and is the evaporation enthalpy. If equation (3.1) is rearranged it can be seen that the slope of the line in the T-Q diagram is inversely proportional to the mass flow and the specific heat. From equation (3.2) it can be understood that the term $\dot{m}\Delta h_{evap}$ is the length of the evaporation line. These relations are central for the analysis of combined cycle power plants.

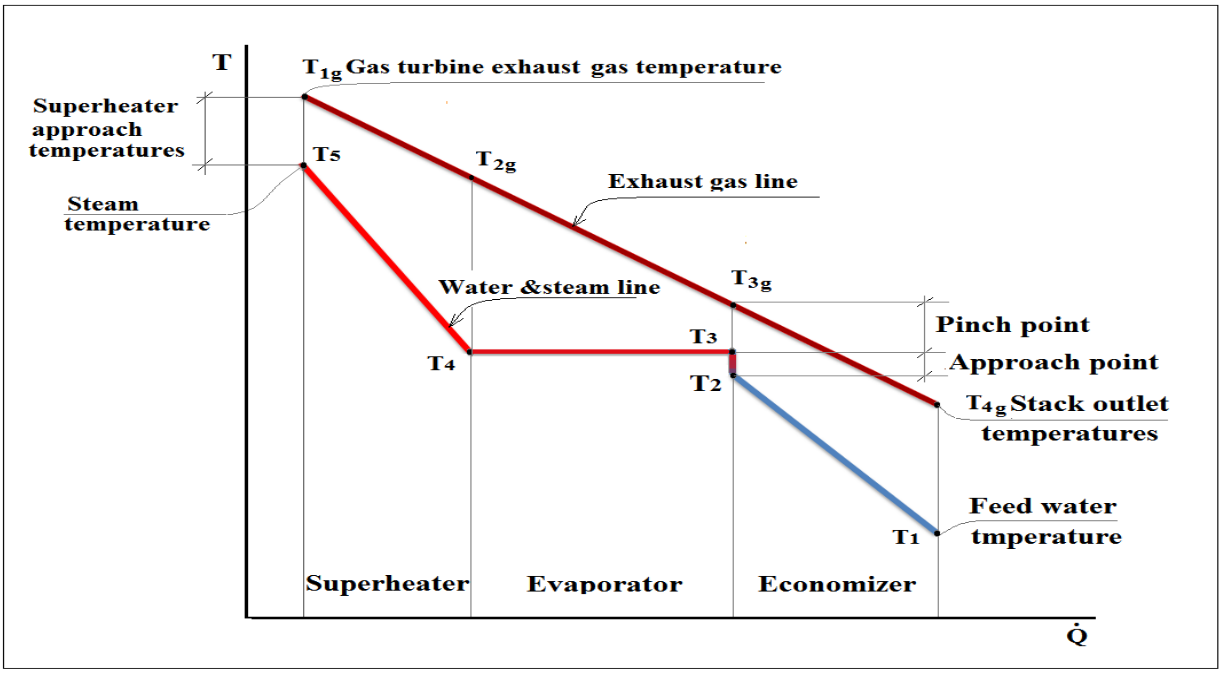


*Figure 3.9: T-Q Diagram of HRSG*

### 3.1.8 Combined Cycle Power Plant

Conventional combined cycle power plant consists of the combination of Brayton and Rankin cycle forming one of the most efficient cycles used for power generation today. In the Brayton Cycle, there is the gas turbine cycle, also called the topping cycle and the Rankin Cycle is the steam turbine cycle, also described as bottoming cycle. Thermal efficiency of the combined cycle plants is somewhat higher today and exceeds 60%.

Gas-turbine cycles regularly work at significantly higher temperatures than steam cycles. The most extreme fluid temperature at the turbine inlet is around 620°C for current steam power plants, however more than 1425°C for gas-turbine power plants. It is more than 1500°C at the burner exit of turbojet motors. The utilization of higher temperatures in gas turbines was made conceivable by late improvements in cooling the turbine blades and coating the blades with high-temperature safe materials, for example, pottery.

As a result of the higher normal temperature at which heat is provided, gas turbine cycles have a more noteworthy potential for higher thermal efficiencies. Be that as it may, the gas turbine cycles have one innate weakness: the gas leaves the gas turbine at high temperatures (as a rule over 500°C), which deletes any potential picks up in the thermal efficiency. The circumstance can be enhanced to some degree by utilizing recovery, however, the change is restricted. It bodes well to exploit the extremely attractive qualities of the gas-turbine cycle at high temperatures and to utilize the high-temperature exhaust gasses as the energy sources for the bottoming cycle, for example, a steam power cycle. In this cycle, energy is recuperated from the exhaust gasses by exchanging it to the steam in a heat exchanger that fills in as the boiler. All in all, (at least one) gas turbine is expected to supply adequate heat to the steam. Likewise, the steam cycle may include regeneration and also reheating. Energy for the heating procedure can be provided by consuming some extra fuel in the oxygen-rich exhaust gasses.

The gas turbine flue gas temperature is inside the range 450-650 °C. The energy contained in the vent gas is a measure of the fuel energy that is not changed over to control by the gas turbine. This energy is utilized to raise steam and to create power by the steam turbine. Contingent on the type of HRSG the pipe gas temperature is diminished to 80-200 °C, where the lower esteem is normal for vast present day combined cycle consuming a fuel with no or next to no sulphur. The steam is delivered with a temperature in the range 450-560 °C, and a pressure in the range 30-170 bar. Steam might be delivered at different pressure levels. The utilization of supercritical steam pressure (>220.64 bar) was proposed by Bolland. Supercritical steam pressure gives the combined cycle possibly higher efficiency, yet it relies on upon size of the steam turbine and reliance of pressure on the steam turbine efficiency. The cycle working at the higher temperatures is known as the topping cycle, while the cycle reutilizing the energy flux is the bottoming cycle. The Combined Cycle Gas Turbine power plant utilizes the Brayton cycle as the topping cycle while the Ranking cycle goes about as the bottoming cycle.

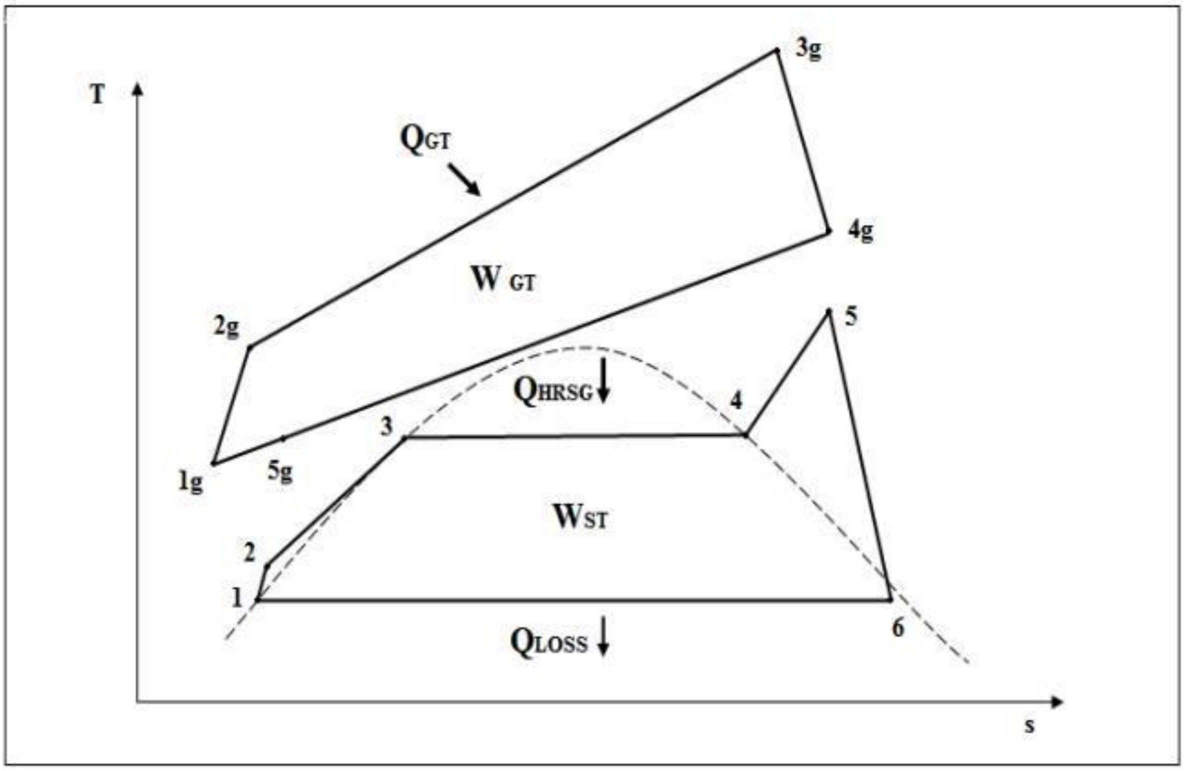


*Fig 3.10 T-s diagram of the combined cycle gas turbine CCGT process*

# CHAPTER 4

## 4.Exergy Analysis and optimization-Background

This chapter provides a fundamental background for the development of exergoeconomic optimization, with the governing equations necessary to achieve the optimization purposes. The exergoeconomic optimization technique consists of the following steps.

### 4.1 Energy Analysis

An energy investigation depends on the “first law of thermodynamics”. The first law of thermodynamics is all the more usually known as the law of energy conversion. The first law of thermodynamics demonstrates that energy can neither be made nor annihilated, and it can just change starting with one shape then onto the next. This law characterizes inner vitality as a state work, and gives a formal articulation of the conservation of energy. The most common energy systems, such as power generation and refrigeration systems, are open systems (systems in which mass flows through the various components). The typical components of power and refrigeration systems are boilers, turbines, evaporators etc., all of which have inlets and outlets.

An energy study is generally used in estimating the performance of a part or a system, and can be used to regulate the first law efficiency ($\eta_{th}$) for a power production cycle.

Be that as it may, an energy adjust gives no data about the heading in which procedures can suddenly happen or potentially the reversibility of the thermodynamic procedures. The first law can't give data about the powerlessness of any thermodynamic procedure to change over heat completely into mechanical work, or any understanding into why mixture can't suddenly isolate themselves.

### 4.2 Exergy Analysis

Exergy can be defined briefly as: the maximum theoretical work obtained from a system when this system is brought from a state to equilibrium with the environment while interacting only with the environment. The state of a system is defined by temperature, pressure, and composition. Moreover, exergy is the base theoretical work expected to carry the system from equilibrium with nature to the given state. This implies exergy is a measure of the take-off of the condition of a system from the condition of the surroundings. This makes exergy a property of both the system and condition together. The meaning of exergy won't be finished, in any case, until we characterize the reference condition.

Exergy investigation is a strategy that uses the Preservation of mass and protection of energy standards together with the second law of thermodynamics for the examination, plan, and change of energy system. The exergy technique is a valuable instrument for facilitating the objective of more productive energy resource use, for it empowers the areas, types, and genuine magnitudes of squanders and losses to be resolved.

### 4.2.1 Exergy of a System

The total exergy is that exergy that can be extracted through heat and work processes, hence,

$$\Delta\dot{E}_{system} = \Delta\dot{E}_{heat} - \Delta\dot{E}_{work} \qquad 4.1$$

**Exergy associated with work transfer:** from the definition of the work, that equivalent a given type of energy as a measure of its exergy, unmistakably work is comparable to exergy in each regard. In this way, exergy transfer can be indicated both in size and in bearing by the work transfer to which it corresponds

$$\Delta\dot{E}_{work} = W + \int P_0\, dV \qquad 4.2$$

**Exergy associated with heat transfer:** supposing a constant temperature dispersal in a thermal energy reservoir, the exergy transfer rate, $\Delta\dot{E}_{work}$ associated with the heat transfer rate Q, can be intended by the subsequent method work:

$$\Delta\dot{E}_{heat} = (1 - \frac{T_0}{T})\dot{Q} \qquad 4.3$$

substituting equation (4.2) and (4.3) in equation (4.1),

$$\Delta\dot{E}_{heat} = (Q - T_0 \int dS) - (W - \int P_0 dV) \qquad 4.4$$

and rearranging

$$\Delta\dot{E}_{heat} = Q - W - T_0 \int dS + \int P_0 dV \qquad 4.5$$

Introducing the first law:

$$\Delta E = Q - W \quad 4.6$$

eliminates Q and W yielding,

$$\Delta \dot{E}_{system} = \Delta E - T_0 \int dS + \int P_0 dV \quad 4.7$$

expanding

$$\Delta \dot{E} = \Delta U - T_0 \int dS + \int P_0 dV + m\frac{1}{2}V^2 + mgz \quad 4.8$$

Integrating the above amongst the state of the system and the dead state produces,

$$\dot{E} = U - U_0 - T_0(S - S_0) + p_0(V - V_0) + m\frac{1}{2}V^2 + mgz \quad 4.9$$

Here, V is the velocity of the system relative to the reference frame of the environment and z is the height of the system relative to the reference frame of the environment (where $z_0 = 0$ usually the ground).

## Exergy Component

The overall exergy of a system $\dot{E}$ can be divided into four components: physical exergy $\dot{E}_{PH}$, kinetic exergy $\dot{E}_{KN}$ , potential exergy $\dot{E}_{PT}$ , and chemical exergy $\dot{E}_{CH}$:

$$\dot{E} = \dot{E}_{PH} + \dot{E}_{KN} + \dot{E}_{PT} + \dot{E}_{CH} \quad 4.10$$

### 4.2.2.1 Kinetic Exergy

Kinetic exergy, $\dot{E}_{KN}$ is equal to kinetic energy calculated with the velocity of movement with respect to the environment:

$$\dot{E}_{KN} = m\frac{1}{2}V^2 \quad 4.11$$

### 4.2.2.2 Potential Exergy

Potential exergy, $\dot{E}_{PT}$ is determined with potential energy with respect to zero level associated to the surroundings. Since potential energy must consider every one of the strengths influencing the analysed matter and condition, it implies that, other than the surrounding constraint, one ought to likewise consider the drive created by the pressure of nature parts. Accepting that the speeding up of gravity does not fluctuate with height, potential energy can be figured from the accompanying:

$$\dot{E}_{PT} = m(gz - g\int_0^h \gamma_0\, dh) \qquad 4.12$$

where g , h and $\gamma_0$ are gravity acceleration, height of matter centre with respect to zero level and environment density respectively.

### 4.2.2.3 Physical Exergy

Physical exergy speaks to a piece of the exergy that shows up because of the distinction in temperature and pressure of the observed matter and the temperature and pressure of the environment$(T_0, P_0)$. It is naturally divisible into two components.

Thermal component $E_{PH}^{\Delta T}$, is the component ensuing from the temperature change between the stream and the environment:

$$E_{PH}^{\Delta T} = mC_p\left[(T - T_0) - T_0 \ln\frac{T}{T_0}\right] \qquad 4.13$$

Pressure component $E_{PH}^{\Delta P}$ , the component ensuing from the pressure change between the stream and the environment:

$$E_{PH}^{\Delta P} = (h - h_0) + T_0(s - s_0) \qquad 4.14$$

### 4.2.2.4 Chemical Exergy

In deciding physical exergy, the last condition of stream is the ecological state. Presently, this state will be the underlying state in the reversible procedures that are utilized to decide the chemical exergy of this material stream. As per the meaning of exergy, the last state to which the substance will be lessened is the standard dead state. Along these lines, concoction exergy is characterized as the most extreme work realistic when the substance under thought is conveyed from natural state to the standard dead state by a process including heat exchange and trade of substances just with the environment.

The general form of chemical exergy equation of mixture can be written as,

$$\dot{E}_{CH}=\sum x_n(\dot{e_0})_n + \bar{R}T_0\sum x_n \ln x_n \qquad 4.15$$

$$\dot{e_0} = T_0 \ln\frac{P_0}{P_{00}} \qquad 4.16$$

here, $x_n$ is the mole fraction of the $k_{th}$ gas in the mixture, $\dot{e_0}$ is the chemical exergy of reference substances, $\bar{R}$is the universal gas constant and $P_{00}$ is partial pressure of the gaseous reference substance.

Equation for the fuel specific exergy given as:

$$\dot{e}_{CH,fuel} = \beta(LHV) \qquad 4.17$$

$$\beta = 1.0437 + 0.1882\left(\frac{h}{c}\right) + 0.0610\left(\frac{o}{c}\right) + 0.0401\left(\frac{n}{c}\right) \qquad 4.18$$

where c, h, o, n are the mass fractions of C, H, O, N respectively.

### 4.2.3 Exergy Balance

By combining the first and second law of thermodynamics, the mathematical equation of an exergy balance is [44]

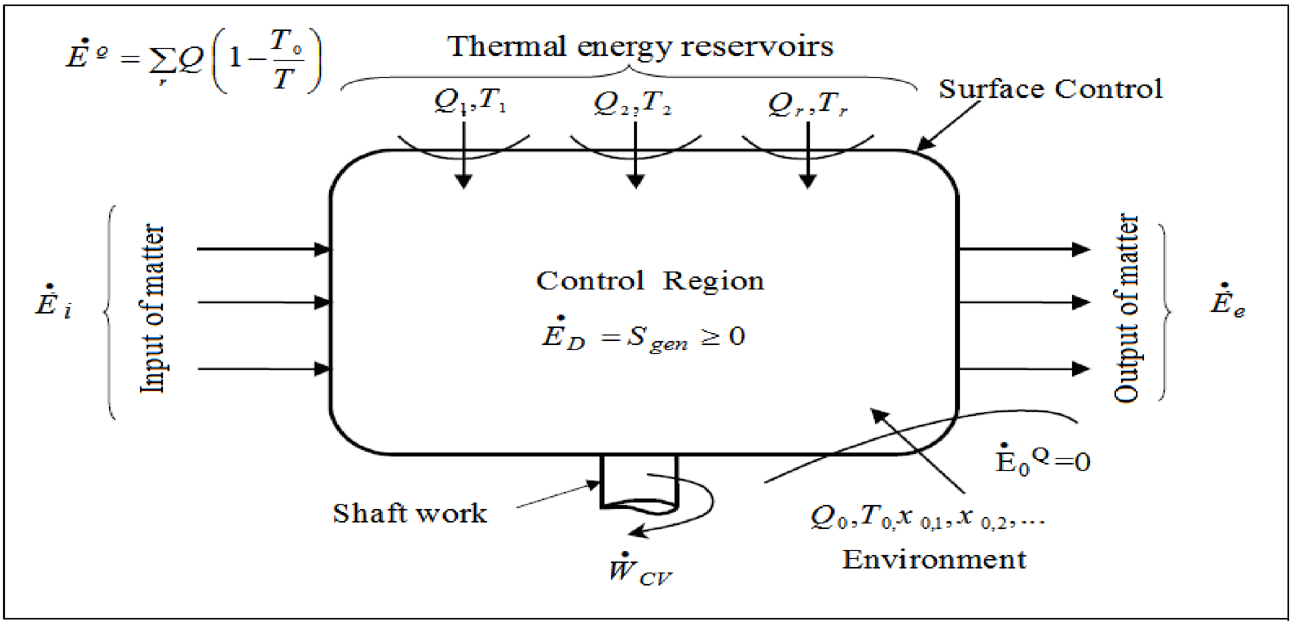


*Figure 4.1: Steady state process in an open control region*

An expression of an exergy analysis for an open system is:

$$\begin{bmatrix} Rate\ of\ exergy \\ change\ within \\ control\ volume \end{bmatrix} = \begin{bmatrix} Net\ exery \\ transfer \\ by\ heat \end{bmatrix} + \begin{bmatrix} Net\ Exergy \\ transfer \\ by\ work \end{bmatrix} - \begin{bmatrix} Rate\ of \\ Exergy\ addition \\ with\ mass \end{bmatrix} + \begin{bmatrix} Rate\ of \\ Exergy\ removal \\ with\ mass \end{bmatrix} + \begin{bmatrix} Exergy \\ destruction \\ within \\ control\ volume \end{bmatrix} \quad 4.19$$

$$\left(\frac{d\dot{E}}{dt}\right) = \sum_j (1-\frac{T_0}{T_j})\dot{Q}_j\text{-}W_{CV} + \sum m_i \dot{e}_i - \sum m_e \dot{e}_e - \dot{E}_D \quad 4.20$$

Where $\left(\frac{d\dot{E}}{dt}\right)$ is the time rate of change of the exergy stored within the control volume, $\left(1-\frac{T_0}{T_j}\right)\dot{Q}_j$ is the net exergy change due to heat transfer, $T_j$ is the temperature at $^{\text{jth}}$ control volume boundary (where the heat is crossing into the control volume at that boundary); $W_{CV}$ is the net exergy transfer due to non-flow work out of the control volume; $\sum m\dot{e}$ is the exergy addition (i) or removal (o) due to mass flow; $\dot{E}_D = T_0 S_{gen}$ and is the exergy destruction within the control volume.

### 4.2.4 Exergy Wastes (Exergy Destructions and Exergy Losses)

The thermal system under consideration for analysis is supplied with some input (fuel exergy $\dot{E}_F$) derived from energy source. This input transfers into some exergy output (product exergy $\dot{E}_p$). For a real process the exergy input always exceeds the exergy output, and this unbalance is due to waste in exergy; it is useful to differentiate between types of exergy wastes in order to study where irreversibilities occur. Two kinds of exergy wastes can be distinguished: internal and external [43].

External exergy wastes (exergy losses $\dot{E}_{loss}$) represent the remaining exergy contents of losses and emissions that are dissipated or removed from the production and embody, thus unused (exergy remaining non-utilized output).

Internal exergy wastes (exergy destruction $\dot{E}_D$) correspond to the wastes of quality due to internal inefficiencies within the process, it is the direct result of the irreversibilities in a system. These internal irreversibilities may be of technical nature due to technical inefficiencies within the plant, e.g. friction or lack of insulation, or they may be of a structural nature. Structural exergy destruction $\dot{E}_D$ is determined by the principle and design of the system. Whereas technical exergy destruction $\dot{E}_D$ can be reduced through optimization, structural waste can be reduced only by redesigning the system.

The exergy loss is associated with the design engineer's choice not to further use the exergy of a stream in a given system (unexploited exergy, i.e. exergy flow to the environment). Both represent exergy waste, but irreversibilities have by definition, no exergy and no environment effects. The exergy destruction is related to the entropy generation equation.

$$\dot{E}_{loss} = \dot{E}_{out} - \dot{E}_p \quad 4.21$$

$$\dot{E}_D = \dot{E}_{in} - \dot{E}_{out} \quad 4.22$$

$$\dot{E}_{waste} = \dot{E}_D + \dot{E}_{loss} \quad 4.23$$

For the exergy analysis, it is necessary to define product and fuel for each component and for overall system. The product is defined according to the purpose of owning and operating the component under consideration and fuel represent the resources consumed in generating the product. Fuel and product are expressed in terms of exergy. Exergy destruction is the amount of exergy lost due to irreversibilities and cannot be used anywhere. The exergy losses are the amount of exergy that is lost from the system under consideration, but can be useful to other system.

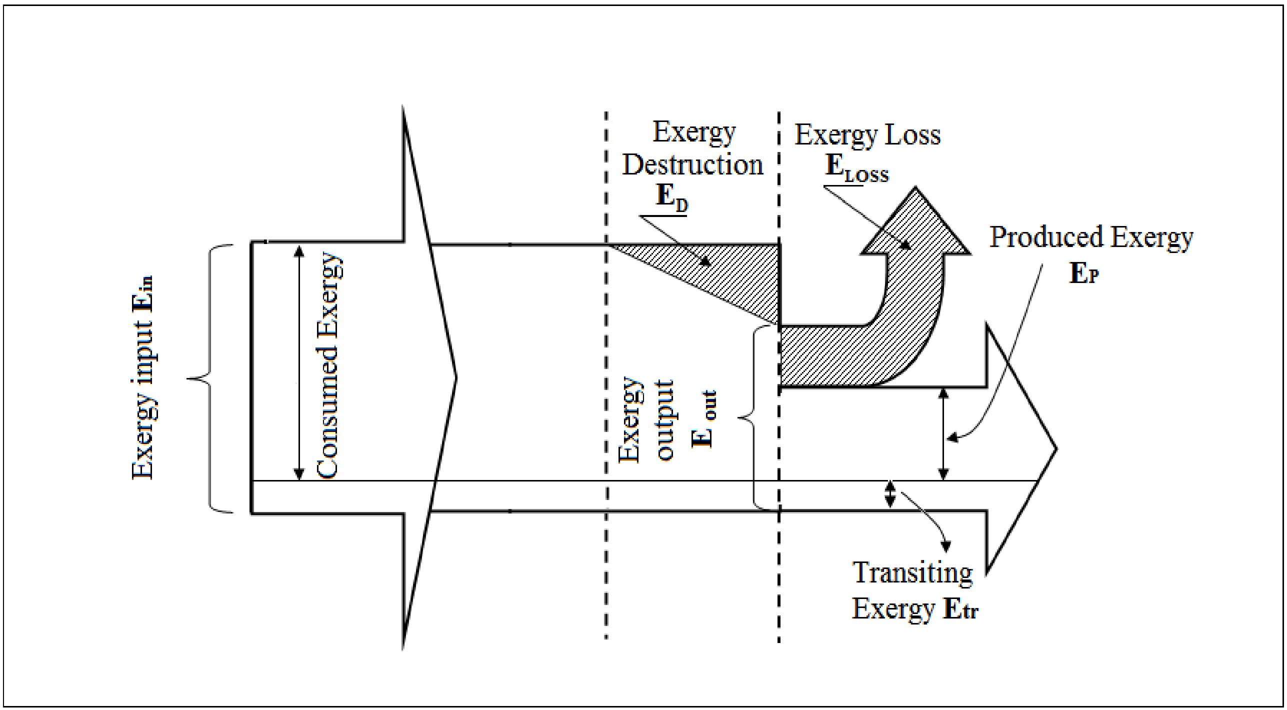


*Figure 4.2: Graphical presentation of overall exergy balance*

The exergy destruction can be calculated from equation (4.20), the exergy destruction equation for any control volume at steady state with insignificant kinetic and potential energy changes can be written as in equation (4.24).

$$\dot{E}_D = \sum_j (1 - \frac{T_0}{T_j})\dot{Q}_j\text{-}W_{CV} + \sum \dot{E}_i - \sum \dot{E}_o \qquad 4.24$$

An exergy balance, by definition, exists for reversible procedures. Subsequently, for genuine procedures, i.e. irreversible procedures, exergy is never in balance, in light of the fact that the aggregate exergy input dependably surpasses the aggregate exergy yield, i.e. Thus, it is deceiving to discuss an exergy adjust for genuine procedures.

By ascertaining the exergy waste, i.e. pulverization and loss, we can imagine conceivable process upgrades. By and large, when the exergy loss is high, we ought to consider enhancing this part first. In any case, this approach is not generally fitting. The reason is that all aspects of the system rely on upon each other and a change in one section may increase losses in different parts so that the aggregate losses in the balanced procedure might be equivalent or significantly bigger than in the first procedure setup. Along these lines, the issue needs an all the more precisely approach, which we will talk about beneath [45].

giving the exergy destruction ratio:

$$y_D = \frac{\dot{E}_D}{\dot{E}_{F,TOT}} \qquad 4.25$$

$\dot{E}_{D,TOT}$ giving the ratio:

$$y_D^* = \frac{\dot{E}_D}{\dot{E}_{D,TOT}} \qquad 4.26$$

exergy of the fuel provided to the overall system:

$$y_{loss} = \frac{\dot{E}_{loss}}{\dot{E}_{F,TOT}} \qquad 4.27$$

### 4.2.5 Exergy Efficiency

Exergy efficiencies can be used for various purposes. An obvious application is to use them for assessing, analysing, and optimizing processes and systems. Exergy efficiencies are particularly valuable in analysing and optimizing systems. An exergy investigation usually includes a comprehensive calculation of the exergy standards of process flows and the exergy waste in the system. Such a calculation shows the places in the system where waste occur. In the analysis, the question that has to be answered is how the exergy losses can be avoided or limited. Based on the absolute value of exergy loss, it is usually difficult to assess whether an exergy waste in an apparatus is unnecessarily large. An exergy efficiency in which the exergy loss is associated with the added or transferred exergy gives a better picture of the value of the processes in the apparatus, and thus gives a better impression of whether exergy waste can be reduced.

The calculation of exergy efficiency can be a rather difficult subject due to the lack of calibration and uncertainty of some terms set up in the literature.

Two main classes of exergy efficiencies definitions will be presented next, universal exergy efficiency and functional exergy efficiency, and several authors have provided these definitions [46].

#### 4.2.5.1 Universal Exergy Efficiency

Universal exergy efficiency is defined as a ratio of gross exergy yield to gross exergy input Figure 4-3. There are two main modules of universal exergy efficiencies reported in the literatures. The primary is simple efficiency and the second is the efficiency with transiting exergy [47].

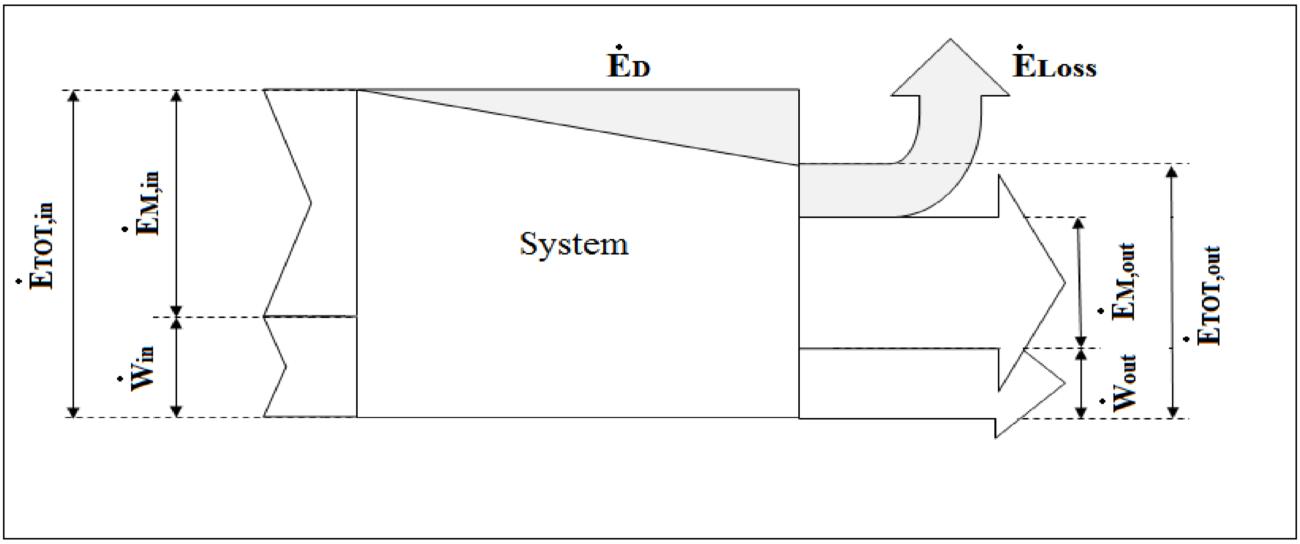


*Figure 4.3: Simple exergy efficiency*

Because of its simplicity, it is a generally applicable definition for exergy efficiency. This form of efficiency is based on the exergy balance to express all exergy input as used exergy, and all exergy output as utilized exergy equation (4-23). Therefore, the exergy efficiency $\varepsilon$ becomes:

$$\boldsymbol{\varepsilon} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = \mathbf{1}- \frac{\dot{E}_D}{\dot{E}_{in}} \qquad 4.28$$

However, this efficiency does not always deliver a passable classification of the thermodynamic efficiency of processes, such as heat transfer, separation, expansion etc.

There is often a part of the output exergy which is unused, i.e. an exergy loss $\dot{E}_{loss}$ to the environment; moreover, the exergy efficiency $\varepsilon_1$ becomes loss:

$$\varepsilon_1 = \frac{\dot{E}_{out} - \dot{E}_{loss}}{\dot{E}_{in}} \qquad 4.29$$

- Efficiency with Transiting Exergy

Efficiency with transiting exergy $\varepsilon_{tr}$ is seen as a development of the simple efficiency. The unreconstructed components are here subtracted from the inward and the outward-bound components. The efficiency will be defined by:

$$\varepsilon_{tr} = \frac{\dot{E}_{out} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} \qquad 4.30$$

Where $\dot{E}_{tr}$ is the transiting exergy and it is defined as the part of the exergy which negotiates a system without taking any part in the mechanical, thermal or chemical changes which take place in the system.

The universal efficiency offers a reasonable definition for an assortment of frameworks. A detriment of this definition, notwithstanding, is that the efficiencies values acquired can be cold-hearted to changes in the system.

### 4.2.5.2 Rational Exergy Efficiency (Functional)

Rational Efficiency: Rational efficiency is defined by [42]. This efficiency is given by the ratio of the anticipated exergy output to the exergy used:

$$\psi = \frac{\Delta\dot{E}_{out}}{\Delta\dot{E}_{in}} = \varepsilon_p = \frac{\Delta\dot{E}_P}{\Delta\dot{E}_F} \qquad 4.37$$

where $\Delta\dot{E}_{out}$, is the summation of all exergy transfers creating up input, and $\Delta\dot{E}_{in}$ is the summation of all exergy transfers building up output.

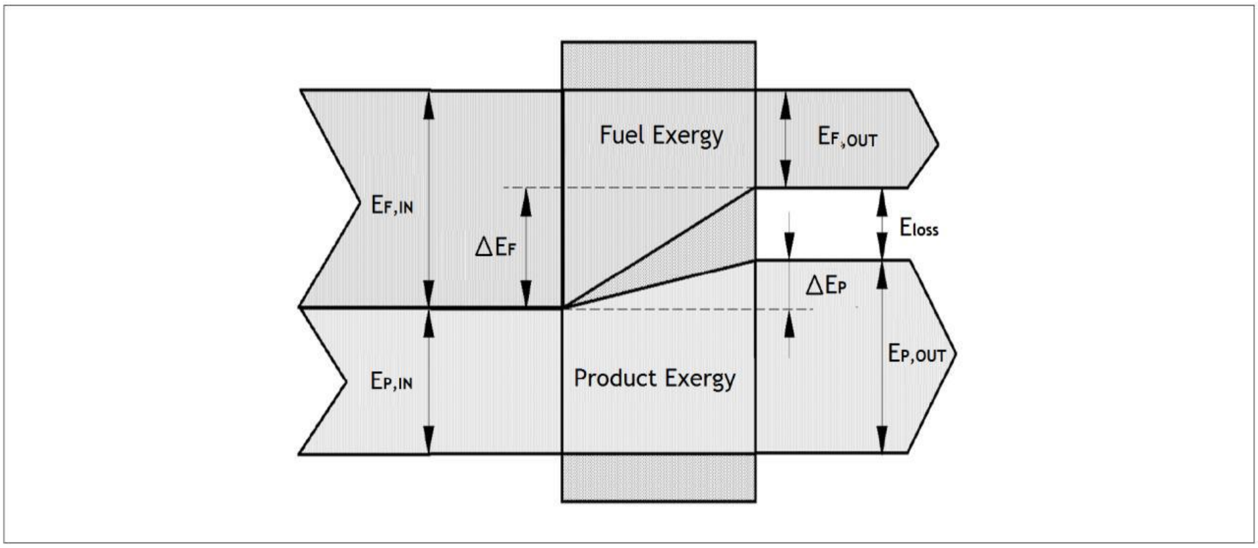


*Figure 4.4 Explanation of efficiency definitions (heat exchanger)*

# CHAPTER 5

## 5.Formulation of Equations

## 5.1 Exergy Analysis of Gas Cycle

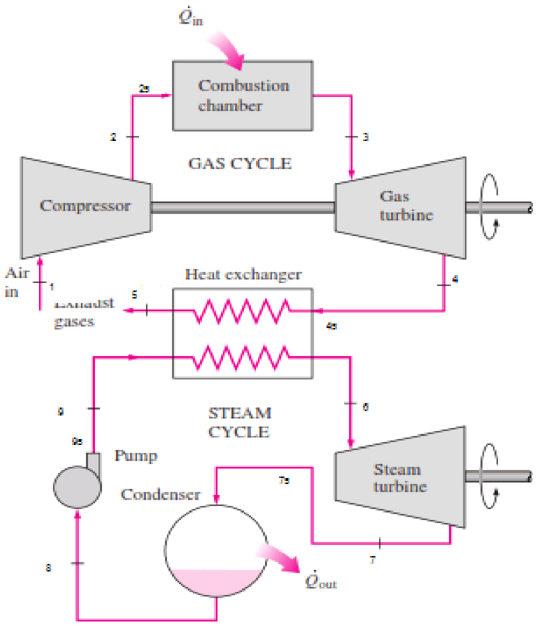


*Fig. 5.1: Schematic diagram of combined cycle gas power plant*

| Point number | Specification |
|---|---|
| 1 | Inlet air entering compressor |
| 2 | Outlet air from compressor |
| 3 | Combustion gases exiting combustion chamber |
| 4 | Outlet hot gases exiting gas turbine |
| 5 | Outlet gases exiting HRSG |
| 6 | Superheated steam entering steam turbine |
| 7 | Outlet steam from steam turbine |
| 8 | Saturated liquid entering feed water pump |
| 9 | Supplied water entering HRSG |

**Table 5.1: Table indicating various points of schematic diagram**

The basic equations employed in the exergy analysis performed on the selected combined cycle power plant are presented in this section. As with the energy analysis, exergy balances for individual components are written, and exergy flows and irreversibilities for each component are found. Then, overall exergy efficiency and exergy destruction are found for the whole system.

The exergy components of each state of the bottoming cycle shown in Figure 5.1 are calculated according to equation (4.14) for physical exergy and equation (4.15) for chemical exergy.

Exergy destruction and exergy efficiency equations for each element in the bottoming cycle are calculated by applying the exergy balance around the element using equation (4.24).

It should be noted that exergy transfer due to heat loss from HRSG is taken as zero due to assumption of ideal insulation.

### 5.1.1 Compressor

$$X_1 + W_C = X_{2s} + \Delta X_{dest.} \qquad 5.1$$

$$\Delta X_{dest.} = X_1 - X_{2s} + W_c \qquad 5.2$$

Where

$$X_1 = \dot{m}_a[h_1 - h_0 - T_0(s_1 - s_0)] \qquad 5.3$$

$$X_{2s} = \dot{m}_a[h_{2s} - h_0 - T_0(s_{2s} - s_0)] \qquad 5.4$$

### 5.1.2 Combustion Chamber

$$X_{2s} + X_f = X_3 + X_{cc} \qquad 5.5$$

$$X_f = X_{ph} + X_{ch} \qquad 5.6$$

$$X_{ph} = \dot{m}_a[h_f - h_0 + T_0(s_f - s_0)] \qquad 5.7$$

$$X_{ch} = \dot{m}_f e_{ch} \qquad 5.8$$

Where

$\boldsymbol{e_{ch}} = \dot{\boldsymbol{x}}_{\boldsymbol{i}} \boldsymbol{e_{chi}} + \boldsymbol{RT_o} \sum \boldsymbol{x_i} \ln \boldsymbol{x_i} + \boldsymbol{G_e}$ 5.9

Where $\mathbf{G_e}$ is Gibbs free energy which is a negligible quantity in a gas mixture operated at low pressure. So for the calculation of fuel exergy, the given expression does not hold good. Thus, the fuel exergy can be calculated as the ratio of fuel exergy to lower heating value of fuel-

$$\Omega = \frac{e_f}{LCV} \quad 5.10$$

$\boldsymbol{e_f}$ is the specific exergy of the fuel.

For gaseous fuel with composition $\boldsymbol{C_x H_y}$, the value of Ω can be calculated as

$$\Omega = 1.033 + 0.0169\frac{Y}{X} - \frac{0.0698}{X} \quad 5.11$$

For Methane**($CH_4$) X=1, Y=4**

Then

**Ω=1.06**

$$X_f = \dot{m}_f(1.06*\mathrm{LCV}) \quad 5.12$$

### 5.1.3 Gas Turbine

$$\Delta X_{dest.} = X_3 - X_{4s} - W_T \quad 5.13$$

Where

$$X_{4s} = (\dot{m}_a + \dot{m}_f)[h_{4s} - h_0 + T_0(s_{4s} - s_0)] \quad 5.14$$

$$X_3 = (\dot{m}_a + \dot{m}_f)[h_3 - h_0 + T_0(s_3 - s_0)] \quad 5.15$$

### 5.1.4 HRSG

$$\Delta X_{dest.} = X_{in} - X_{out} \quad 5.16$$

Where,

$$X_{4s} = (\dot{m}_a + \dot{m}_f)\psi_{4s} \qquad 5.17$$

$$X_{9s} = \dot{m}_w\psi_{9s} \qquad 5.18$$

$$X_{in} = X_{4s} + X_{9s} \qquad 5.19$$

$$X_5 = (\dot{m}_a + \dot{m}_f)\psi_5 \qquad 5.20$$

$$X_6 = \dot{m}_w\psi_6 \qquad 5.21$$

$$X_{out} = X_5 + X_6 \qquad 5.22$$

$$\eta_{II} = \frac{X_{9s} - X_6}{X_{4s} - X_5} \qquad 5.23$$

$$(X_{4s} - X_5) = (\dot{m}_a + \dot{m}_f)[(h_{4s} - h_5) - T_0(s_{4s} - s_5)] \qquad 5.24$$

$$(X_{9s} - X_6) = (\dot{m}_w)[(h_{9s} - h_6) - T_0(s_{9s} - s_6)] \qquad 5.25$$

### 5.1.5 Steam Turbine

$$X_6 = \dot{m}_w\psi_6 \qquad 5.26$$

$$X_{7s} = \dot{m}_w\psi_{7s} \qquad 5.27$$

$$X_6 - X_{7s} = \dot{m}_w[(h_6 - h_{7s}) - T_0(s_6 - s_{7s})] \qquad 5.28$$

### 5.1.6 Condenser

$$X_{in} = \dot{m}_w\psi_{7s} + \dot{m}_{cw}\psi_{cw,in} \qquad 5.29$$

$$X_{out} = \dot{m}_w\psi_8 + \dot{m}_w\psi_{cw,out} \qquad 5.30$$

### 5.1.7 Pump

$$X_8 = \dot{m}_w[(h_8 - h_0) - T_0(s_8 - s_0)] \qquad 5.31$$

$$X_{9s} = \dot{m}_w[(h_{9s} - h_0) - T_0(s_{9s} - s_0)] \qquad 5.32$$

| Components | Exergy Destruction Rate | Exergetic Efficiency |
|---|---|---|
| Compressor | $X_1 - X_{2s} + W_c$ | $\frac{X_{2s} - X_1}{W_c}$ |
| Combustion Chamber | $X_{2s} + X_f - X_3$ | $\frac{X_3}{X_{2s} + X_f}$ |
| Gas Turbine | $X_3 - X_{4s} - W_{GT}$ | $\frac{W_{GT}}{X_3 - X_{4s}}$ |
| HRSG | $(X_{4s} + X_{9s}) - (X_5 + X_6)$ | $\frac{X_{9S} - X_6}{X_{4S} - X_5}$ |
| Steam Turbine | $X_6 - X_{7s}$ | $\frac{W_{ST}}{X_6 - X_{7s}}$ |
| Condenser | $X_{in} - X_{out}$ | $1 - \frac{\Delta X_{dest.,Cond}}{X_{in}}$ |
| Pump | $X_8 - X_{9s} + W_P$ | $\frac{X_8 - X_{9s}}{W_P}$ |

**Table 5.2: Component wise Exergetic destruction rate and Exergetic Efficiencies**

# CHAPTER 6

## 6. Results and Discussion

After the study of different cycles on which the thermal power plant work with respect to exergy. The exergy destruction shows a loss that can be recovered by using the suitable design of the various portions of the system and also it confirms the best possible process of the power plant according to “second law of Thermodynamics”. As the exergy and cost of energy are complimentary to each other exergy destruction shows a loss, which can be quantify by analysis the system in mathematically. In the present work the analysis is done in the combined cycle power plant.

## 6.1 Results from Exergy analysis

### 6.1.1 Exergy Destruction rate

#### 6.1.1.1 Effect of A/F (air fuel ratio) at different pressure ratio

Fig. 6.1 portrays exergy destruction of air compressor as a function of A/F ratio at various pressure ratios. Pressure ratio was varied from 5 to 30 in a step of 5 while air fuel ratio was varied from 50 to 130 in a step of 10. With the increase in air fuel ratio, exergy destruction rate of air compressor increases. Here, the mass of fuel remains constant as 1Kg and mass of air increases so air fuel ratio. To compress more air, compressor has to work more and it results in increased exergy destruction rate. At a particular ratio, as pressure ratio increases exergy destruction rate increases too. This is because more work required by compressor and work done required by compressor is directly proportional to the pressure ratio.

Fig 6.2 demonstrates the variation of Exergy destruction rate of combustion chamber as a function of air fuel ratio at various pressure ratios. Air fuel ratio was varied from 50 to 130 in a step of 20. Pressure ratio was varied from 5 to 30 in a step of 5.

With the increase in air fuel ratio, the exergy destruction rate increases. This is due to the increased amount of heat addition in combustion chamber and it results in increment of exergy destruction rate.

At a particular air fuel ratio, as pressure ratio increases the exergy destruction rate decreases. This happens because due to increased pressure ratio, combustion chamber receives the air with high temperature so it requires less chemical energy addition.

At a particular air fuel ratio, with increasing pressure ratio, the marginal exergy destruction rate decreases. This is due to the reason; available exergy after the compressor is less for higher pressure ratios owing to higher destruction rate in the compressor.

At lower AFRs, the marginal increment in exergy destruction rate is more rapid as compared to that at higher air fuel ratios. This is due to the reason at lower air fuel ratio, the available exergy is higher owing to the fact that average temperature of available heat is higher which goes on decreasing with increase in air fuel ratio.

Fig 6.5 Illustrates the variation of Exergy destruction rate of HRSG as a function of air fuel ratio at various pressure ratios. As air fuel ratios increasing, exergy destruction rate is decreasing very rapidly because at lower air fuel ratio, the inflow temperature is high and decreasing with increase in AFR that results in fall of destruction rate.

At a particular inlet temperature, with increase in air fuel ratio the exergy destruction rate increases. This is due to the increased amount of heat addition in combustion chamber and it results in increment of exergy destruction rate.

At a particular inlet temperature, as air fuel ratio increases, marginal exergy destruction rate decreases.

Figure 6.7 displays the Exergy Destruction Rate of steam turbine at Various Pressure Ratios versus Air Fuel Ratio. The temperature increased by compressor at low pressure ratio is not dominant as compared to temperature decreased due to addition of air fuel ratio. That's why a specific decreasing trend is visible at low pressure ratio. On increasing the pressure ratio, the temperatures increased is very much high as compared to temperature decreased by increasing air fuel ratio that's why exergy destruction rate continuously increases on increasing air fuel ratio and at higher pressure ratio.

Figure 6.8 indicates the Exergy Destruction Rate of condenser at Various Pressure Ratios versus Air Fuel Ratio. This shows the same pattern as of steam turbine and reason is same too.

### 6.1.2 Exergetic Efficiency

#### 6.1.2.1 Effect of A/F (air fuel ratio) at different pressure ratio

Fig. 6.3 demonstrates the variation of Exergetic Efficiency of Air Compressor as a function of air fuel ratio at various pressure ratio. Air fuel ratio was varied from 50 to 130 in a step of 10. Pressure ratio was varied from 5 to 30 in a step of 5. At a particular air fuel ratio, the exergetic efficiency of air compressor continuously decreases with increase in pressure ratio. This is due to the reason that on increasing pressure ratio it increases the compressor work. As the air fuel ratio increases, exergetic efficiency remains constant. This is because exergetic efficiency not a function of air fuel ratio.

Fig. 6.4 demonstrates the variation of Exergetic Efficiency of combustion chamber as a function of air fuel ratio at various pressure ratio. At a particular air fuel ratio, the exergetic efficiency is increasing with increase in pressure ratio because the heat required in combustion chamber is decreased due to high temperature coming from compressor. At a particular pressure ratio, as air fuel ratio is increasing the inflow exergy from compressor is decreasing due to decreased temperature. That's why it is making the exergetic efficiency of combustion chamber increased.

Fig. 6.6 depicts Exergetic efficiency of HRSG at Various Pressure Ratios as a function of air fuel ratio. It is increasing because as air fuel ratio is increasing the destruction rate is decreasing which makes the overall efficiency increasing. At lower pressure ratio, exergetic efficiency increases very rapidly as compared to higher pressure ratio because temperature of exhaust gases entering to HRSG is low so exergy destruction rate is also low so exergetic efficiency is high as vice versa for higher pressure ratio.

Figure 6.9 illustrates the Exergetic Efficiency of combustion Chamber as a function of Inlet Temperature with AFR. At a particular inlet temperature, exergetic efficiency increases as AFR increases because of increase in air which reduces the available temperature at the inlet of combustion chamber which ultimately decreases the exergy entering the combustion chamber which results in increased exergetic efficiency of combustion chamber which increasing AFR.

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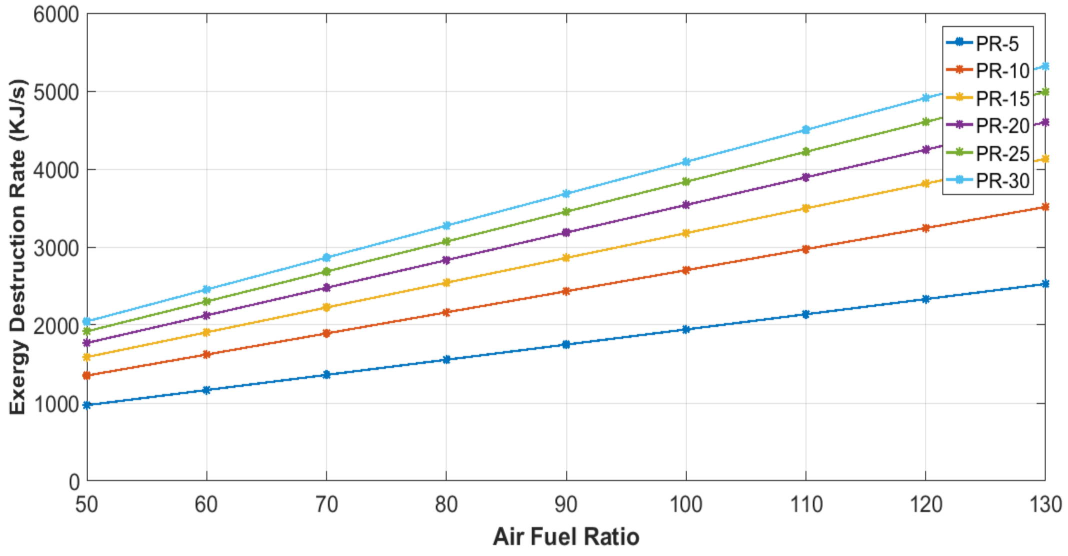
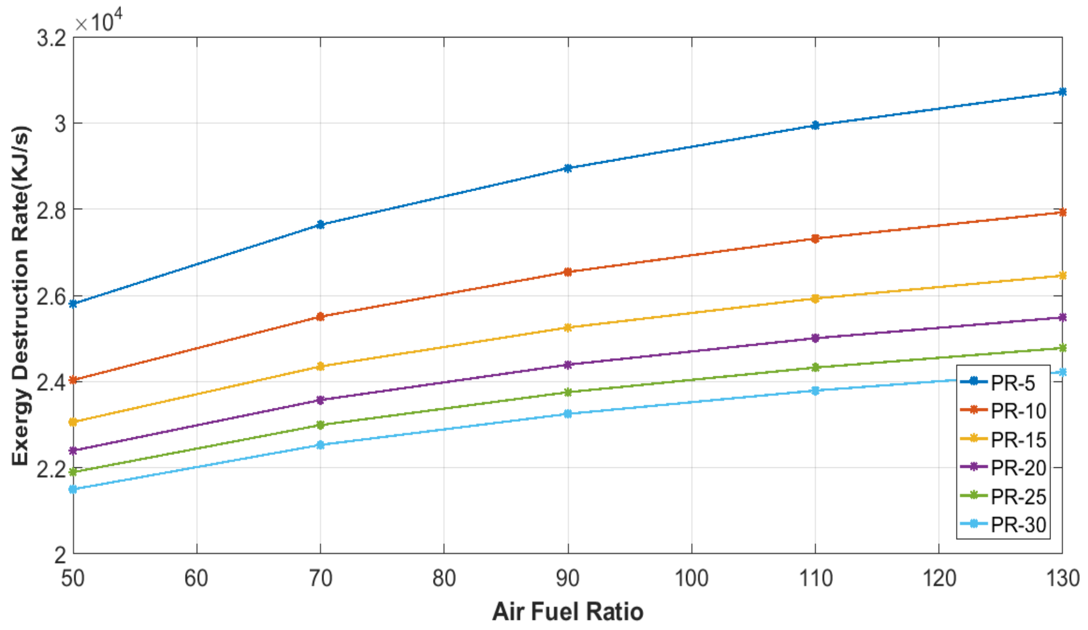



Figure 6.1: Exergy Destruction Rate of Air compressor at Various Pressure Ratios VS




Air Fuel Ratio

Figure 6.2: Exergy Destruction Rate of Combustion chamber at Various Pressure Ratios VS Air Fuel Ratio

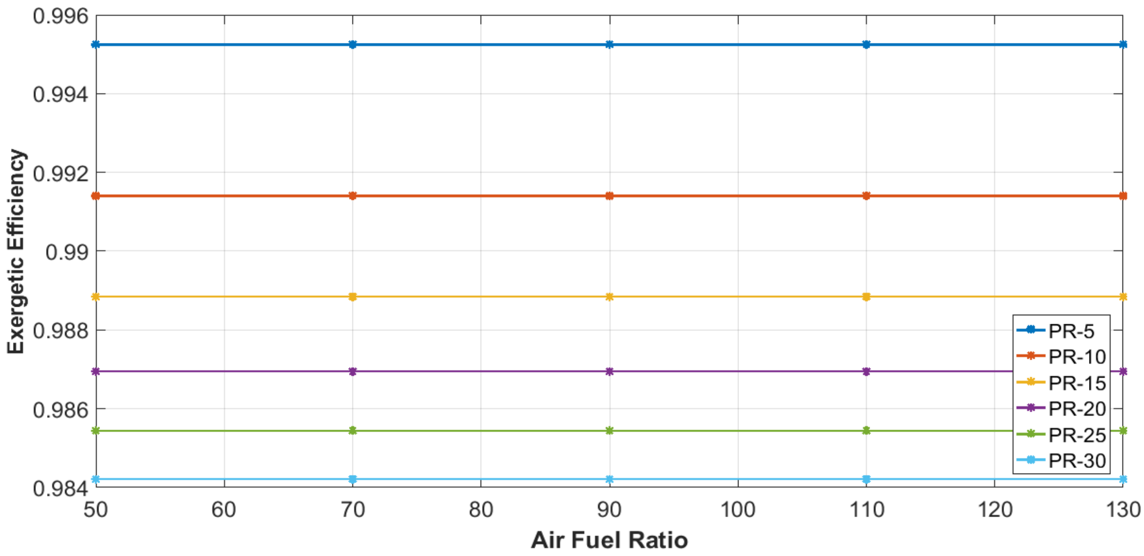



Figure 6.3: Exergetic efficiency of compressor at Various Pressure Ratios VS Air Fuel Ratio

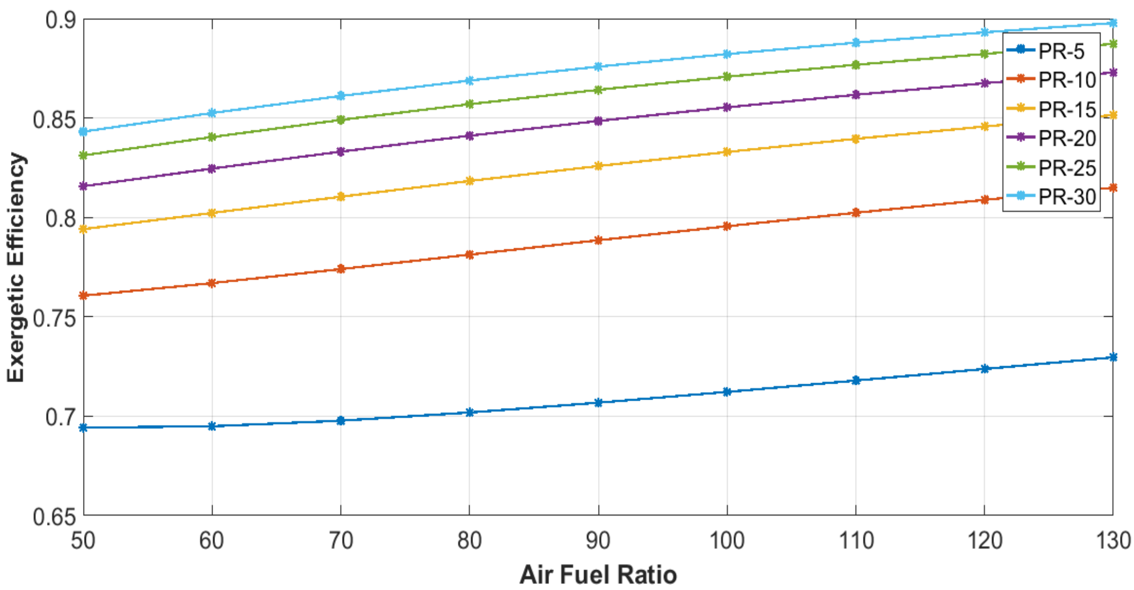



Figure 6.4: Exergetic efficiency of combustion chamber at Various Pressure Ratios VS Air Fuel Ratio

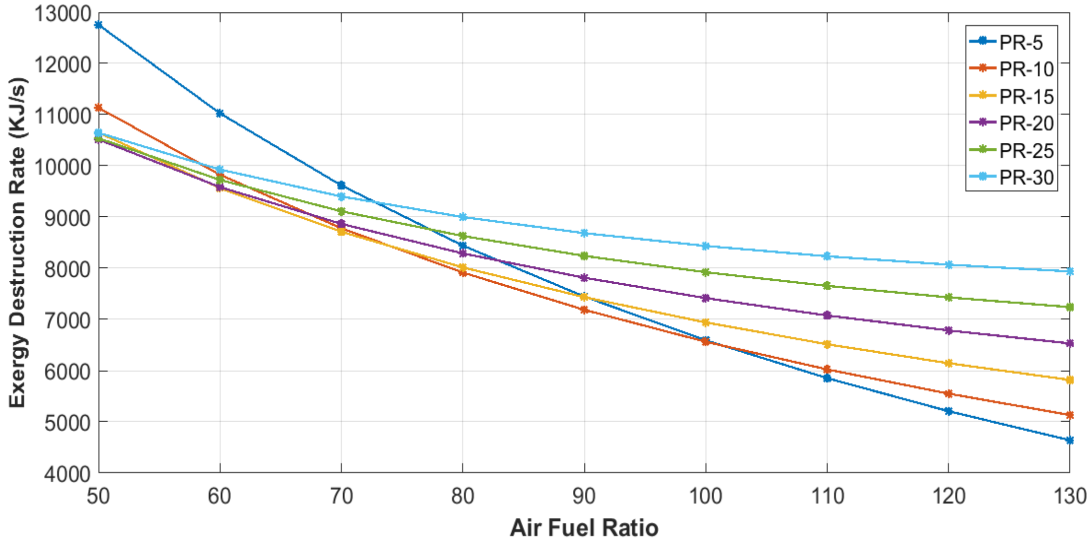



Figure 6.5: Exergy Destruction Rate of HRSG at Various Pressure Ratios VS Air Fuel Ratio

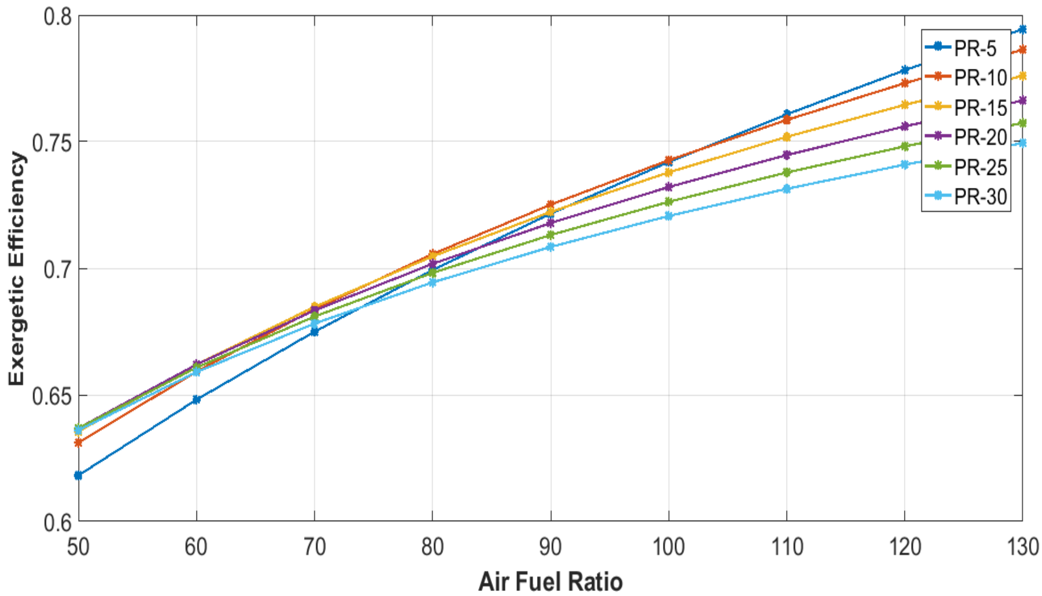



Figure 6.6: Exergetic efficiency of HRSG at Various Pressure Ratios VS Air Fuel Ratio

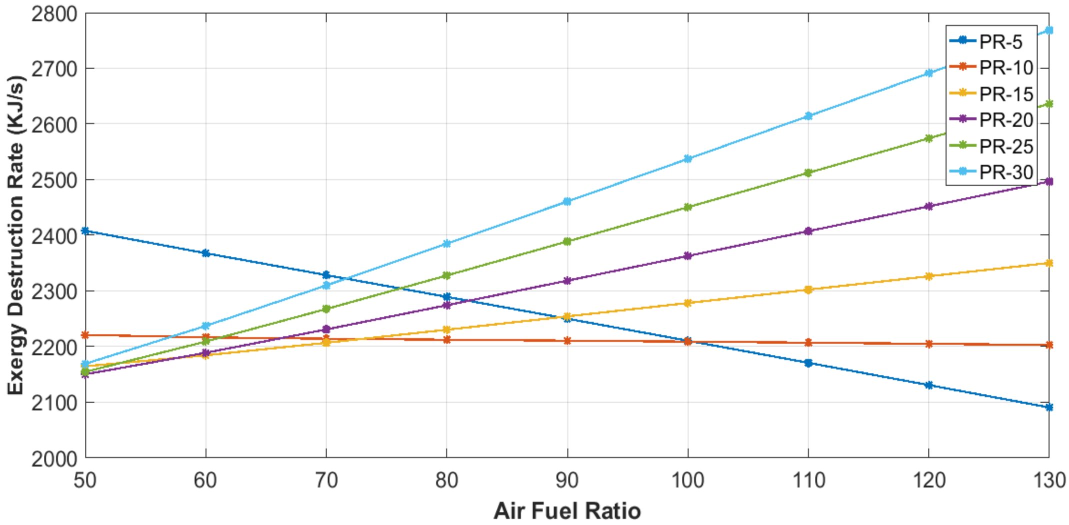



Figure 6.7: Exergy Destruction Rate of steam turbine at Various Pressure Ratios VS Air Fuel Ratio

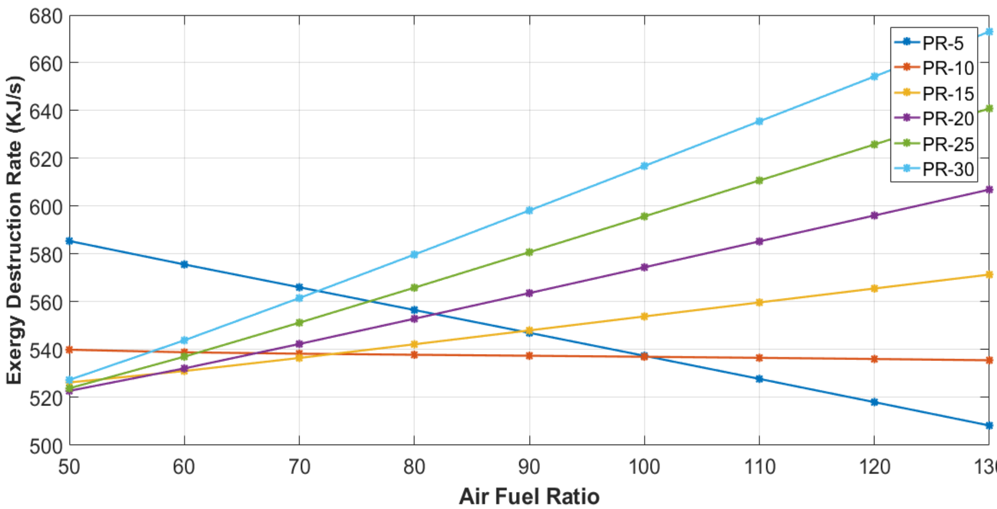



Figure 6.8: Exergy Destruction Rate of Condenser at Various Pressure ratio VS Air Fuel Ratio

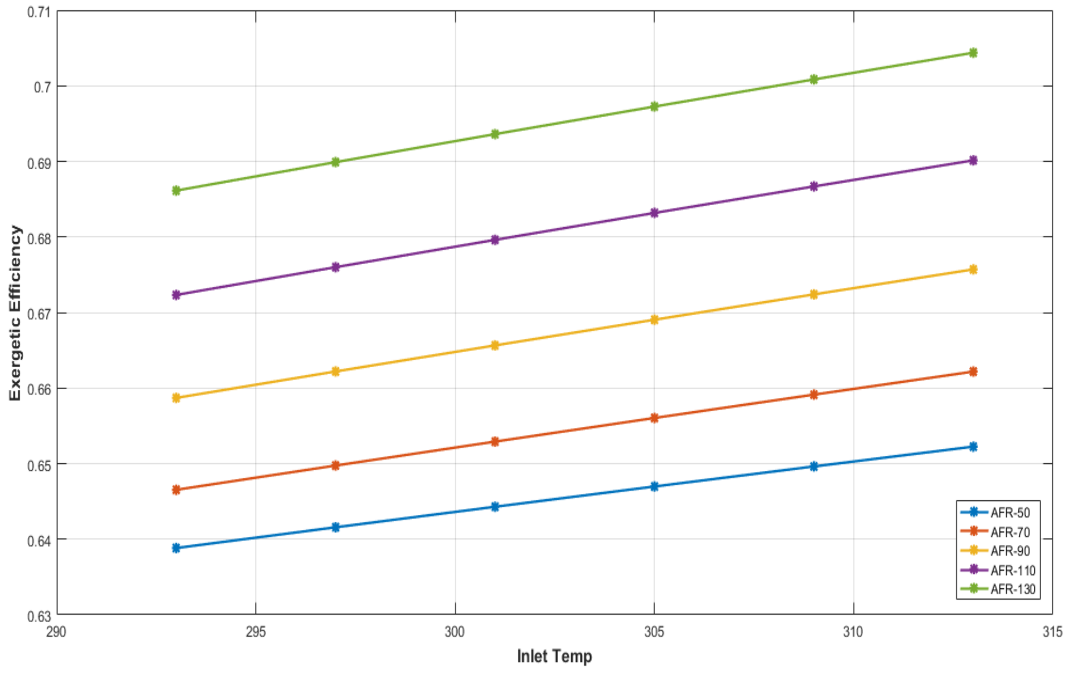



Fig- Variation of Exergetic Efficiency of Combustion Chamber VS Inlet Temp

Figure 6.9: Exergetic Efficiency of combustion Chamber at Various AFR VS Inlet Temperature

# CHAPTER 7

## 7. Conclusion

The point of the exergy examination is to distinguish the extents and the areas of genuine energy losses, keeping in mind the end goal to enhance the current systems, procedures or segments. A second law analysis of a thermal power plant has been performed in this project report, alongside a parametric review that considers the impacts of different parameters like inlet temperature and pressure on the system performance.

Combined cycle power plants take care of the developing energy demand, and consequently, unique consideration must be given to characterize a methodology for the advancement of these systems. Exergy analysis exhibited for a combined cycle power plant has given data on the irreversibilities of each process. While the most energy losses happen at the stack, the exergy investigation made for this complex plant demonstrated that the greatest exergy losses occur in the combustion chamber, gas turbine, and HRSG. Game plans proposed to expand the productivity of the plant are higher gas turbine inlet temperature utilizing uncommon super compounds in gas turbine hot parts, IAC (compressor inlet air cooling) and an advancement strategy for the HRSG working parameters.

The exergy analysis indicates that the HRSG has the highest influenced component on the overall system, and thus, this more significant component is taken into consideration. Therefore, this thesis focuses on the parameters in optimization and suggests that a decrease in exergy destruction (increases exergetic efficiency) of HRSG components can improve the system performance.

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# APPENDICES

## Appendix 1: Gas Turbine Parameters

| Parameters | Value |
|---|---|
| Ambient air pressure [bar] | 1.013 |
| Ambient air temperature [°C] | 293-313 |
| Turbine isentropic efficiency [%] | 90 |
| Compressor isentropic efficiency [%] | 88 |
| Compression ratio [-] | 10-20 |
| Air Fuel Ratio [-] | 50-130 |
| Lower heat value of the fuel [kJkg-1] | 43500 |
| Heat Capacity Ratio for air [$\gamma_a$] [49] | 1.4 |
| Heat Capacity Ratio for gas [$\gamma_g$] [49] | 1.33 |

## Appendix 2: HRSG Parameters

| PARAMETER | Value |
|---|---|
| The pinch point temperature difference[°C] | 13 |
| Pressure in drum of HRSG | 10 |
| Live steam temperature at the inlet of low pressure steam turbine [°C] | 535 |
| Minimum stack temperature [°C] | 150 |
| Feed water temperature at 9 [°C] | 55 |

## Appendix 3: Steam Turbine Parameters

| Parameters | Value |
|---|---|
| The isentropic efficiency of steam turbine | 90 |
| The isentropic efficiencies of water pumps | 82 |
| Minimum dryness fraction of steam at low steam turbine outlet | 0.88 |
| Condenser pressure [bar] | .07 |
| The inlet cooling water temperature in condenser [°C] | 20 |

# Appendix 4: Thermodynamic Properties of steam

| t, °C | 1.0 MPa ($t_s$ = 179.878 °C) | | | |
|---|---|---|---|---|
| | *v* | ρ | *h* | *s* |
| $t_s$(L) | 1.127 23 | 887.13 | 762.52 | 2.1381 |
| $t_s$(V) | 194.36 | 5.1450 | 2777.1 | 6.5850 |
| **0** | 0.999 70 | 1000.30 | 0.98 | -0.000 09 |
| **5** | 0.999 59 | 1000.41 | 22.01 | 0.076 24 |
| **10** | 0.999 87 | 1000.13 | 42.99 | 0.151 00 |
| **15** | 1.000 48 | 999.52 | 63.94 | 0.224 31 |
| **20** | 1.001 38 | 998.62 | 84.85 | 0.296 28 |
| **25** | 1.002 55 | 997.45 | 105.75 | 0.366 97 |
| **30** | 1.003 97 | 996.05 | 126.64 | 0.436 45 |
| **35** | 1.005 60 | 994.43 | 147.53 | 0.504 78 |
| **40** | 1.007 44 | 992.61 | 168.41 | 0.572 02 |
| **45** | 1.009 48 | 990.61 | 189.30 | 0.638 19 |
| **50** | 1.011 71 | 988.43 | 210.19 | 0.703 35 |
| **55** | 1.014 11 | 986.09 | 231.09 | 0.767 53 |
| **60** | 1.016 69 | 983.59 | 252.00 | 0.830 77 |
| **65** | 1.019 43 | 980.95 | 272.92 | 0.893 10 |
| **70** | 1.022 33 | 978.16 | 293.86 | 0.954 55 |
| **75** | 1.025 39 | 975.24 | 314.81 | 1.0152 |
| **80** | 1.028 60 | 972.19 | 335.77 | 1.0750 |
| **85** | 1.031 97 | 969.02 | 356.75 | 1.1340 |
| **90** | 1.035 50 | 965.72 | 377.76 | 1.1922 |
| **95** | 1.039 17 | 962.30 | 398.79 | 1.2497 |
| **100** | 1.043 00 | 958.77 | 419.84 | 1.3065 |
| **105** | 1.046 99 | 955.12 | 440.92 | 1.3626 |
| **110** | 1.051 12 | 951.36 | 462.04 | 1.4181 |
| **115** | 1.055 42 | 947.49 | 483.19 | 1.4729 |
| **120** | 1.059 87 | 943.51 | 504.38 | 1.5272 |
| **125** | 1.064 49 | 939.42 | 525.60 | 1.5808 |
| **130** | 1.069 27 | 935.21 | 546.88 | 1.6339 |
| **135** | 1.074 23 | 930.90 | 568.20 | 1.6865 |
| **140** | 1.079 35 | 926.48 | 589.58 | 1.7386 |
| **145** | 1.084 66 | 921.95 | 611.01 | 1.7901 |
| **150** | 1.090 15 | 917.31 | 632.50 | 1.8412 |
| **155** | 1.095 83 | 912.55 | 654.06 | 1.8919 |
| **160** | 1.101 71 | 907.68 | 675.70 | 1.9421 |
| **165** | 1.107 80 | 902.69 | 697.41 | 1.9919 |
| **170** | 1.114 10 | 897.58 | 719.20 | 2.0414 |
| **175** | 1.120 63 | 892.35 | 741.08 | 2.0905 |
| **180** | 194.44 | 5.1431 | 2777.4 | 6.5857 |
| **185** | 197.42 | 5.0653 | 2790.7 | 6.6148 |

| | | | | |
|---|---|---|---|---|
| **190** | 200.34 | 4.9916 | 2803.5 | 6.6427 |
| **195** | 203.20 | 4.9212 | 2816.0 | 6.6695 |
| **200** | 206.02 | 4.8539 | 2828.3 | 6.6955 |
| **210** | 211.56 | 4.7268 | 2852.2 | 6.7456 |
| **220** | 216.98 | 4.6087 | 2875.5 | 6.7934 |
| **230** | 222.31 | 4.4983 | 2898.4 | 6.8393 |
| **240** | 227.56 | 4.3944 | 2920.9 | 6.8836 |
| **250** | 232.75 | 4.2965 | 2943.1 | 6.9265 |
| **260** | 237.88 | 4.2038 | 2965.1 | 6.9681 |
| **270** | 242.96 | 4.1159 | 2986.9 | 7.0087 |
| **280** | 248.01 | 4.0322 | 3008.6 | 7.0482 |
| **290** | 253.01 | 3.9524 | 3030.2 | 7.0868 |

| $t$, °C | **1.0 MPa ($t_s$ = 179.878 °C)** | | | |
|---|---|---|---|---|
| | $v$ | ρ | $h$ | $s$ |
| **300** | 257.99 | 3.8762 | 3051.6 | 7.1246 |
| **310** | 262.94 | 3.8032 | 3073.0 | 7.1616 |
| **320** | 267.86 | 3.7333 | 3094.4 | 7.1979 |
| **330** | 272.76 | 3.6662 | 3115.7 | 7.2335 |
| **340** | 277.64 | 3.6018 | 3136.9 | 7.2685 |
| **350** | 282.50 | 3.5398 | 3158.2 | 7.3029 |
| **360** | 287.35 | 3.4801 | 3179.4 | 7.3367 |
| **370** | 292.18 | 3.4225 | 3200.7 | 7.3700 |
| **380** | 297.00 | 3.3670 | 3221.9 | 7.4028 |
| **390** | 301.81 | 3.3133 | 3243.2 | 7.4351 |
| **400** | 306.61 | 3.2615 | 3264.5 | 7.4669 |
| **410** | 311.39 | 3.2114 | 3285.8 | 7.4984 |
| **420** | 316.17 | 3.1629 | 3307.1 | 7.5294 |
| **430** | 320.94 | 3.1159 | 3328.5 | 7.5600 |
| **440** | 325.69 | 3.0704 | 3349.9 | 7.5902 |
| **450** | 330.45 | 3.0262 | 3371.3 | 7.6200 |
| **460** | 335.19 | 2.9834 | 3392.8 | 7.6495 |
| **470** | 339.93 | 2.9418 | 3414.3 | 7.6786 |
| **480** | 344.66 | 2.9014 | 3435.8 | 7.7075 |
| **490** | 349.39 | 2.8621 | 3457.4 | 7.7360 |
| **500** | 354.11 | 2.8240 | 3479.1 | 7.7641 |
| **520** | 363.54 | 2.7507 | 3522.6 | 7.8196 |
| **540** | 372.95 | 2.6813 | 3566.2 | 7.8740 |
| **560** | 382.35 | 2.6154 | 3610.1 | 7.9273 |
| **580** | 391.74 | 2.5527 | 3654.2 | 7.9796 |
| **600** | 401.11 | 2.4931 | 3698.6 | 8.0310 |
| **620** | 410.47 | 2.4362 | 3743.2 | 8.0815 |
| **640** | 419.82 | 2.3820 | 3788.0 | 8.1312 |
| **660** | 429.16 | 2.3301 | 3833.1 | 8.1800 |
| **680** | 438.50 | 2.2805 | 3878.5 | 8.2281 |

| | | | | |
|---|---|---|---|---|
| **700** | 447.83 | 2.2330 | 3924.1 | 8.2755 |
| **720** | 457.15 | 2.1875 | 3970.0 | 8.3221 |
| **740** | 466.47 | 2.1438 | 4016.1 | 8.3681 |
| **760** | 475.78 | 2.1018 | 4062.5 | 8.4135 |
| **780** | 485.08 | 2.0615 | 4109.2 | 8.4582 |
| **800** | 494.38 | 2.0227 | 4156.1 | 8.5024 |
| **820** | 503.68 | 1.9854 | 4203.3 | 8.5460 |
| **840** | 512.97 | 1.9494 | 4250.8 | 8.5890 |
| **860** | 522.26 | 1.9147 | 4298.5 | 8.6315 |
| **880** | 531.55 | 1.8813 | 4346.5 | 8.6735 |
| **900** | 540.83 | 1.8490 | 4394.8 | 8.7150 |
| **920** | 550.11 | 1.8178 | 4443.3 | 8.7560 |
| **940** | 559.39 | 1.7877 | 4492.1 | 8.7965 |
| **960** | 568.67 | 1.7585 | 4541.1 | 8.8366 |
| **980** | 577.94 | 1.7303 | 4590.4 | 8.8763 |
| **1000** | 587.21 | 1.7030 | 4639.9 | 8.9155 |
| **1100** | 633.54 | 1.5784 | 4891.4 | 9.1056 |
| **1200** | 679.83 | 1.4710 | 5148.9 | 9.2866 |
| **1300** | 726.10 | 1.3772 | 5411.9 | 9.4593 |
| **1400** | 772.34 | 1.2948 | 5680.0 | 9.6245 |
| **1500** | 818.57 | 1.2216 | 5953.0 | 9.7830 |
| **1600** | 864.78 | 1.1564 | 6230.3 | 9.9351 |
| **1800** | 957.19 | 1.0447 | 6796.7 | 10.222 |
| **2000** | 1049.6 | 0.952 78 | 7376.8 | 10.489 |

# Appendix 4: Water Properties

| $p$,MPa | $t$, ℃ | Density,kg/m3 | | Enthalpy,kJ/kg | | Entropy,kJ/(kg·K) | | Volume,cm$^3$/gm | |
|---|---|---|---|---|---|---|---|---|---|
| | | $\rho_L$ | $\rho_v$ | $h_L$ | $h_v$ | $s_L$ | $s_v$ | $v_L$ | $v_v$ |
| **0.0070** | 39.00 | 992.55 | 0.048722 | 163.35 | 2571.7 | 0.55903 | 8.2745 | 1.00750 | 20524. |