# Exergetic Evaluation of double HRSG 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**

**THERMAL ENGINEERING**

*Submitted by:*

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**(2K17/THE/06)**

**SESSION: 2017-2019**

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# DECLARATION

I hereby declare that the work which being presented in the major thesis entitled "**<u>Exergetic Evaluation of double HRSG Combined cycle Gas Power plant</u>**" in the partial fulfillment 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 **Prof. 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. I confirm that I have read and understood the 'Plagiarism policy of DTU'. I have not committed plagiarism while completing the attached piece of work, similarity found after checking is 5% which is below the permitted limit of 20%.

Place: Delhi **NEERAJ KUMAR**

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# CERTIFICATE

This is to certify that NEERAJ KUMAR, (Roll no. 2K17/THE/06), student of M.Tech., THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled "**Exergetic Evaluation of double HRSG 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.

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# ACKNOWLEDGMENT

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 fulfillment of this thesis project possible.

I take great pride in expressing my unfeigned appreciation and gratitude to my learned mentor **Prof. B. B. ARORA** and Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering), for their invaluable inspiration, guidance and continuous encouragement throughout this project work. Their critics and suggestions on my work have always guided me towards perfection. This work is simply the reflection of their thoughts, ideas, concepts and above all his efforts. Working under their guidance has been a privilege and an excellent learning experience that I will cherish for a long time. Lastly, I would like to thank Ashutosh Mishra sir, Ranjana and Lubna who helped me a lot.

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# ABSTRACT

The increasing demand for power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on the 1st law of thermodynamics only. The really useful energy loss cannot be justified by the 1st 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 dest. and exergy losses) and increasing exergetic efficiency. In this analysis shows exergy efficiency is less at each and every point of unit equipment. Also presents major losses of available energy at combustor, HRSG and gas turbine. The primary objectives of this work are to analyze the system components separately and to identify and quantify the sites having the largest energy and exergy losses at different load.

Program code is established using EES software to perform the calculations required for the exergy plant analysis considering real variation ranges of the main operating parameters such as pressure ratio, AFR, and inlet temp. The effects of these 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

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# NOMENCLATURE

| | | |
|---|---|---|
| Cp | capacity heat 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 |
| $mf$ | 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 |
| η | efficiency | - |
| $\rho$ | density | $kg/m^3$ |

## Subscripts

| | |
|---|---|
| CCGT | Combined cycle gas turbine |
| CV | Control Volume |
| net | Net |
| PH | Physical |
| 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 |

| | |
|---|---|
| CPPC | Combined power plant cycle |
| SPP | Steam power plant |
| GPPC | Gas power plant cycle |
| CPP | combined power plant |
| GTPP | Gas turbine power plant |
| STPP | Steam turbine power plant |

# CHAPTER 1

## 1. Introduction

Energy systems contain an extremely large no. and some coordinated efforts with the outside world. The engineers should address the issues which are wide particular energy, economy. The CPPs cycle have got an important consideration due to their energy efficiency, waste that why lower poisonous, releases ozone-depleting substances and operational flexibility. A combined cycle is comprised of a GPPC (topping cycle) and a SPPC (bottoming cycle) joined together through a HRSG.

For streamline, the efficiency capital attainability and ecological effect of alike plants, it's condemnatory to judge the region, type and authentic areas of wasteful aspects (irreversibility). For such a study, an exergy investigation is a helpful appliance.

Furtherance of energy change configuration turns out to be more essential due to obstruction of non-renewable energy sources and the effect on the ecology in the middle of their utilization wherever it uses in variety from warming and cooling to atomic power plants, the utilization of energy is discovered. The response to the persistently enhancing requirement for electric-era for a significant span of time was to manufacture another SPPC, no one absolutely differentiable from the last one.

The challenges which are faced by today's energy transformation engineer: changing approach, changing the social and technological climate in which different methodologies are possibly going to be acknowledged.

Some unfavorable qualities of modern power schemes are lower capitals and working costs, the capacity to work with different types of problems of energizes, there great resistance to the fluctuation of fuel, short enhancement time, low consequences of toxins, great efficiency, maintainability, and dependability.

The confined presence of natural resources is also being a major problem faced by the energy conversion engineer. To substitute the flammable gas and oil for mechanical help stocks other future requirements power plant need to use coal and nuclear energy. The other important problem arises regarding the use of a constant advantage. A part of immediately attainable coal has a high amount of sulfur which damages the earth when discharged from power plants in unprocessed items.

The remarkable problem of corrosive rains has been accredited to release from the coal-oriented power plant. Condemnatory developments inefficiency can be achieved by merging the regular cycle in proper ways, A like power plants system is suggested to as CPPC. It is clear from the examination of the Rankine cycle and Brayton cycle that the discharge of a great amount of thermal energy from the surroundings goes with production to power. This rejection of heat cannot be ejected of, but it may be decremented by developing the thermal efficiency of cycle.

Traditional thermodynamics gives objective of energy transfer by work and heat, energy balance entropy, entropy modification and approximation of thermodynamic balance. $2^{nd}$ law of thermodynamics develops an energy balances establishing the authentic thermodynamics approximation of energy balances and real thermodynamics inefficient and loss from the method and substructure. There are two conditions upon which a thermodynamic property depends i.e condition of the barrier and the condition of nature. It meets the greatest ability of energy balance to lead out the change. Exergy is dynamically recognized with a financial approximation of the balance considering that client pay the potentially when expenses are parallel out to energy balance, exergy must fill in as undertaking in the costing method.

Generally, first law examination leads just energy utilization the concerning conservation of energy. Both individually and quantitatively, it can not give the data with respect to the losses and can not figure out the extent of the losses. These conditions force us to bring about exergy analysis in perspective of $2^{nd}$ law of thermodynamic. Exergy are not preserved properly yet preferably some of it is destroyed in the real procedure. This examination signifies the statics with respect to losses that involve their area individually and quantitively. This sticks can used for more change in profile and process of the substructure. The system execution can be developed by developing the energetic capability of the component and system by deleting the exergy demolition.

## 1.1The Objective of the Thesis

The extension and motivation behind this exploration is to create successful strategy to accomplish exergetic optimizations of CPPs.Consequently, the motive of this research is to enhance the achievement of the power plant in a way to suggest an exergy optimization method.It would be feasible with the help of this method:

1. Tell about the exergy demolition and exergy losses alongside their position.
2. Estimate the highest exergy destructor component of the system.
3. Demonstrate methods of enhancing the energetic efficiency.
4. Discover the optimum practical values of working variables, which gives the maximal feasible power output and efficiency.

Furthermore, it would be feasible to compute minimal feasible exergy demolition.

## 1.2 Layout of the thesis:

Thesis is separated into 7 chapters. The ordinary idea of exegetic evaluation is instigated in this chapter. The significance of the optimization is concisely considered.

Chapter 2: presents a considerable analysis of literatures awning topics related to this research, about of exergy analysis and their optimization.

Chapter 3: an outline over the CPP system. The combined cycle thermodynamic and its components are given the principal prominence.

Chapter 4: gives out the conceptual framework of energy, exergy examination with the expressions.

Chapter 5: constitutes the element-wise mathematical simulation of a CPP system.

Chapter 6: gives out the outcome of every move of the technique. The consequences of the working variables on the execution variables are also considered.

Chapter 7: deduces the results of the study.

# CHAPTER 2

## 2. Literature Survey

In order to have a thought of the present technological advancement in the area of execution and improvement of combined cycle GTPP, a short review of accessible writing was made.

In any case, this section is concerned about a survey of the writing on optimization performed on various thermal systems. As a rule, a few authors center around the gas turbine operating parameters (topping cycle), others optimize the Steam plant (bottoming cycle) based on a given gas turbine, while others propose suitable optimization techniques for the whole combined cycle power plant.

Moreover, the optimization can be dissected from a thermodynamic perspective, as per the first and additionally $2^{nd}$ law analysis, or utilizing a thermoeconomic or environmental-economic strategy (Kaviri et al [1], Ahmadi and Dinner [2], Boyano et al [3] and Petrakopoulou et al [4]).From the perspective of optimization technique, there are numerous kinds of analysis. In this work, the survey will feature the most basic technique: the exergy dest. 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 impact the combined cycle gas turbine performance are; ambient conditions, Compressor pressure ratio, and turbine inlet temp.

### 2.1.1.1 The Effect of Ambient Conditions

One of the factors that influence gas turbine performance is the ambient conditions, mainly ambient temp., atmospheric pressure, and the relative humidity of the air. These parameters influence the generated electric power and the heat-rate during operation. The location of the power plant assumes a noteworthy job on its performance. The atmospheric air, which enters the Compressor, becomes hotter after compression and it is coordinated to a combustion Chamber. Several authors wrote the effect of ambient temp.: Ibrahim et al [5], Ameri and Hejazi [6], Boonnasa et al [7] and Hosseini et al [8].

**Ameri and Hejazi [6]** seen that the variation in the ambient temp. causes lost 20% of the rated capacity of the 170 gas turbine units in Iran. They considered five gas turbines, where the difference between the ambient temp. and the ISO conditions was average at 11.8 °C. They found that for every 1 °C increment in ambient temp., the power output was diminished by 0.74%, and they proposed cooling the Compressor's intake-air temp. to improve the gas turbine cycle efficiency.

**Hosseini et al** [8] shown that the gas turbine compressor is intended for consistent air volume flow, which makes the electric power output reliant on the ambient temp. through the particular mass flow rate. They included that the increase in the ambient temp. additionally diminishes the compressor's output pressure, which lessens the gas turbine cycle efficiency, while the increment noticeable all-around thickness decreases the gas turbine's heat rate and builds its particular fuel consumption. They expressed that for each 1°C increment in the ambient air temp., the electric power output of the gas turbine diminishes 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 the combustion Chamber depending on the Compressor pressure ratio as reported by Ibrahim et al [5], Ibrahim and Rahman [9], and Khaliq and Kaushik [10].

**Ibrahim and Rahman [9]** showed the parametric thermodynamic analysis of a combined cycle gas turbine. They researched the impact of operating parameters, compression ratio, gas-turbine peak temp. ratio, isentropic Compressor and efficiency and AFR on the overall plant performance. Their outcomes demonstrate that the compression ratios, air to fuel ratio as well as the isentropic efficiencies are unequivocally impacted by the overall thermal efficiency of the combined cycle GTPP. The overall thermal efficiency increases with compression ratio as well as isentropic Compressor and turbine efficiency. Be that as it may, the variety of by an overall thermal efficiency is minor at the lower compression ratio while it is noteworthy at the higher compression ratio for both the isentropic Compressor and turbine efficiency. The overall efficiencies for the combined cycle gas turbine are a lot higher than the efficiencies of gas turbine plants. Efficiency cited range is about 61%. Moreover, the overall thermal efficiency increases and total power output diminishes directly with the increase of the Compressor ratio with constant turbine inlet temp.. The peak overall efficiency occurs at the higher compression ratio with the higher cycle peak temp. ratio as well as higher isentropic Compressor and turbine efficiencies.

#### 2.1.1.3 Effect of Turbine Inlet Temperature

The turbine inlet temp. (TIT) plays a major role in the performance of the 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] expressed that the parameter that influences cycle performance most is the turbine inlet temp. TIT. The TIT ought to be kept on the higher side, as at lower values, the exergy dest. 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 expanding the live Steam and reheat-Steam parameters, and by introducing high-efficiency, low-loss turbine blade geometries. The main objective, to increase the Steam parameters, is fundamentally accomplished by picking proper materials for the components working under live-Steam and reheat-Steam conditions while retaining the proved designs. Collaborative European projects have prompted the advancement and capability of steels with much-improved creep properties at temp. of up to 600 °C, suitable for the production of key components. In the meantime, optimization of the blade profiles and geometries permitted further significant upgrades in operating efficiency. The reachable enhancements in efficiency are about 0.5% per 10 °C live Steam and reheat (RH) temp. increase and 0.2 % per 10 bar pressure increase. The second significant piece of the bottoming cycle is the heat recovery Steam turbine (HRSG), its design and optimization influences to an enormous degree impact the efficiency and the expense of the entire 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 furnaceboiler 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 ofincreasing 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 Combined Power Plant System

### 3.1 Introduction

The CPP cycle of efficiency was around 5–6% higher than compared with the Steam plant. In midst of the 1960s, the usage of heat recovery sort of CPPC system ended up being increasingly normal. Its basic applications were in power plant and heat application where its ability to heat measure was progressively positive. What's more, few of the heat recovery CPPCs were presented in use power age application in the midst of the 1960s. The utilization of this frameworks during the 1980s set up the heat recovery continue water warming CPPC as created innovation as base load and mid go advantage. [31].

By 1970s, there was a different power plant in the task. All through the 1980s, the advancement made with greater GTPP and the commencement of pre-blended ignition for lower NO2 releases. Around 1990s the plant efficiency of CPPCs passed half. All through 1990, incalculable power plants cycle were gathered and an extensive parcel about the base - load activity. Around 1995s, another time of costly gas turbine went to market, bearing the piece of CPP respect 350-400 MW and capability up to 56-58%. Starting in 2011s, the generation of power was expanded up to 570 MW and effectiveness near 61% [32].

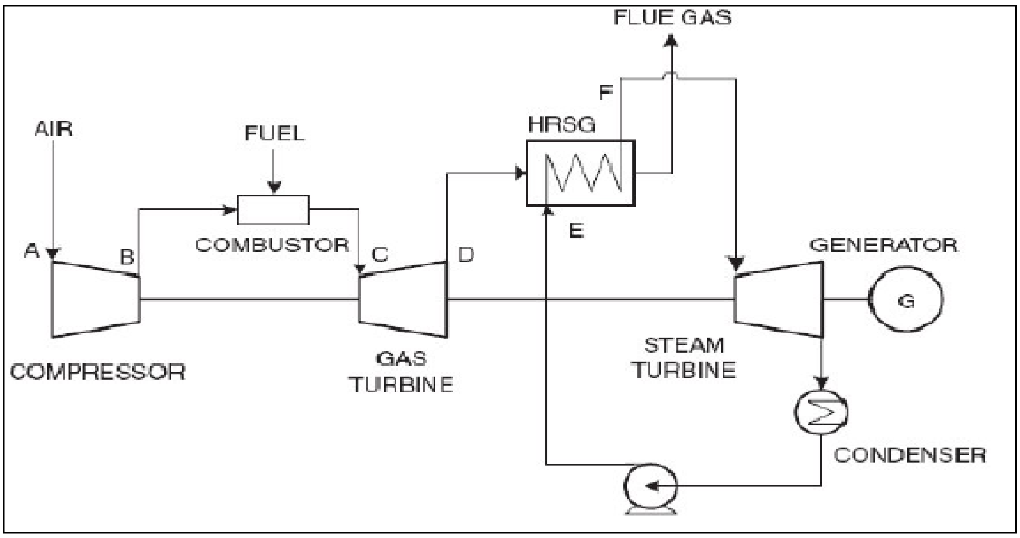


*Fig3.1.Combined Cycle Power Plant* [33]

The HRSG gets approx. 66% of heat of gas turbine exhaust with the other approx. 33% of heat lost in the leaves stack. Lastly, 19% of the fuels data are changed over toward power by methods for this STPP with 1% lost in the turbine and fuel energy of 21% waste in the Steam, which is entered to condenser. The combined total intensity of gas turbine and Steam turbine thinks up to 55% (LHV) of fuel energy. The plant collaborator records for approximation 2% of fuel contribution at the end leaving 53% as total CPPC productivity. As such, the central barrier for applying HRSGs inside service of power plants lays a free bit of leeway from the together of the gas turbine (Brayton cycle) of Steam turbine (Rankine cycle) which achieves an enhanced generally thermal efficiency. [34]

## 3.2 Component of Combined Power Plant Cycle

### 3.2.1 Compressor

The Compressor is an important component in the GTPP in which work is done on gas to increase the gas pressure by a decrease of volume. In the compression process temp. of air increase and this increased air; temp. Air goes to combustor for further process. The GTPP, axial Compressor is used. A selection of Compressor in a Compressor depends on amount of compressed air is used to run the gas turbine. Mostly axial Compressor or centrifugal Compressor is used. Centrifugal Compressor is a radial flow type machine in which air goes through the eye of the impeller in the axial Compressor and moves through the impeller in radially outward direction. In the inlet casing, there is a small pressure drop in pressure along with an increase in absolute velocity while flowing through the impeller, energy is transferred due to which static pressure increase along with increases in velocity head. High kinetic energy is obtained in the impeller is changed into pressure energy in the diffuser.

The centrifugal effect in the centrifugal Compressor of the curved blade creates a bending moment and develops high stress which may change the blade profile. Thus for impeller running at high rpm radial blade is preferred. Forward curve blades in centrifugal Compressor give max power; develop the highest pressure ratio but poor efficiency and narrow operating range. Backward curved blades in centrifugal Compressor give the best efficiency and have a wide operating range.

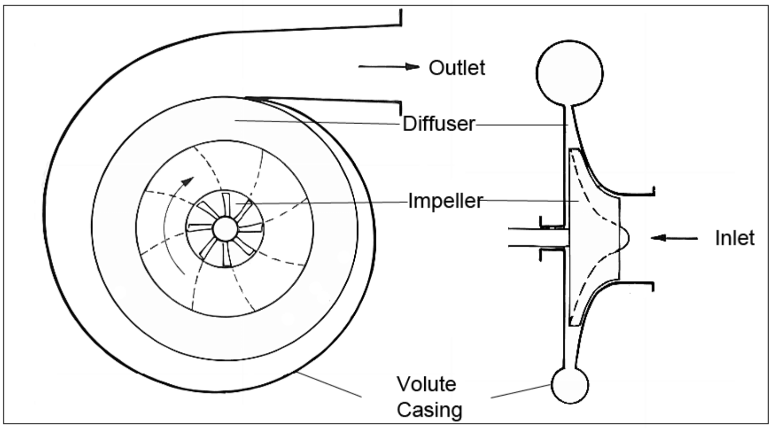


*Fig.3.2. Centrifugal Compressor* [35]

In Axial air Compressor, air flow in direction parallel as the axis of rotation and it consists of number of stage, every stage is comprises of a set of the fixed and moving blade. The blade cross-sectional passage is made diverging type both for fixed and moving. Energy is transferred in the moving blade which increases the kinetic energy along with the static pressure due to reaction effect. The velocity obtained in the moving blade is transferred into pressure in the fixed blade due to impulse effect.

In the axial Compressor, pressure ratio per stage is very less (1.2 to 1.4) compared to the centrifugal Compressor (4 to 6) but overall is more pressure ratio in an axial gas Compressor. The number of the blade in the axial gas Compressor is decreased from inlet to outlet as the pressure in order to get constant axis flow velocity (to neutralize axial thrust).

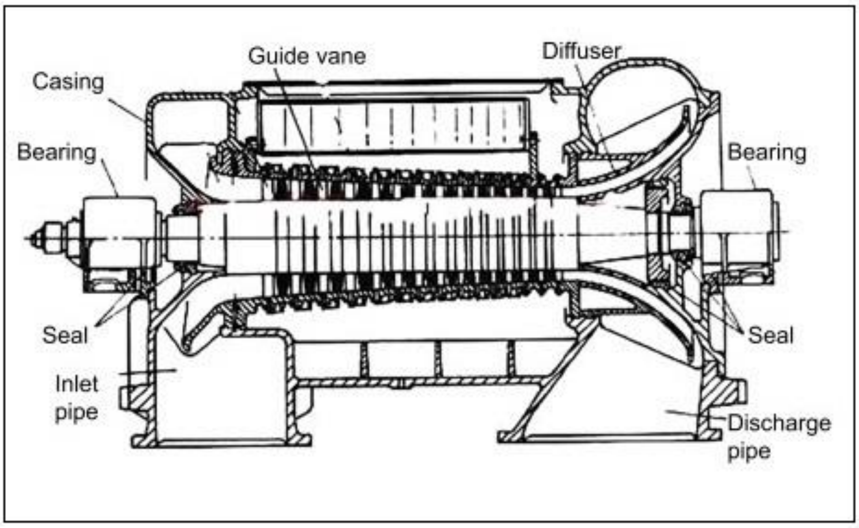


*Fig.3.3. Axial Compressor* [36]

## 3.2.2 Combustor

The combustor is an important component in gas power plant in which fuel is supply along with compressed air in order to get the desirable temp. at outlet of combustor. The Combustion Chamber in GTPP is known as a combustor. There are two important functions of combustor in GTPP (i), the Combustion Chamber change from chemical energy into the thermal energy  of fuel for proper expands in the turbine. (ii) Combustion Chamber gives the temp. contour of hot gases at outlet of the system to avoid the fabric problem. The temp. of hot gases is not very to damage the surface of the turbine blade, it is restricted to 1300$^{o}$C. Air is coming from the Compressor at a very high velocity so it is important to decrease the velocity of compressed air in pre-diffuser.In combustor, there is loss of energy so that take combustor efficiency approx 90%.combustor is mainly divided into three part: primary zone, intermediate and dilution zone. For proper burning of fuel, burners are placed on the periphery of the Combustion Chamber, a number of the burner is placed based on the capacity of power plant.combustor is made of a special type of alloy material and proper refractory is apply to converse heat within the combustor. These hot gases are supplied on gas turbine blades, the kinetic of hot gases are used to generate a rotating effect. In normal conditions, the total air-fuel rate in the combustor can change in the period of 60:1  and 130:1.the effective way to consume the air-fuel rate to 15:1.

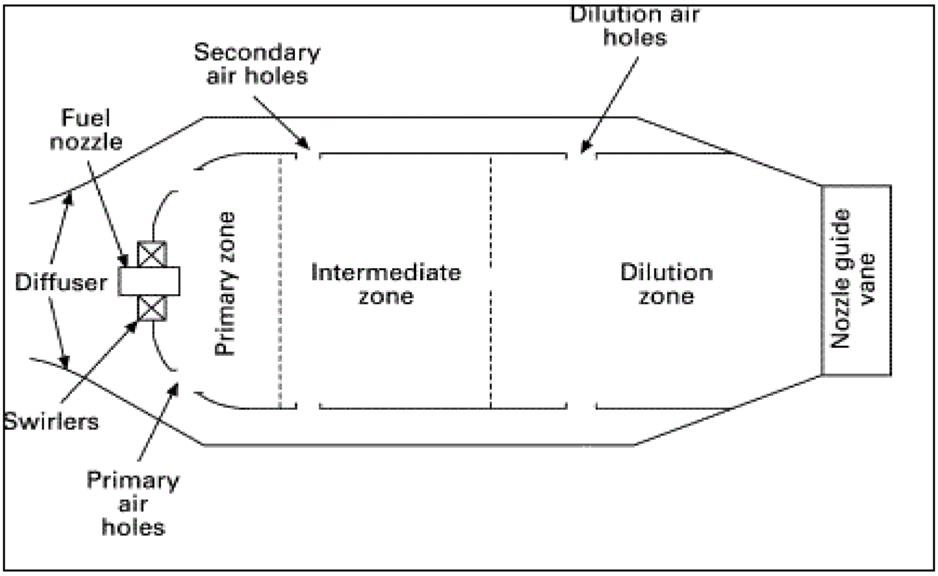


*Fig.3.4.Combustion Chamber* [37]

### 3.2.3 Gas Turbine

Gas turbine is an essential component of the power plant that produces electricity. Gases coming from Combustion Chamber enter into the turbine, gases expand in gas turbine i.e decrease of pressure and enthalpy.Gases supply in the turbine at very high kinetic energy, this kinetic energy rotates the turbine at very high speed. Gas turbine converted the energy of gases into mechanical energy. The shaft of gas turbine is connected with generator, the generator shaft rotates and electricity produced. The inlet temp. of gas turbine depends on metallurgy of turbine blade. Normally inlet temp. of gas turbine is nearly 1300°C. Some part of gas turbine work is utilized to run Compressor. Power generating gas turbine is a normally work on open cycle. Gas power plant working reply on working of its component Compressor, turbine and combustor.[38]

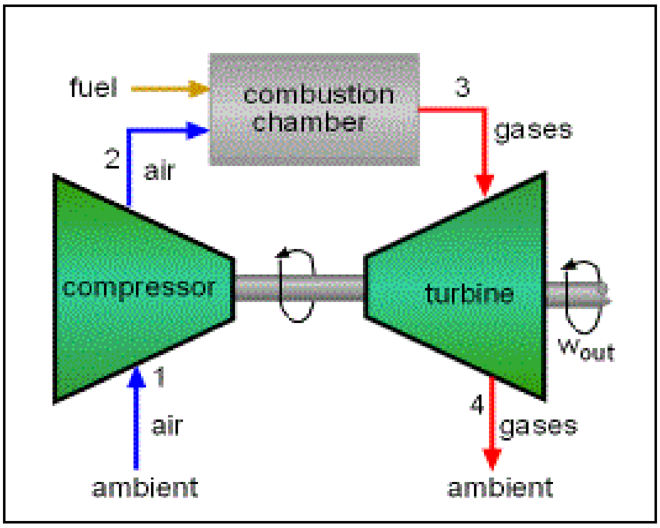


*Fig.3.5. Schematic diagram of the Brayton cycle* [38]

Brayton cycle is thermodynamic cycle of a GTPP . Gas turbine normally works on open cycle. Air is entered in Compressor, the air is compressed up to $P_2$.Both pressure and temp. increase by supplying work input($_{Wcomp}$) from turbine work on Compressor. The high pressure air enters into combustor, where fuel is supplied and burning of fuel at constant pressure. this supplied heat to increase the temp. from $T_2$ to $T_3$ i.e inlet of the gas turbine(maximum cycle temp.)this maximum temp. gas goes to the turbine, then they expand from temp. $T_3$ to $T_4$ and produce power ($W_{turbine}$). some produced power is used to run the Compressor. Heat $Q_L$ is transferred to the ambient.[39]The gases is leaving from the turbine are thrown off of the environment. The compression ratio, AFR, and Compressor inlet temp. are a very important parameter.

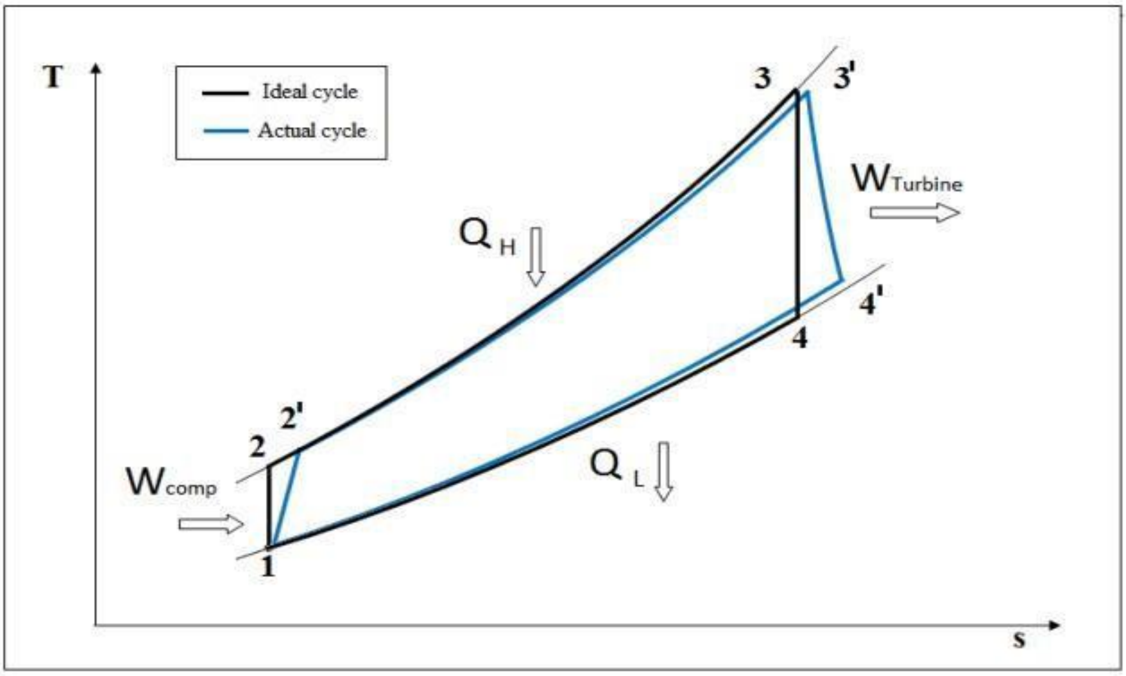


*Fig.3.6. Brayton cycle* [39]

## 3.2.4 Steam Turbine

Steam turbine is heart of the power generating system in which the heat energy of steam converts into mechanical energy. In large power generating power plant, three types of the turbine are used first, high pressure turbine second, an intermediate-pressure turbine third, low pressure turbine. Steam coming from boiler drum in high superheated form enter in high pressure turbine then steam goes to reheater, from reheater goes to intermediate pressure turbine, after that enter to low-pressure turbine.For Steam turbine, steam permitted to expand through number of stage course of action of cutting edges,, for achieving considerably higher efficiencies appeared differently in relation to a lone walk expansion[40]. The steam develops through dynamic set of moving cutting edges on a pole and set of fixed blade of the casing. The propelled Steam turbine has three stages. The high pressure portion has small blade i.e. little in light of the fact that the moving toward steam at high temp. has high energy. After steam experiences the high pressure zone, then goes for reheater for reheating of steam after that steam enters into another area of the turbine, known the intermediate-pressure turbine. The blade of the turbine here is greater than those in the high-pressure segment. In the wake of experiencing this portion, the steam is goes to low-pressure region of steam turbine. Steam moved out from the turbine the base, where it is merged by and by into the water. Then steam from turbine is return to boiler, for made into steam for other supply.[39]

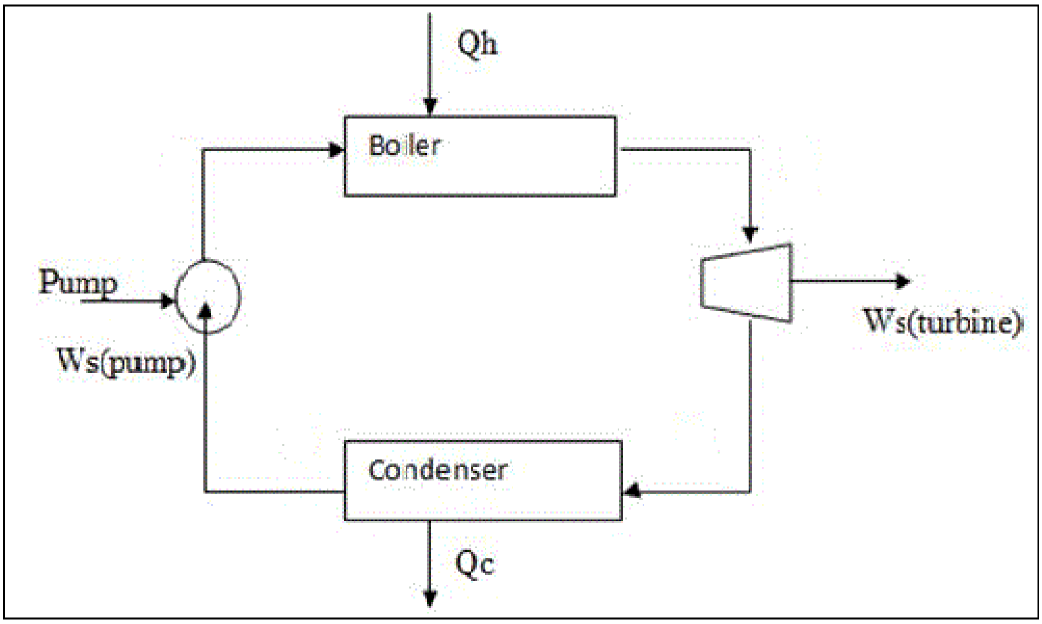


*Fig.3.7. Steam Power Plant Cycle*

Thermal efficiencies of the present-day STPP more than 120MW are like 39% to 41% [41]. The steam power cycles, working fluid is water. The water works over wide extent of temp. and pressure have a huge amount of heat limit and it is consistent, safe, and naturally cordial.

Small turbine and coupled generator limit changes to 7500 KW from 500 KW, while generous turbo-alternators having the farthest contrasting to 90MW from 10MW. Units enormous size, confines up to 500MW. Delivering 200MW units cutoff are winding up doubtlessly extremely ordinary. The Steam turbines usage depends on steam pressure and inlet temp., further deleting stages pressure. The enormous Steam turbines, steam usage in around 3.5 to 5kg for each kWh [42].

Mechanical losses in following generator bearing and losses oil pump. Losses from generator join mechanical and electrical losses. Weakened loss consolidates dynamic energy from stream while leaving at last stage and from exit of the last stage, the pressure drop, to the arranged Condenser. For the proper working of a Steam turbine, it supplies steam at predictable temp. and pressure. By technique for security value fixed on the boiler, steam weight can be overseen easily.

Rankine cycle is most commonly used vapor power cycle. Regardless of way that a depiction of this cycle can be seen in any structure thermodynamics course book, immediately verified here. Pumping the working fluid to high-pressure and hovered into heater. In the boiler liquid rises at predictable pressure and vapor made is stretched out through the turbine, as such separating work. The turbine vapor is consolidated in the Condenser, expelling heat into cold liquid. Couple of change made in Rankine cycle are used for achieving better efficiencies. This consolidates superheating, heating, and recovery.

For vapour power plant, the ideal cycle results from Rankine cycle .Ideal Rankine cycle exclude inside irreversibility and includes the going with following 4 methods : isentropic compression in pump, reliable pressure warm extension in the boiler, isentropic expansion in turbine and relentless pressure warm expulsion in the Condenser. These segments of rankine cycle are faithful state suffering steam gadgets. Disregarding the effects of potential and dynamic energy. The evaporator and Condenser excludes any work, pump and turbine are isentropic [40]

### 3.2.5 Condenser

The Condenser is used to condense the exhaust steam coming out of turbine by rejecting vaporized heat with the help of cooling water. After steam out of turbine, it goes to Condenser and until steam is converted into liquid water, heat is ejected. This is achieved by flow of steam around a large number of tubes.

The cold water is supplied typically from close by sea, lake, conduit as from a cooling tower. The dense steam is accumulated at the base of the Condenser, which returns to the heater with the help of water pumps to begin the cycle (water to steam, steam to water) again.

Condensate temp. chooses either pressure in the steam or its condensate side. This pressure, it calls as turbine back pressure, which is normally a vacuum. By reducing the condensate temp., turbine back pressure is brought down thus extending turbine power output.

Other functions of the Condenser are:

- The condensate is assembled in its hot well, through which condensate pump takes suction.
- Provides storing of condensate.
- Provides a low pressure get together point of condensate directing various frameworks into the plant.
- Provides the de-air course from accumulated condensate. Enormous power plant condenses are used for part of shell and cylinder heat exchangers.

Classifications of Condenser types are:

- Single pressure or multi pressure, depends on cooling water steam path makes at any the rate one turbine back pressure.
- The number of shells, dependent on low pressure turbine lodgings.
- Either single pass or two pass dependent on amount of parallel water steam routing through each of the shell.

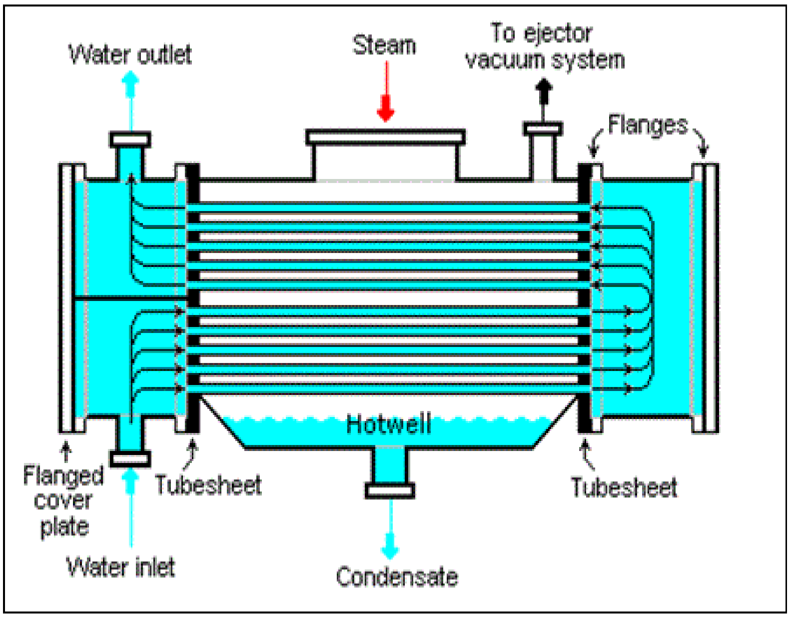


*Fig.3.8. Condenser in Steam power cycle* [40]

### 3.2.6 Water Pump

Water pump controls the proportion of water encouraged to boiler. It has a huge job in the task of boiler. In water pump, impeller, two piece twin volute of stainless steel. Multistage radial is most basic pump of power plants to sustain water to boiler. A generous round and hollow molded steel barrel is vital part for the phases to stack and shut down toward one side by dash on the head. The stages inside barrel casing are part radially. In few pumps, the stages are likewise part axially.

### 3.2.7 Heat Recovery Steam Generator

In the present development, CPPC is the most productive energy transformation system to convey thermal and electrical energy. In a regular CPPC, to make steam into the steam cycle, the gas turbine exhaust heat is recovered inside HRSG. The HRSG execution influences general execution of the CPPC. With various pressure levels, the steam created in HRSG relies on design. The HRSG comprises of 3 heat exchanger bundle i.e. economizer, evaporator and superheater. Combustion gases to superheater, evaporator and economizer respectively.

There is three-ways to recover heat recovery from gas side to water steam side. In economizer, the temp. of feed water is raised to its saturation temp. by heating of exhaust gases in evaporator while water disappears at a realizable pressure and temp. winding up submerged steam. In superheater, exhaust heat is help for superheating the steam produced in evaporator. Superheated steam is passed through steam turbine.

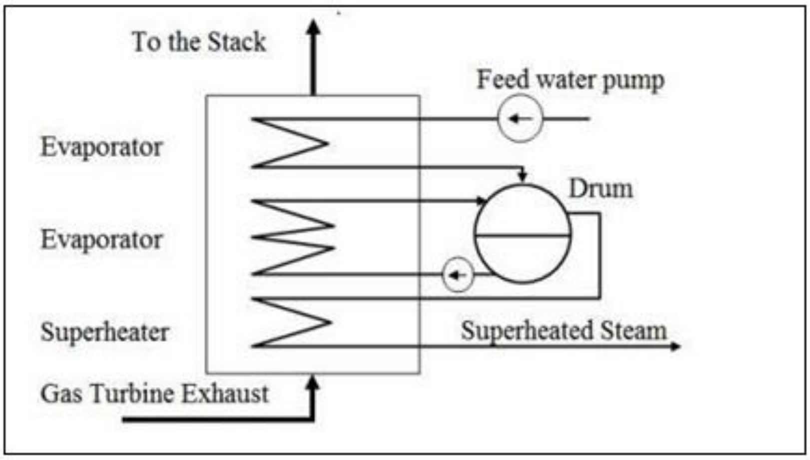


*Fig.3.9.Single pressure HRSG* [39]

Natural circulation normally comprises of the vertical tube and horizontal flow arrangement.

Natural circulation system works up by density difference between lower Chamber cold water and evaporator tubes hot steam water.

Forced circulation: HRSG are depicted by horizontal tubes with vertical gas stream and uses a pump to circle steam water mix.

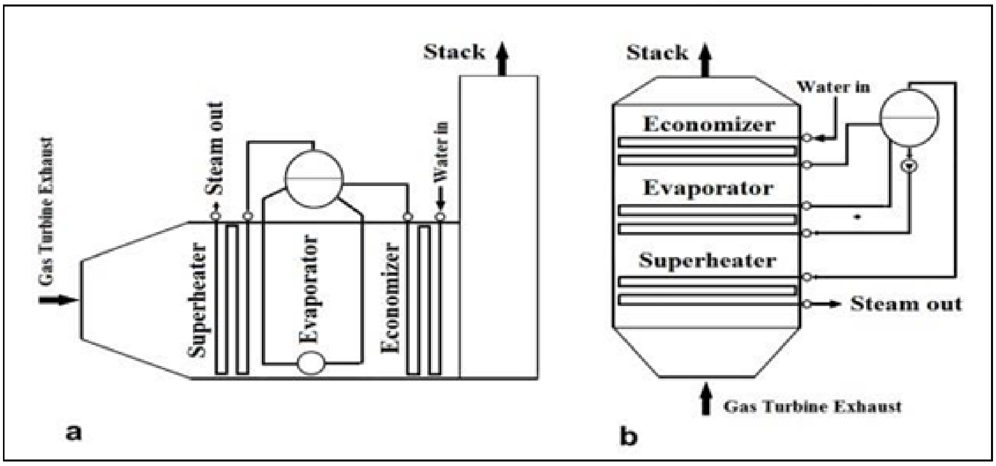


*Fig.3.10. HRSG (a) Natural circulation, (b) Forced circulation* [39]

### 3.2.7.1 Design Parameters and HRSG Performance

### 3.2.7.1.1 Design Parameters of HRSG

Parameters consider for any gas turbine HRSG:

- Pitch point temp.: This temp. is the differentiation between saturation temp. of the water and gas temp. leaving evaporator.
- Economizer approach temp.: This temp. is distinction between the saturated steam temp. and the temp. of water goes to the evaporator. Two parameters additionally influence the size of superheater, the evaporator and economizer. Lower temp. distinction implies that surface region need to create a similar heat transfer will be lot more prominent. The immediate outcome is that progressively material is utilized and these capital expenses are higher.

- Superheater approach temp.: the distinction between superheating steam temp. in superheater and the temp. gas in turbine outlet.
- Stack outlet temp.: the temp.: the temp. gases leaving to the ambient from HRSG.
- Allowable back pressure: the HRSG cross sectional region in a general sense impacts the gas turbine background. Minimized HRSGs requires higher gas turbine back pressure to drive through gas pipe, which reduction in size lesson HRSG cost, need to supply higher pressure turbine exit inconveniently influences gas turbine effectiveness. The ordinary estimations of gas turbine back pressure lies in the range of 2.5 to 3.7 kPa in numerous units. [39]

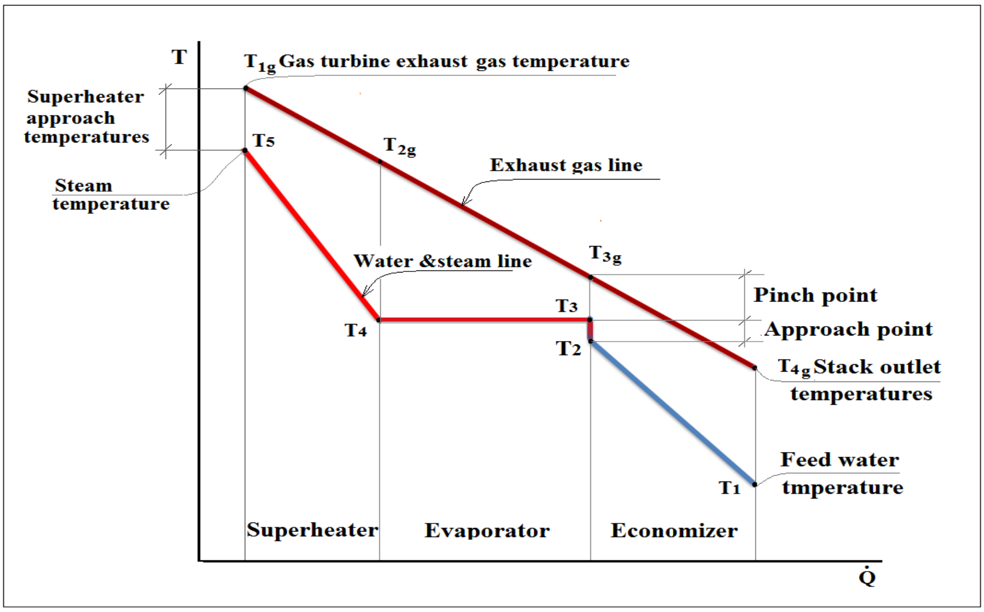


*Fig.3.11. T-Q Diagram of HRSG* [39]

## 3.3. Combined Power Plant Cycle

The CPPC comprises of Brayton and Rankine's cycle framing a standout amongst the most proficient cycles utilized for power age today. In the Brayton Cycle, there is the gas turbine cycle also knows as the topping cycle and the Rankine Cycle is Steam turbine cycle similarly know as bottoming cycle. The thermal efficiency of CPP system is more than 60% which is fairly high as of today.

Gas-turbine cycle works consistently at higher temp. compared to steam cycles.. At combustor exit of turbojet engines, it is more than 1500$^{o}$C. In gas turbines usage of higher temp. is made possible by late up gradation in cooling down the turbine blades and blades of turbine is covered by high temp. withstand material, like ceramics.In all case, the gas turbine cycles have one intrinsic shortcoming: high temp. gas leaving (i.e. over 500°C), the potential grabs in thermal efficiency. The condition can be overcome by using recuperation, not withstanding, limited changes.It looks good to use the most appealing aspects of the gas turbine cycle at high temp. and to utilize the high temp. exhaust gases as sources of energy for the bottoming cycle, for instances, a SPPC. In this process, energy is obtained from exhaust gases by passing it through the stream in the heat exchanger that fill in as boiler. To supply satisfactory heat to steam, gas turbine is relied upon considering all things. Similarly, steam cycles may inculcate recovery and further move warming.

Exhaust gas temp. of the gas turbine lies with the range of 450$^{o}$C-650$^{o}$C. Energy preserved in vent gas is a proportion of the fuel energy that is not changed over to control gas turbine. Some energy is utilized to increase the steam and to take control by the steam turbine. Depending on the sort of HRSG, temp. of pipe gas decrease to 80$^{o}$C-200$^{o}$C, where the lower regard given typical to immense present day combined cycle expending no fuel or no sulfur. The steam is conveyed with a temperature in the range of 30-170 bar. The steam can be transferred at various levels of pressure. The usage of supercritical steam pressure (>220.64 bar) was proposed by Balland. The supercritical steam pressure in combined cycle is higher efficiency. Higher temp. working cycle is known as topping cycle whereas the cycle which reutilize the energy transition is called bottoming cycle.

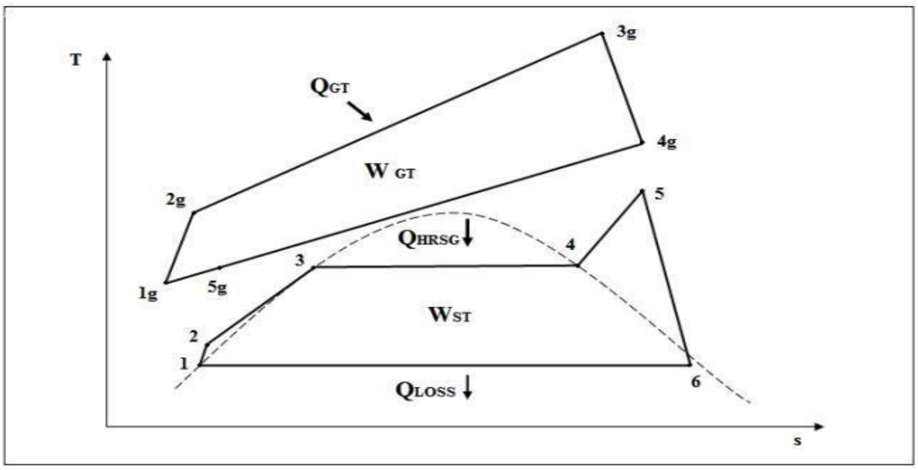


*Fig.3.12 T-s diagram of the combined power plant cycle* [40]

# CHAPTER 4

## 4. Exergy Analysis and Background enhancement

This section gives a crucial foundation to the advancement of exergoeconomic optimization, with administering conditions important to achieve optimization purpose. The exergoeconomic enhancement system has following steps-

### 4.1 Analysis of Energy

An energy examination relies upon 1st law of the thermodynamic. The 1st law of thermodynamics is generally called a law of energy conversion. 1st law of thermodynamics says that energy can neither be made not obliterated and it simply changes shape from one to another. This law defines internal importance as a working state and gives an explanation of conservation of the energy. The most well-known energy system, for example, refrigeration system and power production are an open systems. The usual system of power and refrigeration are Compressor, condenser and so on. All of which have inlet and outlet.

An energy study is typically utilized in assessing the exhibition of section or a system and can be utilized to manage the primary law productivity ($\eta th$) for power generation cycle.

Nevertheless, a vitality modify gives no information about the heading wherein methods can all of a sudden occur or possibly the reversibility of the thermodynamic methodology. The first law can't give information about the weakness of any thermodynamic methodology to change overheat totally into mechanical work, or any comprehension into why blend can't abruptly isolate themselves.

### 4.2 Analysis of Exergy

The exergy may be characterized as: maximum theoretical work acquired from the system when this system is carried from state to equilibrium with ambient while interfacing just with ambient. The condition of system is characterized by pressure, temp. Additionally, exergy is as hypothetical work expected to convey system from equilibrium with nature to given state.

This suggests exergy is a proportion of remove from system from the state of ambient. The significance of exergy won't be done, regardless, until portray the standard given condition.

Exergy examination is the procedures that use preservation of the mass and protection of energy standards together with $2^{nd}$ law of thermodynamics for examination, plan, and change of the energy system. Exergy system is an important instrument for encouraging target of increasingly beneficial vitality asset use, for its enable the zones, type, and certifiable extents of losses to be settled.

### 4.2.1 Exergy of the System

Exergy can be obtained from heat and work forms, thus

$$\Delta E_{system} = \Delta E_{heat} - \Delta E_{work} \quad 4.1$$

**Exergy related to work transfer** the meaning of work, that equal to given kind of energy as a proportion of its exergy, indisputably work is consistent to exergy in each respect. Along these lines, exergy transfer may be shown in size and in bearing by work transfer to which it related to:

$$\Delta E_{work} = W + \int P_0 dV \quad 4.2$$

**Exergy related with heat transfer** providing a consistent temp. dispersal in a thermal energy reservoir, transfer rate of exergy, $\Delta E\ work$ related to transfer rate of heat Q, may be expected by consequent strategy work:

$$\Delta E_{heat} = \left(1 - \frac{T_O}{T}\right)\mathrm{Q} \quad 4.3$$

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

$$\Delta E_{heat} = (Q - T_O \int dS) - (W - \int P_O dV) \quad 4.4$$

and rearranging

$$\Delta E_{heat} = Q - \mathrm{W} - T_O \int dS + \int P_O dV \quad 4.5$$

Introducing the first law:

$$\Delta \mathrm{E} = Q - \mathrm{W} \quad 4.6$$

$$\Delta E_{sytem} = \Delta \text{E} - T_O \int dS + \int P_O \text{dV} \quad 4.7$$

$$\Delta \text{E} = \Delta \text{U} - T_o \int dS + \int P_O \text{dV} + \frac{1}{2}\text{m}V^2 + \text{mgz} \quad 4.8$$

Integrating above all condition of system and end state becomes,

$$\text{E} = \text{U} - U_O - T_O\ (S - S_O) + P_O(V - V_O) + \frac{1}{2}\text{m}V^2 + \text{mgz} \quad 4.9$$

Here, V is velocity of system with respect to reference frame of ambient condition and z is height of system in respect to reference frame of ambient condition.

## 4.2.2 Exergy Component

Total exergy of system $E$ can be typed into four segments: potential exergy $E_{PT}$, kinetic exergy $E_{KN}$ , chemical exergy $E_{CH}$ , physical exergy $E_{PH}$

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

### 4.2.2.1 Potential Exergy

The potential exergy, $E_{PT}$ is resolved with the potential energy concerning zero levels related to the ambient. The potential energy should consider all of the qualities affecting the broke down issue and condition; it infers that, other than the surrounding constraint, one should in like manner consider the drive made by the weight of nature parts. Tolerating that accelerating of the gravity do not change with height, the potential energy can be found from going with:

$$\dot{E}_{\text{PT}} = \text{m}\left(gz - g\int_o^h y_o\, dh\right) \quad 4.11$$

where g, h, and $y_0$ are the gravity acceleration, the height of the matter from the center of with respect to zero level.

### 4.2.2.2 Kinetic Exergy

The kinetic exergy, $E_{KN}$ is equivalent to the kinetic energy determined with velocity of movement compared to ambient:

$$\dot{E}_{KN} = \frac{1}{2}\,\text{m}V^2 \quad 4.12$$

### 4.2.2.3 Chemical Exergy

In choosing physical exergy, condition of last steam is ecological state. By and by, this state will hide state in reversible methodology i.e use to choose substance the exergy of this materials stream. Acc. to that significance of the exergy, last state to which substances will decreased is reference dead state. Thusly, exergy of mixture is portrayed as most extraordinary works sensible when the substances under idea are passed on from regular state reference dead state by a procedure together with trade and exchange of substance just with ambient.

Equation of chemical exergy mixture as:

$$\dot{E}_{CH} = \sum x_n \, (\dot{e}_o)_n + \bar{R}T_O \sum x_n \log x_n \tag{4.13}$$

$$\dot{e}_o = T_O \log \frac{P_O}{P_{O\,O}} \tag{4.14}$$

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

Fuel specific exergy equation as:

$$\dot{e}_{CH,Fuel} = \beta(LCV) \tag{4.15}$$

$$\beta = 1.0437 + 0.1882\left(\frac{h}{c}\right) + 0.0610\left(\frac{o}{c}\right) + 0.0401\left(\frac{n}{c}\right) \tag{4.16}$$

. Where n, c, o, h are mass fraction of N, C, O, H respectively.

### 4.2.2.4 Physical Exergy

The physical exergy addresses that bit of exergy that shows up due to qualification in temp. and pressure of watched issue and temp. and pressure of ambient($T_0$,$P_0$). It is normally distinct into two segments.

Thermal component $E_{PH}^{\Delta T}$, is the segment following from the temp. change between the stream and environment:

$$E_{PH}^{\Delta T} = \mathrm{m}C_P\left[(T - T_O) - T_O \log\frac{T}{T_O}\right] \qquad 4.17$$

Pressure component $E_{PH}^{\Delta P}$, the is the segment following from the pressure change between the stream and the environment.

$$E_{PH}^{\Delta P} = (h - h_o) + T_O(S - S_O) \qquad 4.18$$

### 4.2.3 Exergy Balance

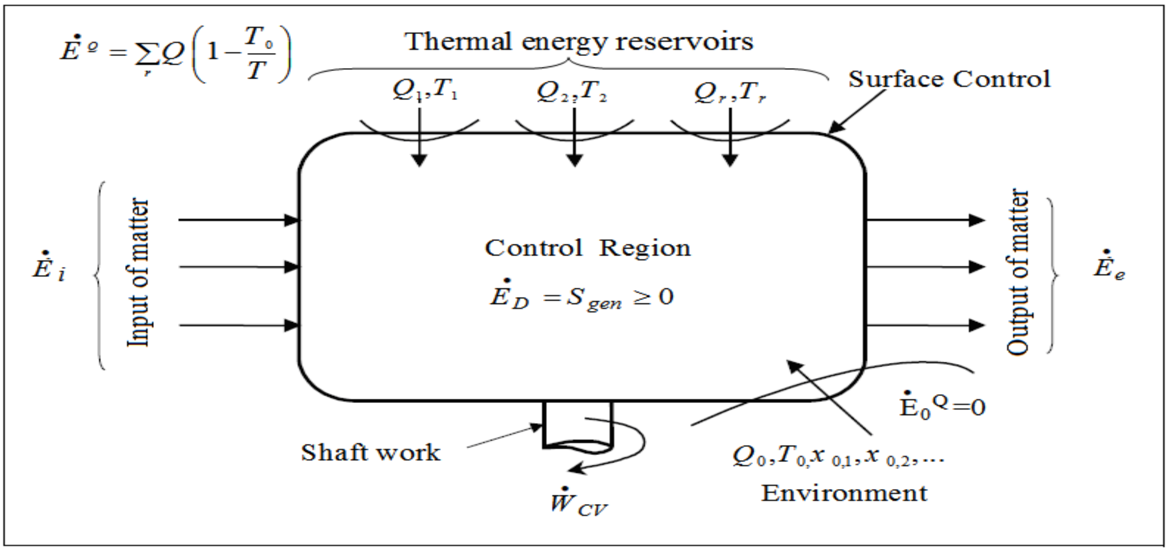


Fig. 4.1: State process in the open control region

Exergy analysis for an open system is [43]:

$[Rate\ of\ exergy\ change\ within\ control\ volume]$ = $[Net\ exery\ transfer\ by\ heat]$ + $[Net\ Exergy\ transfer\ by\ work]$ − $[Rate\ of\ Exergy\quad h\ mass]$ + $[Rate\ of\ Exergy\quad h\ mass]$ + $[Exergy\ destruction\ within\ control\ volume]$ 4.19

$$\left(\frac{dE}{dt}\right) = \sum j\left(1 - \frac{T_O}{T}\right)Q_j - W_{CV} + \sum m_i\,\dot{e}_i - \sum m_e\,\dot{e}_e - \dot{E}_D \qquad 4.20$$

Where $(dE\ dt)$ is time rate of change of the exergy stored inside the control volume, $\left(1 - \frac{T_O}{T}\right)Q_j$ is net exergy change because of heat transfer, $T_j$ is temp. at jth control volume boundary $W_{CV}$ is net exergy transfer because of non-flow works out of control volume; $\sum m\dot{e}$ is exergy addition or removal because of mass flow; $\dot{E}_D$ = $T_0 Sgen$ and is exergy dest. inside control volume.

## 4.2.4 Exergy Wastes

The thermal systems under thought for investigation is provided with little input i.e fuel exergy $\dot{E}_F$ got from the energy sources. This input moves into little exergy yield i.e product exergy $\dot{E}_P$ . For normal condition exergy input greater than exergy output, and this unbalance is because of losses in exergy. It is valuable to separate between sorts of exergy losses so as to think about irreversibilities happen. Two sorts of exergy waste may be recognized: internal exergy losses and external exergy losses [44].

External exergy losses $\dot{E}_{loss}$ speak to the remainings exergy substance of losses and discharges that are disseminated or expelled from generation and exemplify, hence unused.

Internal exergy losses $\dot{E}_D$ compare to the losses of quality because of internal inefficiencies aspects inside the process, it is the immediate aftereffect of the irreversibilities in a framework. Internal irreversibilities might be technical nature because of technical inefficiencie aspects inside the power plant, for example, friction, absence of insulation, they might be of basic sort. Structural exergy dest. $\dot{E}_D$ is controlled by standard and structure of system. Though exergy dest. $\dot{E}_D$ may be diminished through streamlining, auxiliary wastes can be decreased distinctly by upgrading system.

The exergy dest. depends on the entropy generation equation. Exergy dest. rate can write as:

$$\dot{E}_{loss} = \dot{E}_{out} - \dot{E}_D \qquad 4.21$$

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

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

## 4.2.5 Exergy Efficiency

Universal exergy efficiency is characterized by ratio of net exergy output respect to net exergy input. There are 2 principal modules of universal exergy efficiency revealed in written works. [45].

There is regularly a piece of output exergy which is not in use, for example, an exergy loss $\dot{E}_{loss}$ to ambient also, the exergy efficiency $\varepsilon_1$ moves toward becoming loss:

$$\varepsilon_1 = \frac{\dot{E}_{out} - \dot{E}_{loss}}{\dot{E}_{in}} \quad 4.24$$

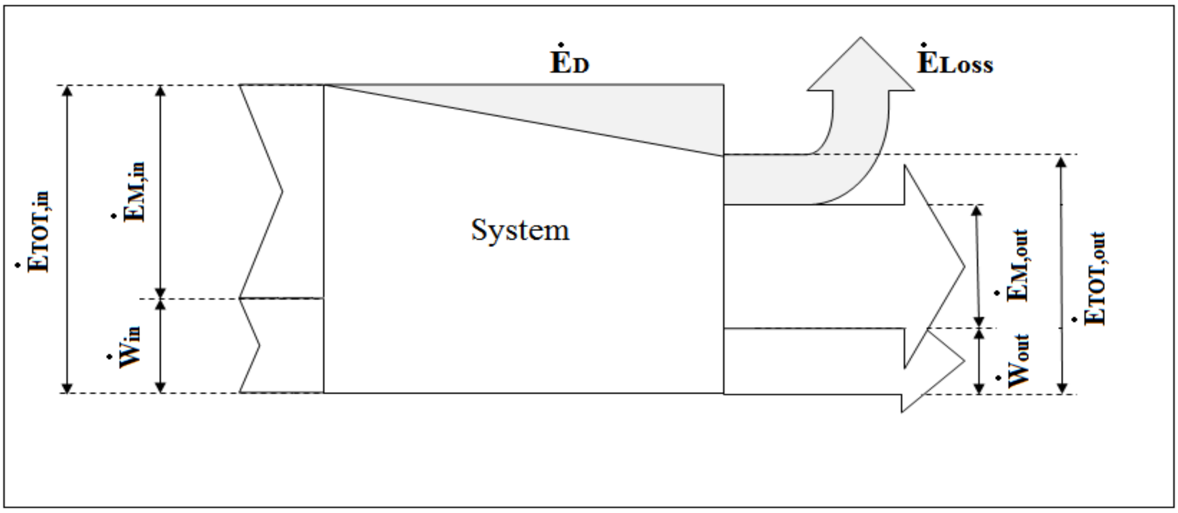


Fig.4.2. Exergy efficiency [45]

It is a for the most part generally definition of exergy efficiency, depends on exergy balance as express that all exergy used as exergy input and all exergy used as exergy output condition . Accordingly, the exergy efficiency becoming:

$$\boldsymbol{\varepsilon} = \frac{\dot{E}_{out}}{\dot{E}_{in}} = 1 - \frac{\dot{E}_D}{\dot{E}_{in}} \quad 4.25$$

The efficiency with traveling exergy $\varepsilon_{tr}$ is viewed as an advancement of basic efficiency. The unreconstructed parts are subtracted from internal and outward-bound segments. The efficiency will be characterized by:

$$\varepsilon_{tr} = \frac{\dot{E}_{out} - \dot{E}_{tr}}{\dot{E}_{in} - \dot{E}_{tr}} \quad 4.26$$

# CHAPTER 5

## 5. FORMULATION OF EQUATION

### 5.1. Exergy Analysis of Combined Power Plant

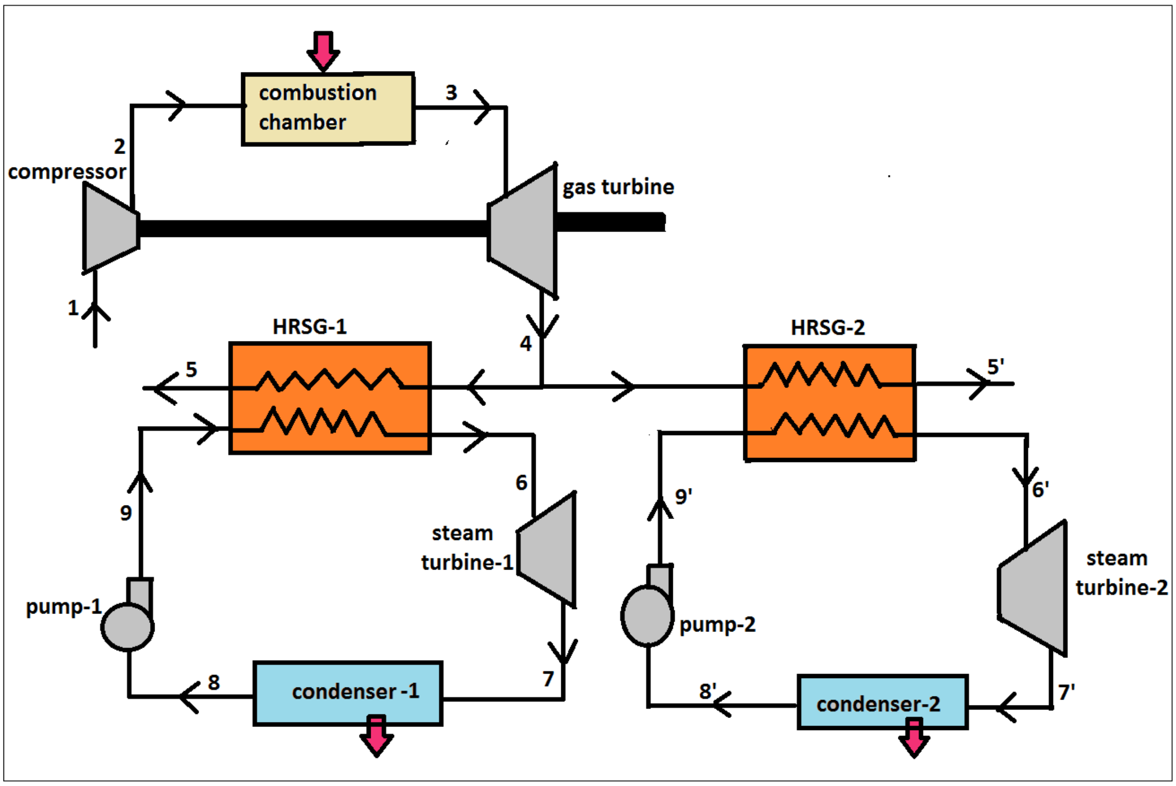


*Fig. 5.1: Diagram of combined power plant*

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

| Point number | Specification |
|---|---|
| 1 | Inlet air of Compressor |
| 2 | Outlet air from the Compressor |
| 3 | Combustion gases exiting the Combustion Chamber |
| 4 | Outlet hot gases exiting gas turbine |
| 5 | Outlet gases exiting from HRSG -1 |
| 5’ | Outlet gases exiting from HRSG -2 |
| 6 | Superheated steam entering Steam turbine -1 |

| | |
|---|---|
| 6’ | Superheated steam entering Steam turbine -2 |
| 7 | Outlet steam from the Steam turbine -1 |
| 7’ | Outlet steam from the Steam turbine -2 |
| 8 | Saturated liquid entering feed water pump -1 |
| 8’ | Saturated liquid entering feed water pump -2 |
| 9 | water entering HRSG – 1 |
| 9’ | water entering HRSG – 2 |

The essential conditions utilized in exergy examination carried out on the selected CPPC are exhibited in chapter. Likewise, the energy analysis, exergy balance for every segment is composed, and the exergy streams and irreversibilities for every segment are found.

### 5.1.1. Compressor

$$\triangle X_{dest} = X_1 - X_2 + W_c \quad 5.1$$

$$\eta_{ex\ comp} = \frac{X_2 - X_1}{W_C} \quad 5.2$$

Where, $X_1 = m_a[h_1 - h_O - T_O(s_1 - s_O)]$ 5.3

$$X_2 = m_a[h_2 - h_O - T_O(S_1 - S_O)] \quad 5.4$$

$$W_C = m_a(h_2 - h_1) \quad 5.5$$

### 5.1.2. Combustion Chamber

$$\Delta_{dest} = X_2 + X_f - X_3 \quad 5.6$$

$$\eta_{ex\prime\ cc} = \frac{X_3}{X_2 + X_f} \quad 5.7$$

$$X_f = X_{ph} + X_{ch} \quad 5.8$$

$$X_{ph} = m_a\left[h_f - h_O - T_O\left(s_f - s_O\right)\right] \quad 5.9$$

$$X_{ch} = \dot{m}_f e_{ch} \qquad 5.10$$

Where, $e_{ch} = \dot{x}_i e_{chi} + RT_O \sum x_i \log x_i + Ge$ 5.11

Where Ge is Gibbs free energy which is a neglected quantity in a gas mixture operated at lower pressure. Thus, the fuel exergy can be calculated as the ratio fuel exergy to low value of heating value of fuel.

$$\Omega = \frac{e_f}{LCV} \qquad 5.12$$

$e_f$ is specific exergy of the fuel.

For gaseous fuel with creation $\boldsymbol{CxHy}$, the estimation of Ω can be determined as

$$\Omega = 1.033 + 0.0169\frac{Y}{X} - \frac{0.0698}{X} \qquad 5.13$$

For Methane (CH4) X=1, Y=4

Then $\Omega = 1.06$

$$X_f = \dot{m}_f(1.06 * LCV) \qquad 5.14$$

## 5.1.3. Gas Turbine

$$\triangle X_{dest} = X_3 - X_4 - W_{GT} \qquad 5.15$$

$$\eta_{ex\prime\, GT} = \frac{W_{GT}}{X_3 - X_4} \qquad 5.16$$

Where, $X_4 = (\dot{m}_a + \dot{m}_f)[h_4 - h_O - T_O(s_4 - s_o)]$ 5.17

$$X_3 = (\dot{m}_a + \dot{m}_f)[h_3 - h_O - T_O(S_3 - S_O)] \qquad 5.18$$

$$W_{GT} = (\dot{m}_a + \dot{m}_f)[h_3 - h_4] \qquad 5.19$$

## 5.1.4. HRSG-1

$$\Delta X_{dest} = X_{in} - X_{out} \qquad 5.20$$

$$\eta_{HRSG-1} = \frac{X_9 - X_6}{X_4 - X_5} \qquad 5.21$$

Where, $X_{in} = X_4 + X_9$ 5.22

$X_{out} = X_5 + X_6$ 5.23

$X_4 = (m_a + m_f)[(h_4 - h_0) - T_O(S_4 - S_O)]$ 5.24

$X_9 = (m_s)[(h_9 - h'_o) - T_O(S_9 - S_O)]$ 5.25

$X_5 = (m_a + m_f)[(h_5 - h'_O) - T_O(S_5 - S_O)]$ 5.26

$X_6 = (m_s)[(h_6 - h'_O) - T_O(S_6 - S_O)]$ 5.27

## 5.1.5. HRSG-2

$$\Delta X_{dest} = X_{in} - X_{out} \quad 5.28$$

$$\eta_{HRSG-2} = \frac{X'_9 - X'_6}{X_4 - X'_5} \quad 5.29$$

Where, $X'_{in} = X_4 + X_9'$ 5.30

$X'_{out} = X'_5 + X'_6$ 5.31

$X_4 = (m_a + m_f)[(h_4 - h_0) - T_O(S_4 - S_O)]$ 5.32

$X_9 = (m'_s)[(h'_9 - h'_o) - T_O(S'_9 - S'_O)]$ 5.33

$X'_5 = (m_a + m_f)[(h'_5 - h_O) - T_O(S'_5 - S_O)]$ 5.34

$X'_6 = (m'_s)[(h'_6 - h'_O) - T_O(S'_6 - S'_O)]$ 5.35

## 5.1.6. Steam Turbine-1

$$\Delta X_{dest} = X_6 - X_7 - W_{ST} \quad 5.36$$

$$\eta_{ex'ST} = \frac{W_{ST}}{X_6 - X_7} \quad 5.37$$

Where, $X_6 = m_s[(h_6 - h'_O) - T_O(S_6 - S_O)]$ 5.38

$X_7 = m_s[(h_7 - h'_O) - T_O(S_7 - S_O)]$ 5.39

$W_{ST} = m_s(h_6 - h_7)$ 5.40

### 5.1.7. Steam Turbine-2

$$\Delta' X_{dest} = X'_6 - X'_7 - W'_{ST} \quad 5.41$$

$$\eta_{ex' ST} = \frac{W'_{ST}}{X'_6 - X'_7} \quad 5.42$$

Where, $X'_6 = m'_s[(h'_6 - h'_O) - T_O(S'_6 - S'_O)]$ 5.43

$$X'_7 = m'_s[(h'_7 - h'_O) - T_O(S'_7 - S'_O)] \quad 5.44$$

$$W'_{ST} = m'_s(h'_6 - h'_7) \quad 5.45$$

*Table 5.2: Component wise Exergetic dest. rate and Exergetic Efficiencies*

| **components** | **Exergy Destruction Rate** | **Exergetic Efficiency** |
|---|---|---|
| Compressor | $X_1 - X_2 + W_C$ | $\frac{X_2 - X_1}{W_C}$ |
| Combustion Chamber | $X_2 + X_f - X_3$ | $\frac{X_3}{X_2 + X_f}$ |
| Gas Turbine | $X_3 - X_4 - W_{GT}$ | $\frac{W_{GT}}{X_3 - X_4}$ |
| HRSG-1 | $X_4 + X_9 - X_5 - X_6$ | $\eta_{HRSG-1} = \frac{X_9 - X_6}{X_4 - X_5}$ |
| HRSG-2 | $X_4 + X'_9 - X'_5 - X'_6$ | $\eta_{HRSG-2} = \frac{X'_9 - X'_6}{X_4 - X'_5}$ |
| Steam Turbine-1 | $X_6 - X_7 - W_{ST}$ | $\frac{W_{ST}}{X_6 - X_7}$ |
| Steam Turbine-2 | $X'_6 - X'_7 - W'_{ST}$ | $\frac{W'_{ST}}{X'_6 - X'_7}$ |

# CHAPTER 6

## 6.Results and Discussion

Subsequently the research of distinct cycles on which the thermal power plant operates with respect to exergy losses and efficiency of a component in CPP system. Exergy dest. exhibits a loss that can be recuperated by proper designing of the different sections of the system and it verifies the best feasible process of the power plant according to 2nd law of thermodynamic. The energy and exergy cost are commendatory, exergy dest. reveals a loss which can be determined by the study mathematically. In the work, the studied is done on the CPPC.

## 6.1 Results from Exergy analysis

### 6.1.1 Exergy Destruction rate

### 6.1.1.1.Effect of A/F (air-fuel ratio) at a different inlet temp. of air Compressor

Fig.6.1 The variation of Exergy dest. rate of the air Combustion Chamber as a function of AFR at a different inlet temp of the air Compressor. AFR was changed from 50 to140 in a period of 10.Inlet temp was changed from 290 to 310 in a period of 5.

As the AFR increases the exergy dest. rate of Combustion Chamber increase.this is because of a large amount of heat addition.

Aat a particular AFR, as exergy dest. rate is decreasing with increase of inlet temp of the Air Compressor. This is due to increased inlet temp of Air Compressor, the supply of air in the Combustion Chamber at high temp so it needs low chemical energy addition.

This CPPC uses two HRSG. If exit temp of gas turbine is more than 723k than HRSG-1 will come in action otherwise HRSG -2.

Fig.6.2.Tthe variation of exergy dest. rate of HRSG-1 as a function of AFR at a different inlet temp of air Compressor. Inlet temp was changed from 290 to 310 in a period of 5 and the air fuel ratio was changed from 50 to 70 in a period of 10.Exergy dest. rate decrease with an increase of AFR because inflow temp is high at a lower value

of AFR and inflow temp decrease as the AFR increase that why decease of exergy dest. rate.

At a certain AFR, exergy dest. rate increase with increase of inlet temp of air Compressor. This is because more amount of heat transfer at higher temp takes place in HRSG-1.

Fig.6.3. The variation of exergy dest. rate of HRSG-2 as a function of AFR at a different inlet temp of the air Compressor. AFR was changed from 80 to 140. Exergy dest. rate decrease with the increase of AFR similar to what happens in HRSG-1 but the value of dest. rate in HRSG-1 lower as compared to HRSG-2.

At a certain AFR exergy dest. rate of HRSG-2 increase with the increase of inlet temp of air Compressor. In HRSG-2 less dest. rate value as compared to HRSG-1 because HRSG-2 works on lower temp as compared to HRSG-2.

This CPPC uses two Steam turbine but only one Steam turbine works at a time. If exit temp of gas turbine is more than 723k then Steam turbine-1 will come in action otherwise Steam turbine -2.

Fig.6.4. The variation of exergy dest. rate of the Steam turbine-1 as a function of AFR at a different inlet temp of air Compressor. Inlet temp. was changed from 290 to 310 in a period of 5 and the AFR was changed from 50 to 70 in a period of 10. As the AFR increases the exergetic efficiency of Steam turbines-1 decrease because of the amount of fuel decreases and the temp. at the inlet to the steam turbine decreases.

Fig.6.5. The variation of exergy dest. rate of Steam turbine-2 as a function of AFR at a different inlet temp. of air Compressor. Inlet temp. was changed from 290 to 310 in a period of 5 and the AFR was changed from 80 to 140 in a period of 10. As the AFR increases exergy dest. rate of Steam turbines-2 decrease. Reason is same as Steam turbine-1.At a particular AFR, exergy dest. rate of Steam turbine-2 increase with the increase of inlet temp. of air Compressor.

## 6.1.2 Exergetic Efficiency

### 6.1.2.1 Effect of A/F (AFR) at a different inlet temp. of air Compressor

Fig.6.6.The variation of Exergetic efficiency of air Compressor as a function of AFR at a different inlet temp. of the air Compressor. Inlet temp. was changed from 290 to 310 in a period of 5 while the AFR was changed from 50 to 140 in a period of 10. The mass of fuel is constant and that is equal to 1kg and the mass of air increase as the AFR increase. Exergetic efficiency does not change with the change in the AFR. This is due to the exergetic efficiency of the air Compressor is not a function of the AFR.

Fig.6.7. The variation of Exergetic efficiency of air Combustion Chamber as a function of AFR at a different inlet temp. of the air Compressor. As the AFR increases the exergetic efficiency of Combustion Chamber decrease at a certain point after that it starts increasing. At a particular AFR, exergetic efficiency of air Combustion Chamber increase with the increase in inlet temp. of air Compressor because the required amount of heat in the Combustion Chamber is decreased by high temp. coming from the Compressor.

Fig.6.8. The variation of exergetic efficiency of gas turbine as a function of AFR at a different inlet temp. of the air Compressor. As the AFR increases the exergetic efficiency of gas turbine decrease. At a particular AFR, Exergetic efficiency of gas turbine increase with the increase of inlet temp. of air Compressor.

Fig.6.9. The variation of exergetic efficiency of HRSG-1 as a function of AFR at a different inlet temp. of air Compressor. Inlet temp. was changed from 290 to 310 in a period of 5 and the AFR was changed from 50 to 70 in a period of 10. As the AFR increases exergetic efficiency of HRSG-1 increase because the temp.of hot gases entering will be low athigher AFR. At a particular AFR, Exergetic efficiency of gas turbine decrease with increase of inlet temp. of air Compressor.

Fig.6.10.Shows the variation of exergetic efficiency of HRSG-2 as a function of AFR at a different inlet temp. of air Compressor. Inlet temp. was changed from 290 to 310 in a period of 5 and the AFR was changed from 80 to 140 in a period of 10. As the AFR increases the exergetic efficiency of HRSG-2 increase. Reason is same as HRSG-1. At a particular AFR, Exergetic efficiency of gas turbine decrease with the increase of inlet

temp. of air Compressor. In HRSG-2 value of exergetic efficiency is higher as compared to HRSG-1.

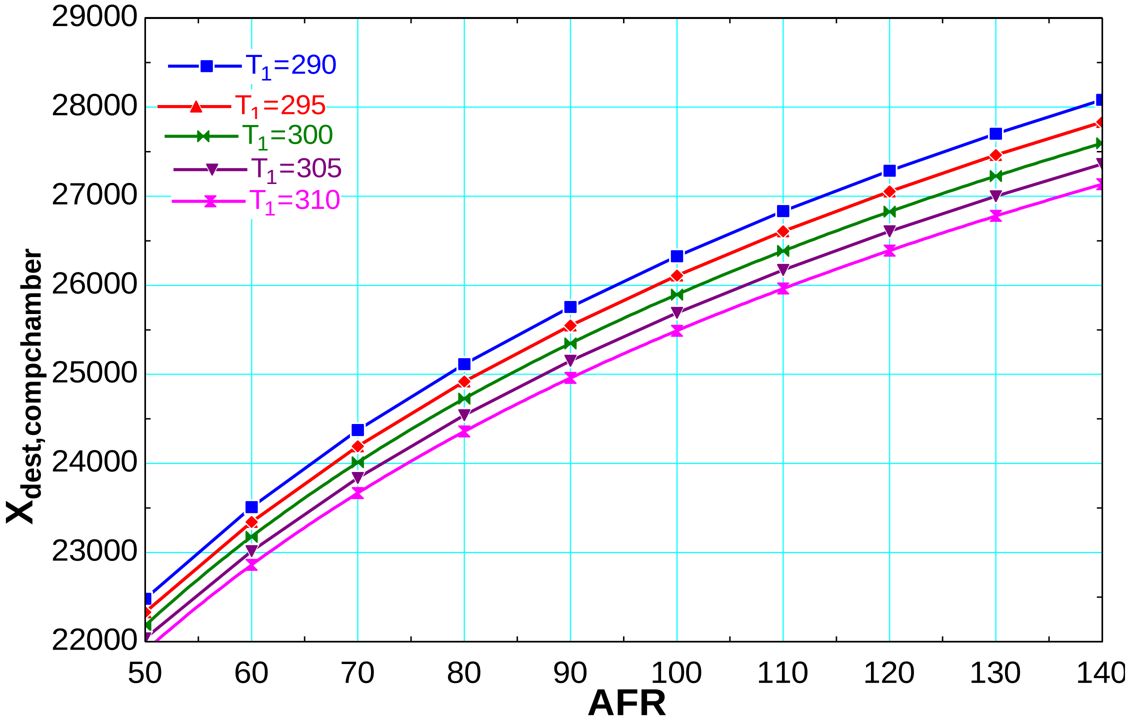


Fig.6.1: Exergy Dest. Rate of Combustion Chamber at various inlet temp VS Air Fuel Ratio

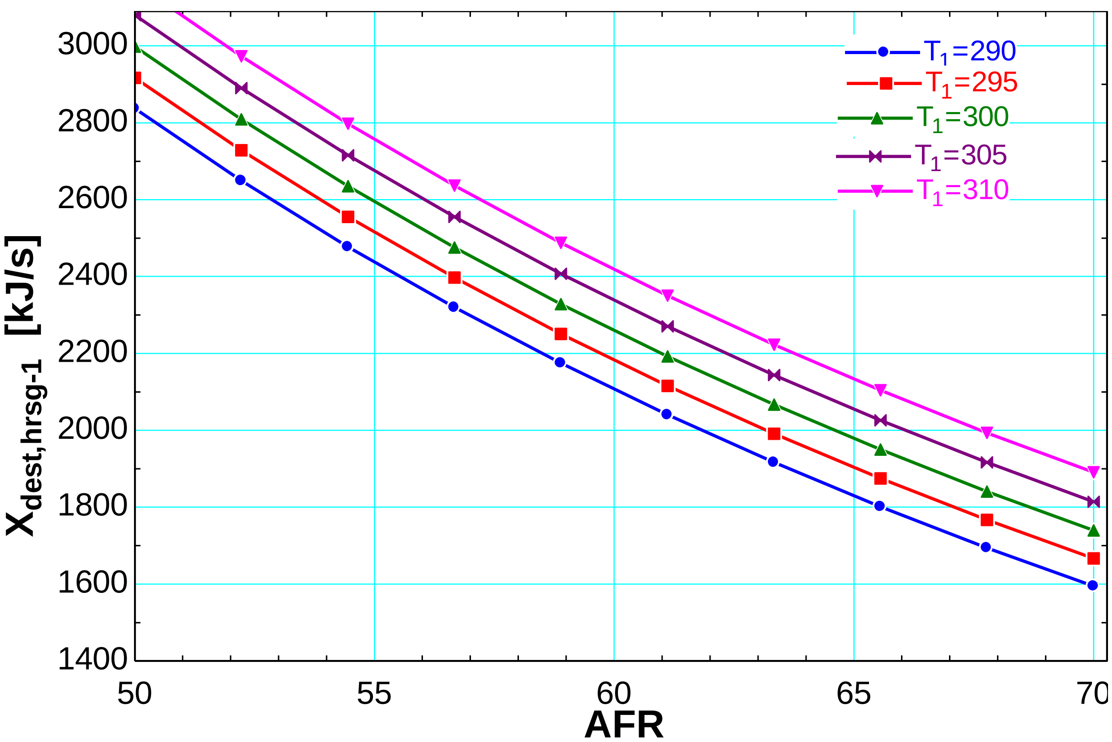


Fig.6.2: Exergy Dest. Rate of HRSG-1 at various inlet temp VS Air fuel Ratio

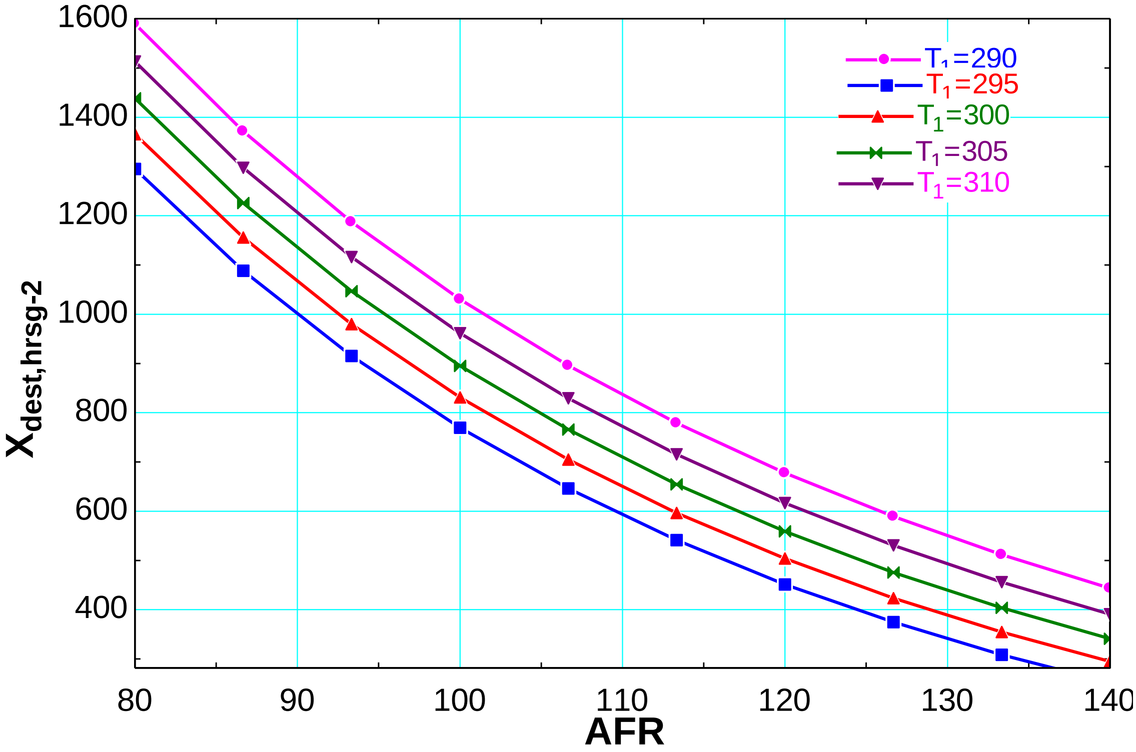


Fig.6.3: Exergy Dest. Rate of HRSG-2 at various inlet temp VS air fuel Ratio

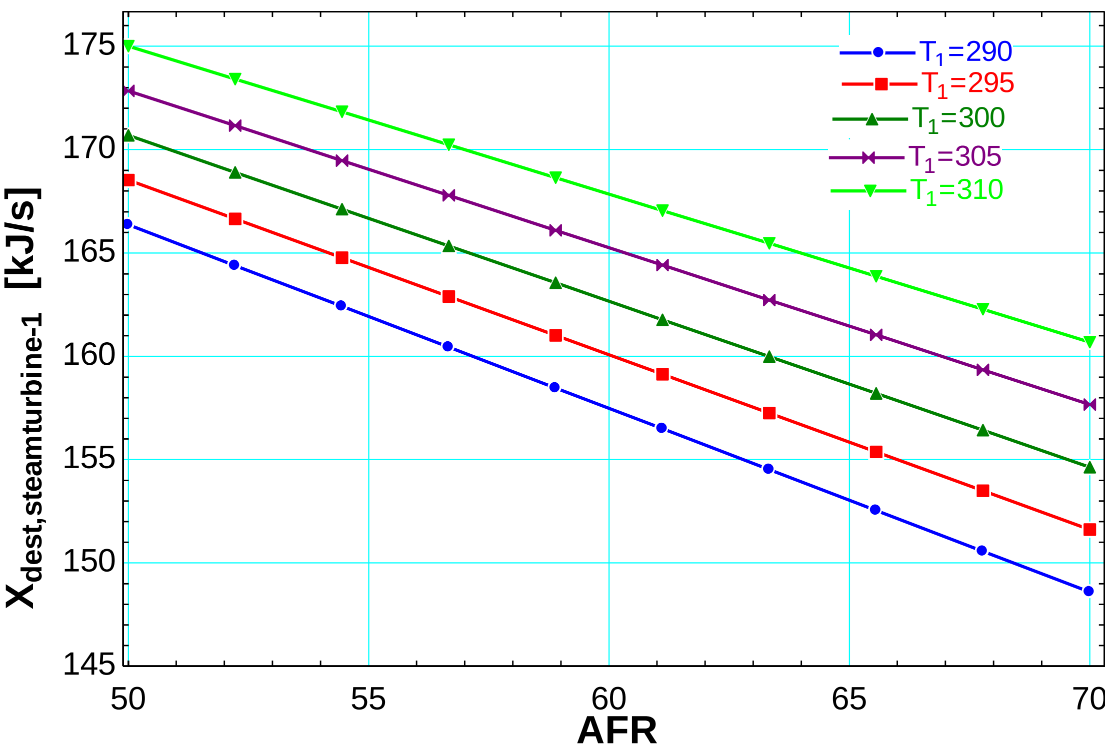


Fig.6.4: Exergy Dest. Rate of Steam turbine-1 at various inlet temp VS air fuel Ratio

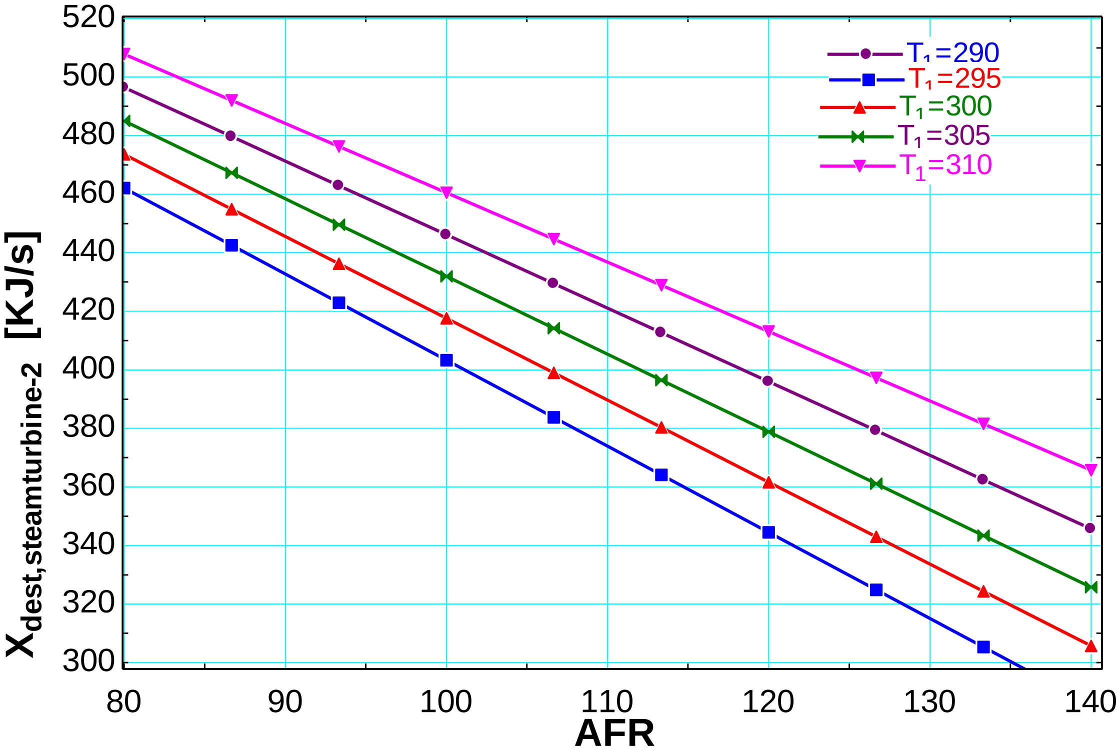


Fig.6.5: Exergy Dest. Rate of Steam turbine-2 at various inlet temp VS air fuel Ratio

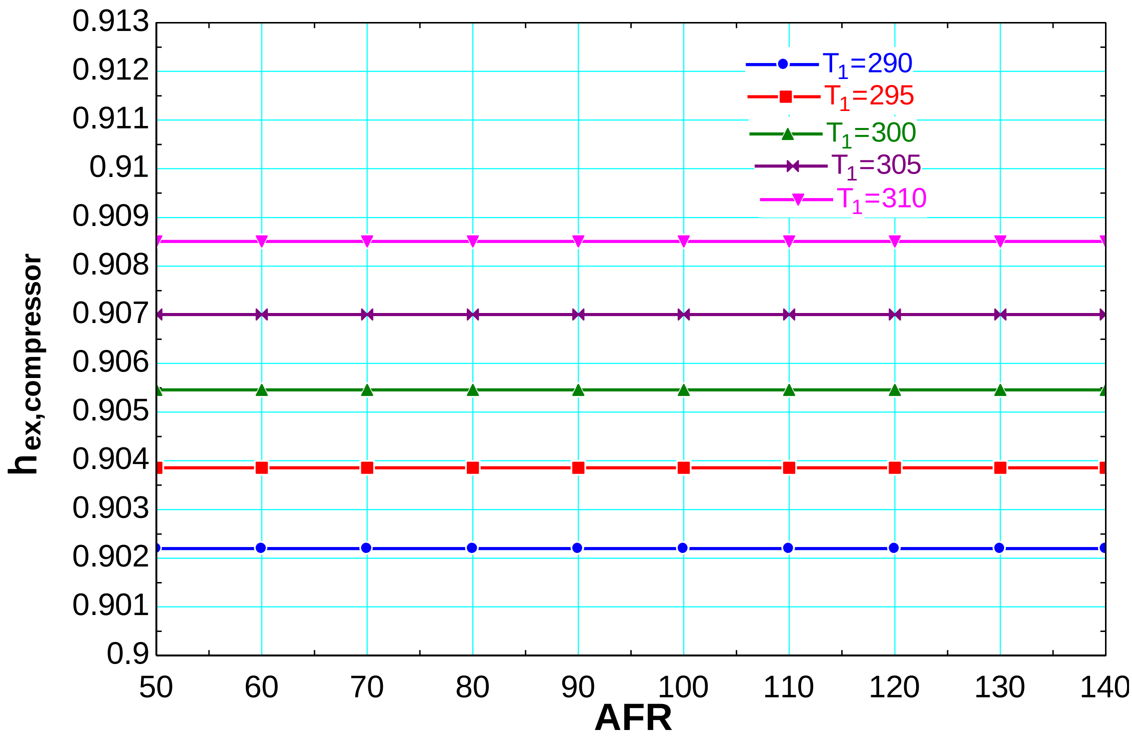


Fig.6.6: Exergetic Efficiency of the Compressor at various inlet temp VS Air Fuel Ratio

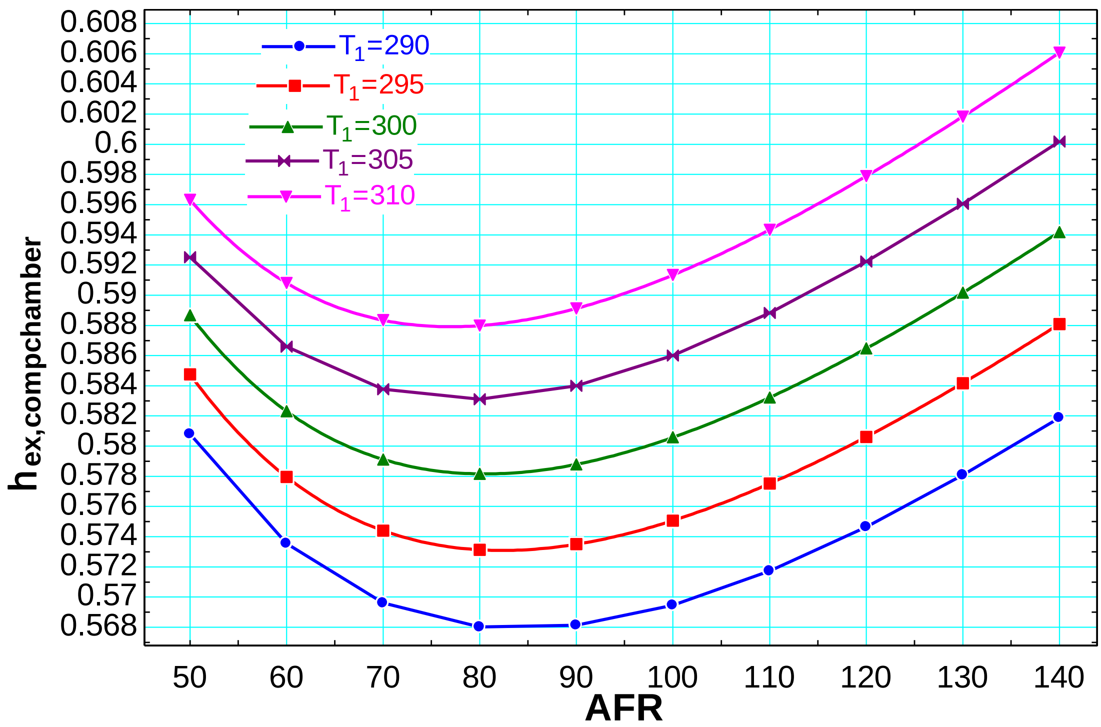


Fig.6.7: Exergetic Efficiency of Combustion Chamber at various inlet temp VS Air Fuel Ratio

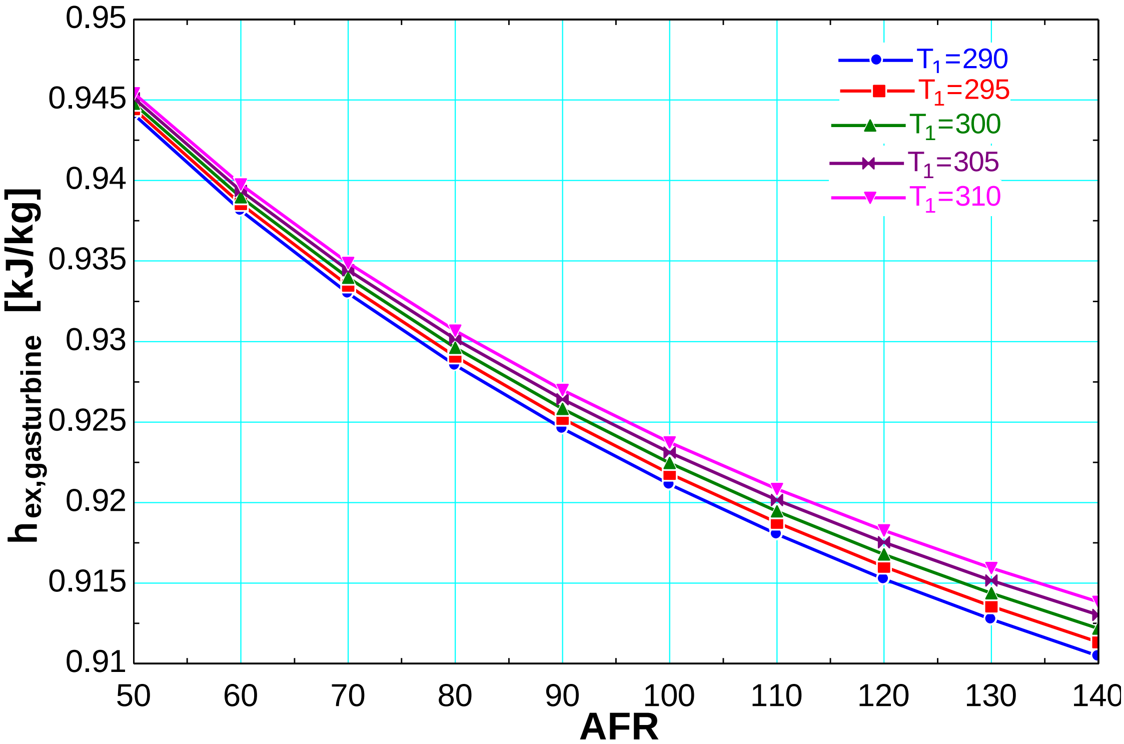


Fig.6.8: Exergetic Efficiency of Gas turbine at various inlet temp VS Air Fuel Ratio

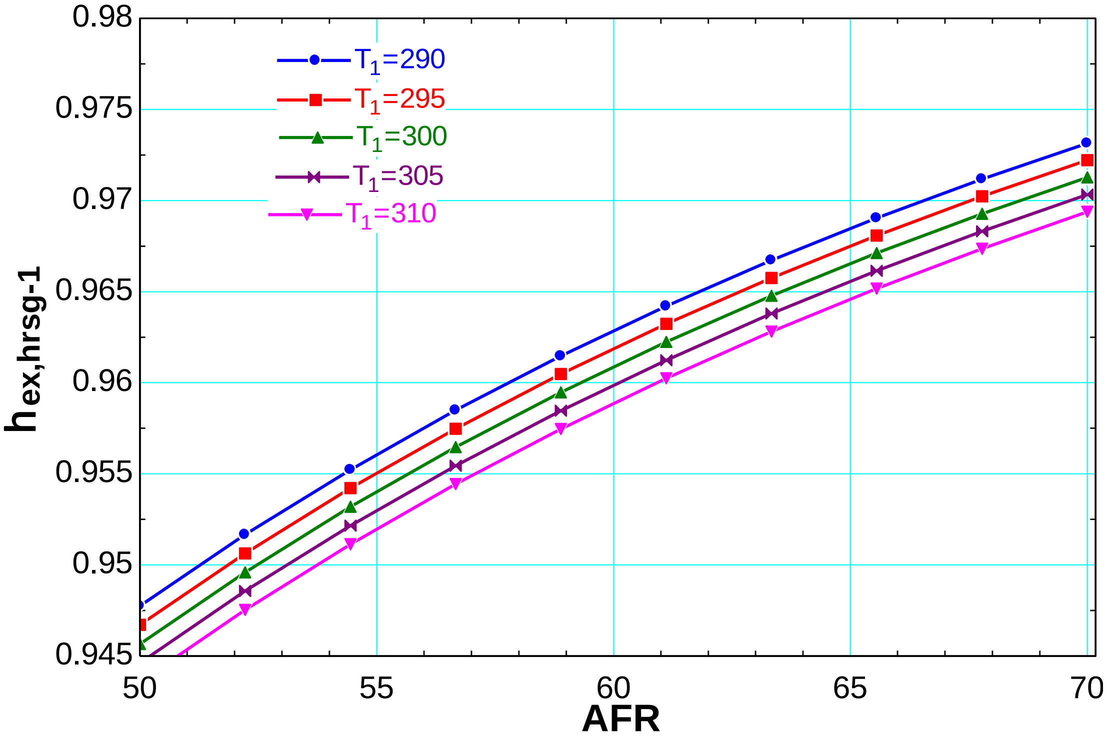


Fig.6.9: Exergetic Efficiency of HRSG-1 at various inlet temp VS Air Fuel Ratio

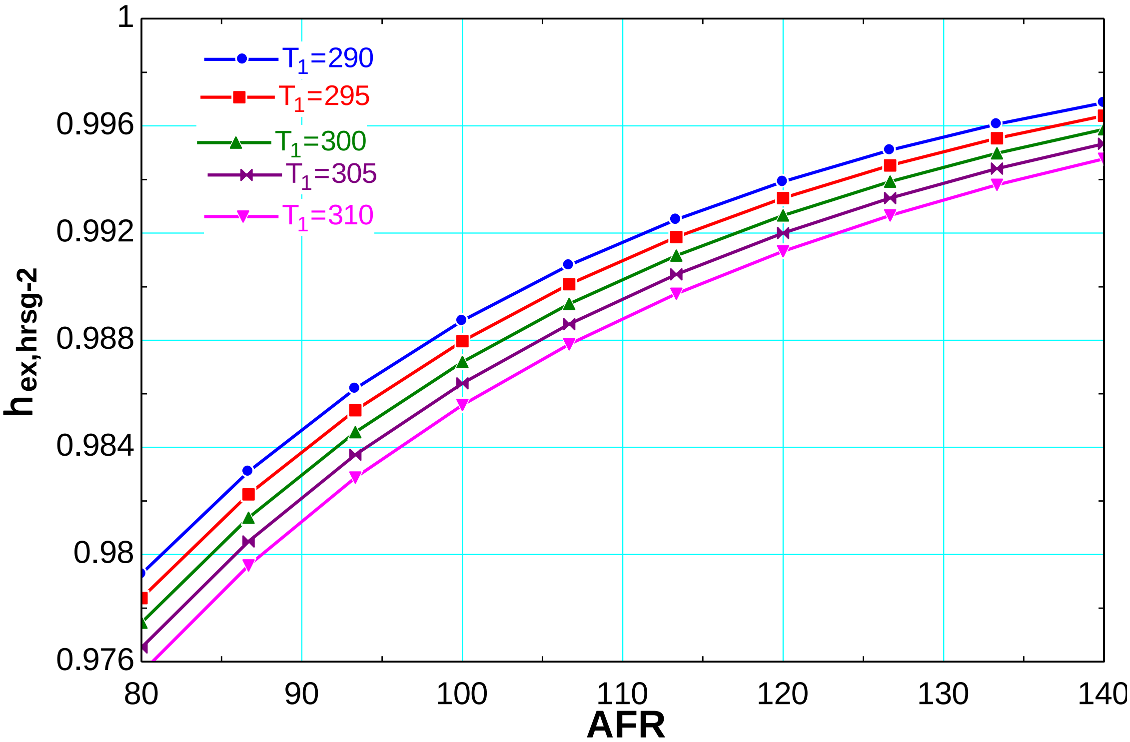


Fig.6.10: Exergetic Efficiency of HRSG-2 at various inlet temp VS Air Fuel Ratio

# CHAPTER 7

## 7. Conclusion

The key point of the exergy analysis is to differentiate the scope and the range of real energy losses to upgrade the ongoing systems or methods. The effect of different variables namely inlet temp. and pressure of system execution is analyzed in this report parametrically alongside 2nd law of thermodynamics of analysis of a thermal power plant has been performed in this report.

CPPC oversee the growing energy requirement therefore, the distinctive analysis must be inclined to distinguish a methodology for the enhancement of these systems. The exergy examination manifested for a CPPC has disposed of statistics on the irreversibility's of each operation. The exergy analysis made for this composite plant exhibited that the prominent exergy losses exist in Combustion Chamber, gas turbine, and HRSG while most of the energy losses occur at the stack. The strategies suggested to enlarge the efficiency of plant is higher gas turbine inlet temp. using usual super compounds in gas turbine hot segments, Compressor inlet air cooling and a progressive strategy for the HRSG working variables.

The exergy examination shows that he HRSG has the greatest effect component on overall system and therefore, these further notable components were taken into analysis. Thus the center of interest this thesis is on variables in optimization and indicates that a decrement in the exergy dest. of HRSG component can enhance system performance.

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# APPENDICES

## Appendix 1: Gas Turbine Parameters

| PARAMETERS | VALUE |
|---|---|
| Ambient air pressure [bar] | 1.013 |
| Ambient air temp. [°C] | 290-315 |
| Turbine isentropic efficiency [%] | 90 |
| Compressor isentropic efficiency [%] | 88 |
| Compression ratio [-] | 4 |
| 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

| Parameters | Value |
|---|---|
| The pinch point temp. difference[°C] | 13 |
| Pressure in the drum of HRSG | 10 |
| Live Steam temp. at the inlet of low-pressure Steam turbine-1[°C] | 450 |
| Live Steam temp. at the inlet of low-pressure Steam turbine-2[°C] | 380 |
| Minimum stack temp. [°C] | 150 |
| Feed water temp. at 9 [°C] | 55 |

## Appendix 3: Steam Turbine Parameters

| Parameters | value |
|---|---|
| The isentropic efficiency of the Steam turbine | 90 |
| The isentropic efficiencies of water pumps | 52 |
| Minimum dryness fraction of Steam at the low Steam turbine outlet | .88 |
| Condenser pressure [bar] | .07 |
| The inlet cooling water temp. in Condenser [°C] | 20 |

# Appendix 4: Thermodynamic Properties of Steam

| $t$, °C | 1.0 MPa ($t_S$ = 179.878 °C) | | | |
|---|---|---|---|---|
| | $v$ | $\rho$ | $h$ | $s$ |
| $t_S$(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 | .98 | -0.00009 |
| **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** | 211.5 | 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 |

| | **1.0 MPa ($t_s$ = 179.878 °C)** | | | |
|---|---|---|---|---|
| $t$, °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 | 0.9527 | 7376.8 | 10.489 |

# Appendix 5: Water Properties

| $p$, MPa | $T$, $^{o}$C | Density,kg/m3 | | Enthalpy,kJ/k | | Entropy,kJ/(kgk)· | | Volume,cm$^3$/g | |
|---|---|---|---|---|---|---|---|---|---|
| | | $\rho_L$ | $\rho_v$ | $h_L$ | $h_v$ | $s_L$ | $s_v$ | $v_L$ | $v_v$ |
| **0.0070** | 39.0 | 992.5 | 0.04872 | 163.3 | 25717 | 0.55903 | 8.274 | 1.0075 | 20524. |