Thermoeconomic Effect of Air Fuel Ratio On Combined Cycle Power Plant at Various Pressure Ratios

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

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DECLARATION

I hereby declare that the work which being presented in the major thesis entitled "**Thermoeconomic Effect of air fuel ratio on combined cycle Power Plant at various pressure ratios**" in the partial fulfilment for the award of the degree of Master of Technology in **"Thermal Engineering"** submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of **Dr. B. B. ARORA,** Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever. I confirm that I have read and understood 'Plagiarism policy of DTU'. I have not committed plagiarism while completing the attached piece of work, similarity found after checking is **15%** which is below the permitted limit of 20%.

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This is to certify that CHINMAY JAIN, (2K15/THE/05), student of M.Tech, "THERMAL ENGINEERING", Delhi Technological University, has submitted the dissertation titled **"Thermoeconomic Effect of air fuel ratio on combined cycle Power Plant at various pressure ratios"** under our guidance towards the partial fulfilment of the requirements for the award of the degree of Master of Technology under our guidance and supervision.

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ABSTRACT

The objective of the presented research work is to develop an Thermoeconomic optimization method in order to predict the cost effectiveness of a combined cycle gas turbine (CCGT) power plant and suggest ways of improving the cost effectiveness from both thermodynamic and economic points of view

The Thermoeconomic analysis provides a complete diagnosis of the performance of the combined cycle power plant, both in energetic and in monetary values. In this regard, Thermoeconomic optimization is a better tool as it combines the thermodynamic analysis with the economic principles. Here, appropriate costs are assigned to the thermodynamic inefficiencies of the system components, which add to the hidden cost. The system considered in this thesis is a combined cycle power plant. For this system, an energetic and economic analyses are performed to predict thermodynamic and economic parameters of the system. This system is Thermoeconomically optimized to reduce the specific total cost of the products, which leads to a reduction in the investment cost, and to an increase in the power output and efficiency

A program of numerical code is established using MATLAB software to perform the calculations required for the thermal and economic analysis considering real variation ranges of the main operating parameters such as pressure, temperature air fuel ratio and mass flow rate. The effects of theses parameters on the system performances are investigated. The optimization results demonstrate that all three optimization methods can improve the thermodynamic and economic performance, but with different values. Thermoeconomic optimization method is the most effective method for designing and operating a system with high efficiency and low investment cost.

Key words: Combined cycle, heat recovery steam generator, exergy, thermodynamic optimization, Thermoeconomic optimization.

Scientific field:
Mechanical engineering, Thermal engineering
Narrow scientific field:
Thermal engineering

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NOMENCLATURE

C	Cost	₹
C_p	Heat capacity at constant pressure	kJ/kg·K
$C_{p_{steam}}$	Heat capacity at constant pressure for steam	kJ/kg·K
C_{p_g}	Heat capacity at constant pressure for exhaust gas	kJ/kg·K
C_{p_a}	Heat capacity at constant pressure for air	kJ/kg·K
g	Gravity acceleration	m/s^2
h	Specific enthalpy	kJ/kg
H	Enthalpy	kJ
LH	Latent heat	kJ/kg
CV	Calorific value	kJ/kg
LHV	Low heating value	kJ/kg
$\dot{m}$	Mass flow rate	kg/s
p	Pressure	bar
PP	Pinch point	K or ℃
$\dot{Q}$	Heat transfer	kW
Q	Heat supplied or rejected	kJ
q	Specific heat supplied or rejected	kJ/kg
S	Entropy	kJ/K
s	specific entropy	kJ/kg·K
S_{gen}	Entropy generation	kJ/K
T	temperature at various points	K or ℃
TIT	Inlet temperature of the gas turbine	K or ℃
v	specific volume	m^3/kg
V	velocity	m/s
$\dot{W}$	Work rate	kW
z	Height	m

CCPP	Combined Cycle Power Plants	-
HRSG	Heat Recovery Steam Generator	-
r_p	pressure ratio	-
AFR	Air Fuel Ratio	-

Greek Symbols

η	Efficiency	
ρ	Density	Kg/m^3
γ_a	Heat capacity Ratio of air	
γ_g	Heat capacity Ratio of exhaust gases	

Subscripts

CCGT	Combined cycle Gas turbine
CV	Control Volume
w	Water
c	Compressor
p	pump
i	inlet
e	outlet
a	air
f	fuel
evap	Evaporator
econ	Economiser
sup	Superheater
sat	Saturated
cw	Cooling water
gt	Gas turbine

st	Steam turbine
ip	Intermediate pressure
loss	Losses
lp	Low pressure
net	Net
out	Outlet
surr	Surrounding
sys	system
th	Thermal
comb	combustion

Superscripts

ΔT	Temperature difference
ΔP	Pressure difference

CHAPTER 1

1.Introduction

Optimization of energy conversion systems becomes more important due to limits of fossil fuels and the environmental impact during their use. The use of vitality is discovered wherever in an assortment of utilizations from warming and cooling to atomic power plants. For a considerable length of time, the reaction to the constantly developing requirement for electric era limit was to fabricate another steam control plant, one not altogether different from the past one.

The energy transformation specialist is confronted with an assortment of issues today: rising innovations, changing social and mechanical atmosphere in which a differing qualities of methodologies is probably going to be acknowledged. Some essential attributes of new power activities are low capital and working costs, capacity to work with an assortment of powers and with high resilience to fuel fluctuation, short development time, low outflow of poisons, attractive or possibly inactive and effectively dispensable waste items, and high proficiency, viability, fund capacity, and dependability.

Another key problem facing the energy conversion engineer is the finiteness of natural resources critically important for human beings (such as natural gas and oil) in the world and ever-increasing energy demands by developing countries. Conceivably future power plants ought to use coal and atomic vitality to spare the flammable gas and oil for mechanical sustain stocks and other more basic future needs. Then again, difficult issues exist concerning use of common assets. A significant part of the promptly accessible coal has unsuitably high sulfur, which altogether debases nature when discharged from power plant stacks in untreated burning items. The outstanding issue of corrosive rain has been ascribed to outflows from coal-consuming force plants. In addition, there are economic problems. As it becomes harder to exploit fossil fuels a deposit in the world, the price of energy is increasing, coupled with higher demand due to increased technology implementation and population. Moreover, replacement energies (renewable energies) are economically less efficient than fossil fuels.

Progressively, the new option arrangements may appear as repowering the old plant to expand productivity, achieving contamination gauges, and limiting the money related effect of taking care of new influence requests. The change of the effectiveness of energy plants that utilization ordinary cycles is generally transformative in nature, by ethicalness of high-temperature impediments and advances in materials. Thus, just slow upgrades in productivity can be normal. Then again, critical changes in efficiency can once in a while be gotten by joining traditional cycles in suitable ways. Such power plants are alluded to as combined cycle plants. It is clear from the investigation of the Rankine and Brayton cycles, and truth be told, all heat engines, that the dismissal of a lot of thermal energy to the surroundings goes with the creation of valuable power. This heat dismissal can't be dispensed with; however, it can be decreased by enhancing “the thermal efficiency of the cycle”.

Due to the mentioned problems, it turns out to be progressively critical to comprehend the components that debase energy and resources. Also, creating deliberate methodologies is imperative for enhancing the outline of energy systems and lessening the effect on the earth; in such manner, progressive energy problems have empowered the investigation of discovering more productive courses for the utilization of the accessible energy in fuels. This means that the optimization of power generation systems becomes one of the most important subjects in the energy-engineering field. Recent thermoeconomic analysis and optimization of thermal systems became the key solution in providing a better system in both optimal energy consumption and optimal system configuration. Classical thermodynamics gives the idea of energy, energy exchange of heat and work, energy balance, entropy, and entropy and estimations of thermodynamic properties at equilibrium. The second law of thermodynamics improves an energy balance by computing the genuine thermodynamic estimation of an energy transport and real thermodynamic wasteful aspects and losses from the procedure and system.

The expression "thermoeconomic" was formally used to demonstrate a proper blend of exergetic and economic investigation in which the cost was doled out to the exergy (not the economy) substance of an energy carrier (exergy costing), in parallel. Be that as it may, the expression "thermoeconomic examination" was utilized by others to report conventional thermodynamic investigations construct just in light of the first law of thermodynamics and economic investigations, led independently from the thermodynamic ones and without the thought of exergy or exergy costing. However, “thermos”, is a derivative of the Greek word for heat and it is used in most major languages. Thus, thermoeconomic does not imply exergy costing or exergy economics, but a combination of heat and economics.

Along with the thermodynamic analysis, economic analysis gives the information regarding fixed cost e.g. investment cost, running cost, operating and maintenance cost. In most of the cases, the overall cost of the system will increase with the increase in the system exergetic efficiency and capacity. Thus, thermodynamic improvement in a system is accompanied by an increase in the economic cost. Therefore, the system should be optimized between these two conflicting requirements. In this regard, thermoeconomic analysis evolved and joined thermodynamic and economic parameters to one common platform and now combines thermodynamic analysis with economic analysis. As discussed above, exergy analysis is preferred for thermodynamic analysis; the newly evolved field is called exergoeconomic analysis.

The exergoeconomic methods help in the system improvement using thermodynamic as well as economic points of view, by simultaneous modeling of thermodynamic and economic aspects of the system and its components. These methods are based on optimization techniques, which search for all possible solutions for the optimum design and operation of the system and its components. Just like the exergoeconomic analysis, exergoeconomic optimization combines thermodynamic and economic aspects. For thermodynamic optimization based on exergetic consideration, the exergy destruction method is identified as a methodology.

Thus, the objective of thermoeconomic is to obtain the compromise between these two competing objectives. In this methodology, appropriate costs are assigned to the thermodynamic inefficiencies of the system components through some meaningful fuel-product definition. For maximum exergetic efficiencies, these costs need to be minimized.

1.1The Aim of the Project

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

a) Predict the cost effectiveness of a combined cycle gas turbine (CCGT) power plant.

b) Provide information about the energy losses along with their location.

c) Suggest ways of improving the cost effectiveness from both thermodynamic and economic points of view.

d) Find the optimal realistic values of operating parameters, which gives the maximum possible power output, efficiency, and annual cash flow. Additionally, is would be possible to calculate minimum possible exergy destructions, cost per unit of generated electricity, and purchase investment cost.

1.2Outline of the Thesis

The thesis is divided into seven chapters. In this chapter, the general concept of Thermooeconomic effect of air fuel ratio on CCPP at different pressure ratios is introduced. The importance of the Thermoeconomic effect and the objective of the study are briefly discussed.

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

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

Chapter 4 deals with the theoretical background of energy analysis, and Thermoeconomic optimization, with the expressions and equations used in the mathematical model. It also explains the optimization technique used in this paper.

Chapter 5 is the base plants modeling chapter and it contains a detailed description of the plant, and energy exergy economic analysis. It presents the mathematical procedure of the solution for the optimization problem.

Chapter 6 presents the results of each step of the methodology. In addition, it discusses the effect of the operating parameter on the performance parameters. The comparisons between the cases were also presented.

Chapter 7 concludes the study results.

CHAPTER 2

2.LITERATURE SURVEY

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

2.1Review of Analysis and Optimization of Topping Cycle

2.1.1Thermodynamic Analysis and Optimization

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

2.1.1.1Effect of Ambient Conditions

One of the factors that affect gas turbine performance is the ambient conditions, mainly ambient temperature, atmospheric pressure, and the relative humidity of air. These parameters affect the generated electric power and the heat-rate during operation. The location of power plant plays a major role on its performance. The atmospheric air, which enters the compressor, becomes hotter after compression and it is directed to a combustion chamber. Several authors reported the effect of ambient temperature: Arora and Rai [4]**,** Ibrahim et al [5], Ameri and Hejazi [6], Boonnasa et al [7] and Hosseini et al [8].

Arora and Rai [4] 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%.

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

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

2.1.1.2Effect of Compressor Pressure Ratio

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

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

2.1.1.3Effect of Turbine Inlet Temperature

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

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

Khaliq and Kaushik [10] and **Khaliq** [12] reported in their detailed analyses that the exergy destruction in the combustion chamber increases with the cycle temperature ratio, and the second-law efficiency of the primary combustor behaves in reverse from the second-law analysis. Increasing the maximum cycle temperature gives a significant improvement in both efficiency and specific work-output. The study also concludes that the efficiency reduces rapidly with a reduction in the TIT.

2.1.2Thermoeconomic Optimization

If the sole objective of a CCGT design were to maximize the thermodynamic efficiency, its total cost would be very high. Therefore, the design of a modern power plant means a product with a low investment cost and high efficiency. Thermoeconomic analysis represents a very important tool for the thermal systems designer to determine the optimal configuration for a new system or plan changes in an existing. The thermoeconomic study was very important in order to get a compromise between plant efficiency and costs.

Kaviri et al [13] show that increase in the compressor pressure ratio decreases the cost of exergy destruction. The reason is that by increasing the compressor ratio, the outlet temperature increases as well. Therefore, the temperature difference decreases. Because the cost of exergy destruction is a direct function of exergy destruction, it leads to a decrease in the cost of exergy destruction. As the compression ratio increases, the air exiting the compressors is hotter, therefore less fuel is required (lowering the air fuel ratio) to reach the desired turbine inlet temperature in a fixed gas flow to the gas turbine. The work required in the compressor and the power output of the gas turbine steadily increases with compression ratio, and then causes decreases in the exhaust gases temperature. This lower gas temperature causes less steam to be produced in the HRSG, therefore lowering the outputs of the steam cycle. It is noticed that the total power output increases with compression ratio. However, the variation of the total power output is minor at the lower compression ratio while it is significant at the higher compression ratio for all gas turbine configurations.

2.2Review of Analysis and Optimization of Bottoming Cycle

2.2.1Thermodynamic Analysis and Optimization

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

Mohagheghi and Shayegan [14] 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 [15] 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 [16] 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 [17] 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 [24] 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.

2.2.2Thermoeconomic Analysis

As we mentioned, the combined cycle gas turbine power plants are thermodynamically attractive. Thermodynamics plays an important role in selection of the type of power plan **Kamate and Gangavati** [18], but thermodynamics is not the only criterion for decision. Other factors, such as price, environmental impact, fuel availability are also important

Kehlihofer [19]. The most important part of a CCPP is the heat recovery steam generator. Therefore, the optimal design of HRSG in CCPPs is an important subject due to the increase in fuel prices and decrease in fossil fuel resources.

Alus and Petrović [20] performed an optimization of a triple pressure CCGT. The objective of their work was developing a new system for optimization of parameters for CCGT with triple-pressure heat recovery steam generator. The objective of the thermodynamic optimization is to enhance the efficiency of the CCGT and to maximize the power production in the steam cycle (steam turbine gross power). Improvement of the efficiency of the CCGT plants was achieved through optimization of the operating parameters: temperature difference between the gas and steam pinch point (PP) and the steam pressure in the HRSG. The aim of the thermoeconomic optimization was to minimize the production costs per unit of the generated electricity, optimization was to minimize the production cost of electricity in the CCGT power plant based on energetic and economic analysis.

Casarosa et al [21] minimized the total cost of the exergy losses of the HRSG for a combined cycle using the Simplex method. The objective function was defined as the total installed cost of the HRSG and the cost of the increased fuel consumption when the area of the HRSG was reduced.

Ahmadi and Dincer [22] performed the thermodynamic analysis and thermoeconomic optimization of a dual pressure combined cycle power plant with a supplementary firing unit. They conducted an exergy and exergoeconomic analyses for the power plant. The design parameters of this study were compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, gas turbine inlet temperature, duct burner mass flow rate, high pressure stream, low pressure stream, high pressure main steam temperature, low pressure steam temperature, high pressure pinch point temperature difference, low pressure pinch point temperature difference, condenser pressure, steam turbine isentropic efficiency, and pump isentropic efficiency. They introduced an objective function, a new objective function, representing the total cost of the plant (in terms of dollar per second) defined as the sum of the operating cost related to the fuel consumption and the capital investment for equipment purchase and maintenance costs. The optimum key variables were obtained by minimizing the objective function using a generic algorithm. The optimum design parameters obtained for the plant showed a trade-off between the thermodynamic and economic optimal designs. The sensitivity analysis was also performed. Two factors were considered: unit cost of fuel, and net output power of the combined cycle power plant. They concluded that by increasing the fuel price, the optimized decision variables in the thermoeconomic design tend to reach those of the thermodynamic optimum design.

Behbahani-nia et al [23] presented an exergy based thermoeconomic method, which was applied to find the optimum values of design parameters for a single pressure HRSG in combined cycle power plants. The design variables optimized in this work were pinch point and gas side velocity. Optimization was performed by being based on two different objective functions. The first function was the thermodynamic (the summation of exergy loss due to an outflow of hot gas escaping from the HRSG through stack, and exergy destruction due to internal irreversibility inside the HRSG). The second function was a thermoeconomic objective function (the summation of exergy loss and destruction in terms of expenses including the cost of fuel and electricity, and the capital cost of HRSG). They investigated the effects of pinch point and gas-side velocity on the components of objective functions. The study concluded that a considerable amount of exergy is destroyed due to gas pressure drop, especially when pinch point is very close to zero.

Sanjay [11] investigated the effect of HRSG configuration on exergy destruction of bottoming cycle components and concluded that the distribution of exergy destruction is sensitive to a type of bottoming cycle configuration. He found that the best utilization of heat energy in bottoming steam cycle is exhibited in the case of triple pressure reheat configuration. In all bottoming cycle components (HRSG, Steam turbine, and Condenser), it was observed that component-wise exergy destruction is lower in reheated configuration with respect to the same configuration without reheat.

Ghazi et al [24] carried out a thermo-economic modeling and optimization method to obtain the optimum values of design parameters (high and low drum pressures, steam mass flow rates, high pressure and low pressure pinch point temperature differences, and the duct burner fuel consumption flow rate) for a dual pressure HRSG. They performed the complete sensitivity analysis of changes in inlet gas temperature entering the HRSG and exergy unit cost. Total cost per unit of produced steam exergy was defined as the objective function. They found that at higher inlet gas enthalpy the required heat transfer surface area (capital cost) increases.

Hajabdollahi et al [25] modeled an HRSG with a typical geometry and a number of pressure levels used at CCPPs, and developed a thermodynamic model and thermoeconomic optimization. They conducted exergoeconomic analysis and multi-optimization of an HRSG through energy and exergy, and compared their results with data provided from a power plant situated near the Caspian Sea in Iran. They introduced a new objective function (the total cost per unit of steam produced exergy). Then, optimum design parameters were selected when objective function was minimized while HRSG exergy efficiency was maximized. Authors summarized that an increase in high and low-pressure drums increases exergy efficiency, while an increase in pinch point decreases exergy efficiency. Additionally, an increase in the HRSG inlet gas enthalpy results in an increase of the exergy efficiency.

Naemi et al [26] developed the thermodynamic model of a dual pressure HRSG coupled with a heavy-duty gas turbine. They investigated thermodynamic and thermoeconomic analyses to achieve the optimum operating parameters of a dual pressure heat recovery steam generator, and computed exergy waste and exergy destruction for different pinch points. They discussed the effects of non-dimensional parameters on the HRSG performance. They also investigated optimum design of HRSG regarding financial considerations, and performed a sensitivity analysis.

Najjar [27] described that the efficiency of a gas turbine engine is relatively low at design point and it deteriorates further at part load and at off-design high ambient temperatures. His work comprises of the study of adding an inlet air pre cooler driven by the tail-end heat recovered from the engine exhaust gases. A heat recovery boiler was used to partly recover the exhaust heat. The performance of this combined system, namely power, efficiency, and specific fuel consumption was studied and compared with the simple cycle. The variables in this parametric study were mainly compressor pressure ratio, turbine inlet temperature, and ambient temperature. Results show that the combined system achieves gains in power. The performance of the combined system showed less sensitivity to variations in operating variables. Thermoeconomic evaluation shows that the combined system is viable.

2.3Review of Optimization of whole CCGT

One of these alternative methods is optimizing the combined cycle, which has been the subject of many investigations. Some investigators focused on optimizing the thermal performance: Franco and Casarosa [30], Valdes and Rapun [31], Bassily [32], and Bassily [33]; whereas other investigators optimized an objective function of the net revenue or total cost: Valdés et al [29], and Casarosa et al [21].

Tyagi and Khan [28] studied the effects of gas turbine exhaust temperature, stack temperature and ambient temperature on the overall efficiency of combine cycle power plant keeping the gas turbine efficiency as well as steam turbine efficiency constant. They concluded that the stack temperature should be minimum and gas turbine exhaust temperature should be maximum. Out of these three variables i.e. turbine exhaust temperature, stack temperature and ambient temperature, the dominating factor of increasing the overall efficiency of the combine cycle power plant is the stack temperature.

Valdés et al [29] 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 [32] 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 modeled 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 [33].

Boonnasa et al [7] 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.

2.3.1Thermoeconomic Analysis and Optimization Method

Exergoeconomic methods can be classified in two groups: algebraic and calculus methods. Algebraic methods use algebraic balance equations, always require auxiliary cost equations for each component, focus essentially on the cost formation process, and determine average costs. Many researchers have conducted algebraic methods for thermal system optimization:

Lozano and Valero [34], Kim et al [38], Kwon et al [39], Tsatsaronis [43] and Vieira et al [44].

Lazzaretto and Tsatsaronis [35] proposed a methodology for defining and calculating exergetic efficiencies and exergy related costs in thermal systems. It was based on the SPECO. Separate forms of exergy and costs associated with these exergy streams were used to define exergetic efficiencies in a detailed manner. It was concluded that the SPECO was a powerful approach to express the validation of the calculated cost values.

Kanoglu et al [36] developed methodology for calculating exergy flows, cost formation, and allocation within high temperature steam electrolysis system. They used specific exergy costing methodology while applying exergetic fuel and product approaches to obtain the cost balance equations. They examined exergy efficiency, exergy destruction rates, exergy loss–exergy destruction ratio, capital investment, operating, maintenance costs, and exergoeconomic factor. The capital investment cost, the operating and maintenance costs, and the total cost of the system were calculated as 422.2, 2.04 and 424.3 €/kWh, respectively. The cost distribution among the components was also determined. The exergetic costs of the steam were 0.000509, 0.000544 and 0.000574 €/kWh at the outdoor temperatures of 25 °C, 11°C and −1°C, respectively.

Orhan and Dincer [37] studied the minimization cost of a copper–chlorine (Cu-Cl) thermo-chemical cycle for hydrogen production. The specific exergy costing method was used to determine changes in design parameters of the cycle, which could improve the cost effectiveness of overall system. It was found that the cost rate of the exergy destruction took the values between $1 and $15 per kg hydrogen. The exergoeconomic factors were calculated between 0.5 and 0.02.

Kim et al [38] introduced modified productive structure analysis (MOPSA) method where an exergy costing method is used without flow-stream cost calculations.

For the entire system a set of equations for the unit exergy costs are obtained by assigning a unit exergy cost for the cost balance equation for each component.

Kwon et al [39] compared specific exergy cost method and modified productive structure analysis methods by applying them to the CGAM problem.

Calculus methods are built on differential equations. These methods are generally based on the Lagrange multipliers technique and are considered subjective with regard to the mathematical description of the function of each component in the system. A particular difficulty in the application of calculus methods to complex systems is the fact that the Lagrange multipliers vary from iteration to iteration when component thermoeconomic isolation is not achieved. This problem has led to the development of the Thermoeconomic Functional Analysis (TFA)

Frangopoulos [40]. Calculus method use differential equations, such that the system cost flows are obtained in conjunction with optimization procedures based on the method of Lagrange multipliers, and determine marginal costs El-Sayed and Gaggioli [41], and Gaggioli and El-Sayed [42].

CHAPTER 3

3.Introduction to combined cycle power plant

To begin with, eras of combined cycle power generations systems introduced amid the 1950s and mid-1960s included conventional fired boilers. These systems were adjustments of conventional steam plants with the gas turbine exhaust gas filling in as combustion air for the boiler. The effectiveness of this kind of combined cycle was roughly 5–6% higher than that of a comparative traditional steam plant. These systems could financially use exposed tubes in the boiler because of the high mean temperature difference between the combustion products and the water/steam. The second generation, which combined cycle system with finned tube boilers, entered service in1959. Amid the 1960s, the utilization of the heat recuperation type of combined cycle systems turned out to be more pervasive. Its underlying application was in power and heat applications where its energy to-heat proportion was more great. Furthermore, few the heat recuperation type combined cycles were introduced in utility power generation applications amid the 1960s. The utilization of these frameworks in the 1980s built up the heat recuperation feed water heating combined cycle as a developed innovation for base load and mid-extend benefit.

By 1970, there were various plants in operation. All through the 1980s, the innovation created with bigger gas turbines and the presentation of pre-blended burning for low emissions. Around 1990 the net plant productivity of combined cycles passed half (LHV). All through the 1990s, a substantial number of combined cycle power plants were constructed, and huge numbers of them in base load operation. Around 1995 another era of vast gas turbines went to the market, bringing the block size of combined cycle power yield to 350-400MW and productivity up to 57-58%. Starting at 2011, the power yield was expanded to around 570MW and efficiency near 61%.

Table 3-1 shows the energy utilization for a typical combined cycle plant. The gas turbine may regularly change over 36% of the fuel energy into power, leaving 63% as heat gone to the HRSG from the exhaust of the gas turbine (run of the mill mechanical electrical and heat losses in the GT represent 1%).

Table 3-1: Typical modern day combined cycle performance

COMBINED CYCLE PERFORMANCE				
	% OF FUEL INPUT			
Fuel Input LHV	**100**			
Gas Turbine Power	36			
Gas Turbine Losses	1			
Gas Turbine Exhaust Heat	**63**			
Stack Loss		22		
Input to Steam		**41**		
Steam Turbine Power			19	
Steam Turbine Losses			1	
Heat to Condenser			**21**	
Gross Electric Power				**55**
Auxiliaries Power				2
Total Net Power and Efficiency				**53**

The HRSG catches roughly 66% of the gas turbine exhaust heat with the staying third being lost in the exit stack. At long last, 19% of the fuel input is changed over into power by means of the steam turbine with 1% lost in the turbine and 21% of the fuel energy lost in the spent steam, which is sent to the condenser. The consolidated gross energy of gas and steam turbines likens to 55% (LHV) of the fuel energy. Plant auxiliary represents ~2% of the fuel input, at last, leaving 53% as net yield combined cycle efficiency. In this way, the principle legitimization for using HRSGs inside utility power plants lays free advantage from the superposition of the gas turbine Brayton cycle over the steam turbine Rankine cycle which brings about an improved general thermal efficiency.

3.1Description of the Main Components of Combined cycle power plant

3.1.1Compressor

Gas turbine compressors, by and large, utilize axial stage compressors versus centrifugal stage compressors. Axial flow stream compressors coordinate wind stream parallel and in-line to the axis of rotation, though centrifugal compressors for the most part direct air stream radially far from the hub.

Large pressure ratios guarantee optimal gas turbine yield, and albeit axial stage compressors don't offer a similar pressure ratio as a centrifugal stage compressor of a similar diameter across, multistage axial compressors offer the open door for substantially more prominent pressure ratios by and large. The improved pressure ratios offered by multistage axial compressors empower bigger mass flow rates and along these lines more power yield.

One critical viewpoint that must be considered is the elevation and nearby atmosphere

The main role of the compressor is to quicken and coordinate extensive mass flows of air into the gas turbine. The compressor is involved a few stages of rotating blades on plates and on stationary vanes. Each row of rotating blades quickens air towards its trailing edges and its stage's vanes. The stage's vanes slow down the air and guide it to the following stage of blades.

After the air is quickened and compressed through each stage of the compressor modules, it enters the diffuser area. Now, the air has achieved its greatest speed and most extreme compressor pressure. The diffuser gives an expanding cross sectional range with a specific end goal to moderate the air's speed and increment its static pressure. At the diffuser outlet, the static pressure has achieved its most extreme and enters the combustor module.

of the gas turbine. Since air density diminishes with expanding height, expanding temperature, and expanding humidity, surrounding conditions can assume a noteworthy part in a gas turbine's energy yield. At high elevations, high temperature, or high humidity, the weight of a given volume of air will be littler, which compares to a littler measure of fuel necessity and bringing about less power yield. [45]

3.1.2Combustor

The combustor module is involved a few segments, basically combustion chambers, igniter plugs, and fuel nozzles. Combustors consume a fuel and air blend, conveying the burning products to the turbine bay at its design parameters.

Amid the beginning cycle, start happens through igniter plugs connects situated to the burning liners close to the fuel nozzles. The flame starts in the burning can nearest to the igniter and spreads to the next combustion cans.

Combustors utilize three fundamental sorts of ignition chambers: annular, can, and can-annular. Fuel is sprayed into the ignition chambers with either a simple type fuel nozzle, which conveys either vaporous or liquid fuel or double fuel nozzles, which conveys gas or liquid at various circumstances all the while. A few gas turbines are even considered "bi-fuel" in which a gas and liquid fuel blend are ignited at the same time.

Whirled vanes are utilized to blend air with the fuel from the fuel nozzles. Air that flows over the twirled vanes is viewed as primary air and constitutes around 25% of the aggregate air devoured by the gas turbine. The rest of the air is viewed as secondary air and is utilized to cool the burning gasses to temperatures reasonable for turbine entering conditions. This likewise constrains the measure of NOx development. [45]

3.1.3Gas Turbine

A gas turbine is a machine conveying mechanical power or thrust. It does this utilizing a vaporous working liquid. The mechanical power created can be utilized by, for instance, in power production. The active vaporous fluid can be utilized to create thrust or to produce power. In the gas turbine, there is a persistent flow of the working liquid. This working fluid is at first compressed in the compressor. It is then heated in the burning chamber.

At last, it experiences the turbine as found in Figure 3-1. The turbine changes over the energy of the gas into mechanical work. Some portion of this work is utilized to drive the compressor. The rest of, "valuable power", is utilized as the yield shaft energy to turn an energy converter device, for example, an electrical generator. In the power generation field, the gas turbine can be utilized as a remain solitary unit or with combined cycle power plants. Power producing gas turbines are typically open cycle worked. The gas turbine execution relies on upon the execution of its parts i.e. compressor, burning chamber, and turbine.

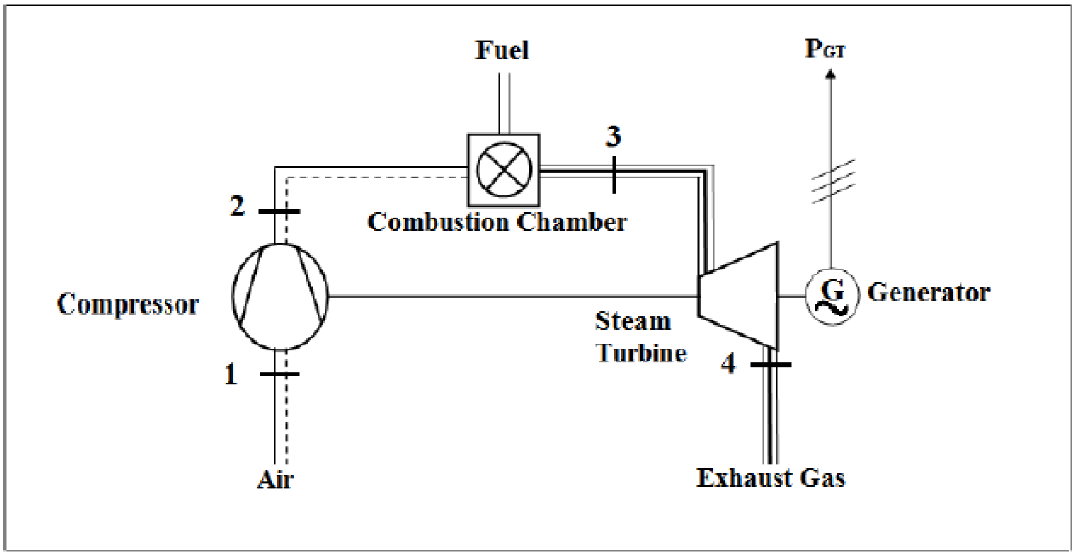

Figure 3-1: The schematic gas turbine

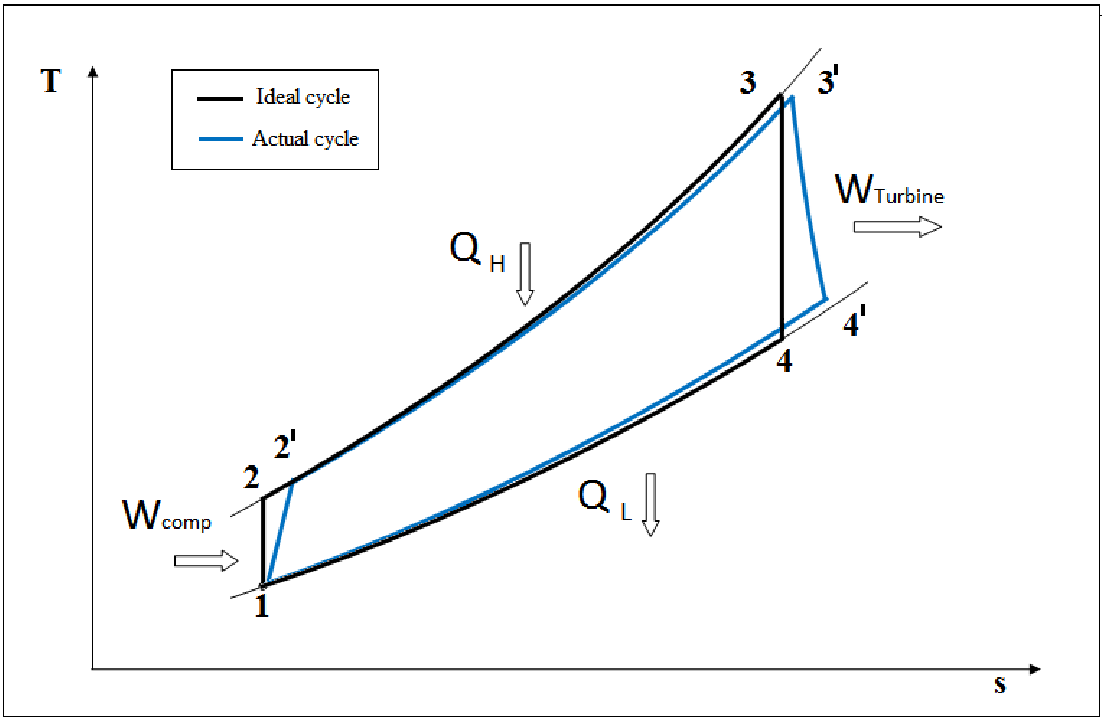

Figure 3-2: Brayton cycle

The thermodynamic cycle of a gas turbine is known as the Brayton cycle. Gas turbines more often work on an open cycle. As found in Figure 3-2, the air is first compressed in the compressor, the air drawn at surrounding conditions into the compressor admission, where the compressor pressurizes the air up to P2, expanding both pressure and temperature at the costs of utilizing compression work (W_{comp}) which is provided by the turbine itself. The high-pressure air continues into the ignition chamber, where the fuel is ignited at a consistent pressure. This additional heat ($\dot{Q}_h$) raises the temperature from T_2 to the turbine entering temperature T_3 (most noteworthy cycle temperature). The subsequent high-temperature gasses then enter the turbine, where they extend to the surrounding pressure while delivering power ($W_{turbine}$) enough to drive the compressor and create net shaft work (W). At long last, heat (Q_l) is rejected to the surrounding. The majority of the gas turbines in power production utilize axial flow compressors. The exhaust gases leaving the turbine are thrown out (not re-circulates), causing the cycle to be classified as an open cycle. The Compression ratio and turbine inlet temperature are important parameters.

3.1.4Heat Recovery Steam Generator

In the present innovation, the most proficient energy transformation systems to deliver electrical and heat energy are the combined cycle power plants. In an average CCPP, exhaust gas heat from the gas turbine GT is recuperated in a heat recovery steam generator to produce steam in the steam cycle. HRSG performance has a large impact on the overall performance of a combined-cycle power plant. Steam generated in HRSG with different pressure levels depend on the design. HRSG consist of three heat exchanger packages (economizer, evaporator, and superheater) Figure 3-3. Combustion gases enter superheater, evaporator, and economizer package respectively. The heat recovery from gas side to the water-steam is achieved in three steps:

In the economizer, the feed water is heated to a temperature near its saturation temperature.

In the evaporator, the water evaporates at a consistent temperature and pressure and saturated steam.

In the superheater, the high esteem heat from the exhaust is utilized for superheating the steam produced in the evaporator. Superheated steam is bolstered to the steam turbine.

Classification of HRSG: Heat recovery steam generator can be classified according to:

a. The generated steam pressure; single pressure Figure 3-3, dual pressure Figure 3-4, and triple pressure Figure 3-5

b. The type of circulation system being used Figure 3-9:

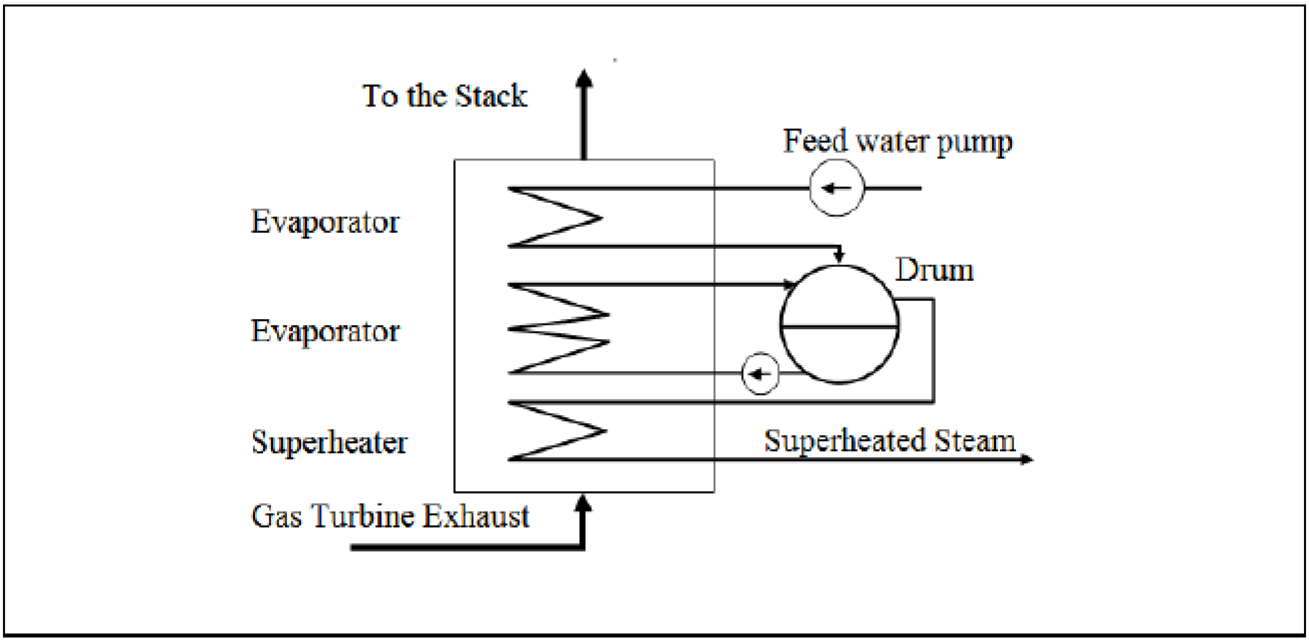

Figure 3-3 Single pressure HRSG

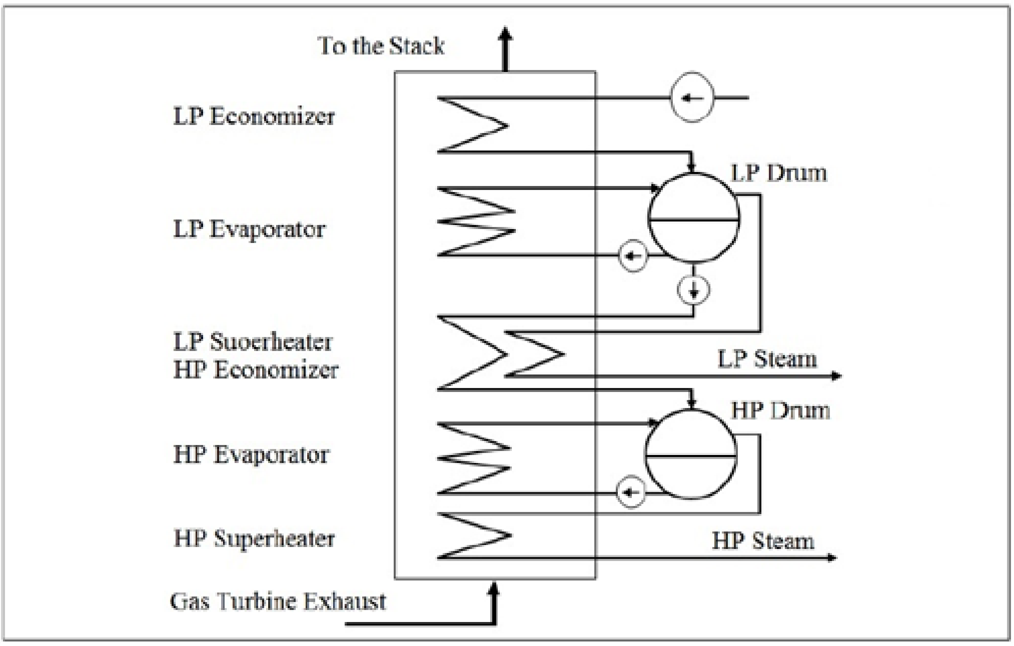

Figure 3-4 Dual pressure HRSG

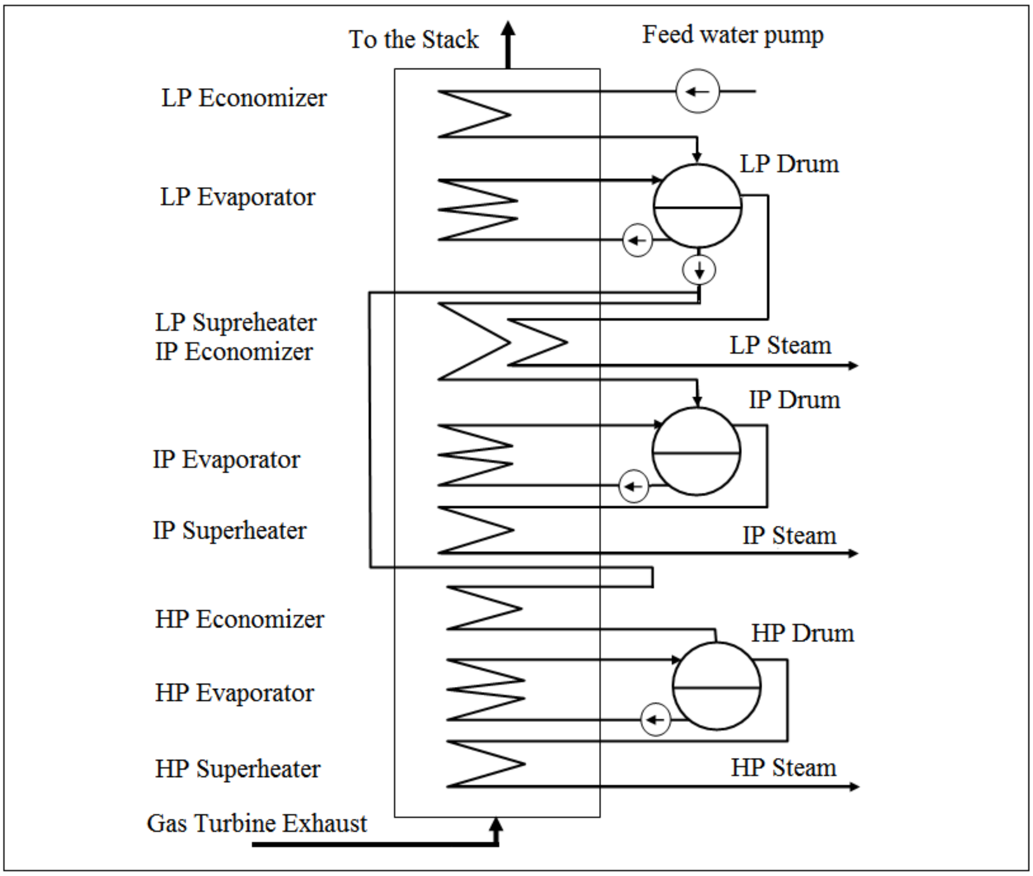

Figure 3-5 Triple pressure HRSG

Natural circulation regularly comprises of “vertical tubes and horizontal tubes” with flow arrangement.

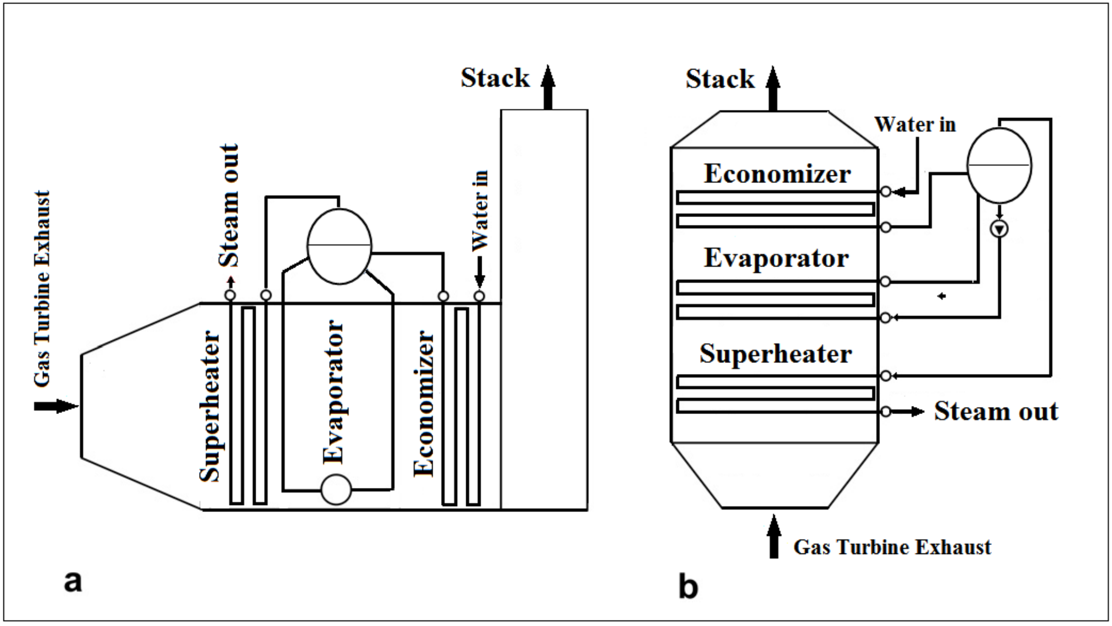

Figure 3-6 HRSG (a) Natural circulation, (b) Forced circulation

Circulation kept up by the density differences between cold water in lower chamber and hot steam-water blend in evaporator tubes are appeared in Figure 3-6a.

Forced circulation; HRSGs are described by horizontal tubes with vertical gas stream flow and utilize pumps to circle steam-water blend (Figure 3-6b).

3.1.4.1Important HRSG Performance and Design Parameters

a) HRSG Main Design Parameters

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

a) Pinch point temperature: pinch point temperature is the difference between the saturation temperature of water and the gas temperature of the gas leaving the evaporator (Figure 3-7).

b) Economizer approach temperatures (approach point): the approach point is the difference between the temperature of saturated steam and the temperature of the water entering the evaporator (Figure 3-7). Selection of these two variables also affects the size of the superheater, the evaporator, and the economizer. The smaller temperature difference means that the surface area required to produce the same heat transfer will be much greater. The direct consequence is that more material is used and hence capital cost is higher.

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

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

e) Stack outlet temperatures: the temperature of the gases that leave the HRSG to the atmosphere (Figure 3-7).

f) Allowable backpressure. The HRSG cross sectional area significantly influences the gas turbine backpressure. Smaller, more compact HRSGs require higher gas turbine backpressures to drive through the flue gas, however, while the size reduction may reduce HRSG cost, the requirement

g) to provide a higher pressure at the turbine exit has a detrimental effect on

h) gas turbine efficiency. The typical values of gas turbine back-pressures are

i) 2.5 to 3.7 kPa in most units [50].

3.1.4.2T-Q Diagram of HRSG

The T-Q diagram shows profiles for the “heat transfer process” between exhaust gas and “water/steam, using temperature on the” ordinate axis and heat transferred on the abscissa axis.

The use of T-Q diagram is crucial in understanding and designing combined cycles. Figure 3-7 shows the T-Q diagram for a single-pressure combined cycle. The smallest temperature difference in the HRSG is called the pinch-point, and it is located on the cold side of the evaporator. The upper line, with an almost constant slope, represents the temperature profile of the flue-gas, and the lower line represents the temperature of the water/steam. The HRSG of a single-pressure combined cycle consists of three different sections.

First section HRSG: starting at the lowest temperature, the first section is called the economizer, and is the place where liquid water is heated to the saturation temperature. To avoid evaporation, which could cause steam blockage that may result in “water hammering” in the economizer, the outlet temperature is always kept a few degrees below the saturated state. This temperature difference is called the approach point.

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

Third section HRSG is the superheater where the evaporated steam is superheated.

The relation between temperature and heat can be described by:

$$\dot{Q} = \dot{m}c_p\Delta t \qquad 3\text{-}1$$

This equation (3-1) is valid when the working medium does not undergo a phase transition. In the HRSG, a phase transition from water to steam occurs in the evaporator [51], which means that equation (3-2) must be replaced by:

$$\dot{Q} = \dot{m}\Delta h_{evap} \qquad 3\text{-}2$$

where Q the energy is transferred and Δh_{evap} is the evaporation enthalpy. If equation (3-1) is rearranged it can be seen that the slope of the line in the T-Q diagram is inversely proportional to the mass flow and the specific heat. From equation (3-2) it can be understood that the term $\dot{m}\Delta h_{evap}$ is the length of the evaporation line. These relations are central for the analysis of combined cycle power plants.

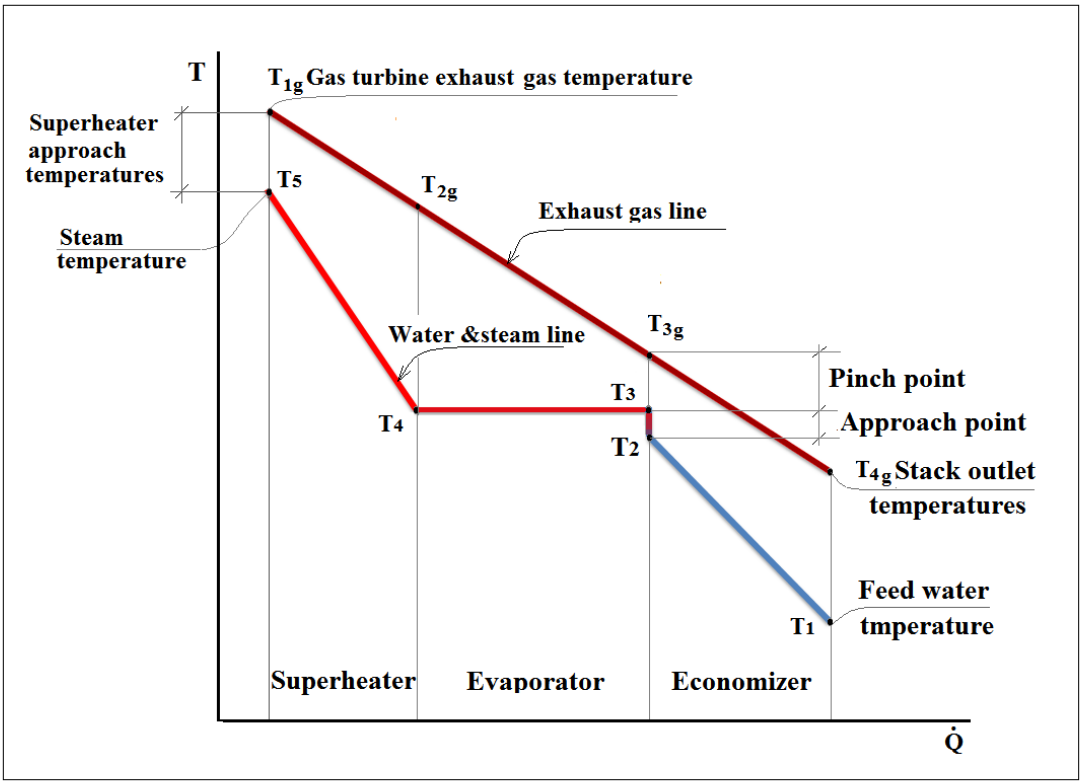

Figure 3-7: T-Q Diagram for a single pressure HRSG

At the beginning of this section it was mentioned that the HRSG should be designed taking both the first and second laws of thermodynamics into consideration.

The first law implies that as much heat as possible should be recovered from the flue gas. The second law, which also embodies a very important factor in the HRSG design, states that the potential or exergy of the flue-gas energy should be utilized as efficiently as possible. In other words, as small amount of entropy as possible should be generated through the process. To evaluate this, an exergy analysis of the system can be performed, which will quantify the deficiencies of the process. If the temperature difference throughout the T-Q diagram is minimized, the process generates a minimum amount of entropy generation (exergy destruction). More details can be found in Section 4.2.4. A reversible or perfect process is one that can return both the system and the surroundings to their initial conditions with no network input. A reversible process will never occur in reality. However, for a heat exchanger, the process is reversible if there is no temperature difference between the hot and the cold sides. Thus, the irreversibility increases with increasing temperature difference. Considering the first and second laws of thermodynamics when designing a

HRSG means a compromise between the following [51]:

- As much energy as possible should be recovered, i.e., the T-Q diagram should be extended as far as possible along the x-axis
- The temperature difference, i.e. the area between the lines in the T-Q diagram, should be minimized.

3.1.5Steam Turbine

The steam turbine is a magnificent prime mover to change over heat energy of steam into mechanical energy. It is one of the notable prime movers, for example, gas engines, diesel engines, gas turbines, jet engines, and so on. All steam based engines, regardless of whether turbines or not, are intended to concentrate energy from high-pressure steam and change it into movement by enabling the steam to flow over a system. For the turbine outlines, steam permitted to extend steadily through more than one arrangement of blades, for accomplishing considerably higher efficiencies contrasted with a solitary step expansion. The steam flows through progressive rings of moving blades on a hub and fixed blades in a casing, delivering simply rotation. At the point when combined with an electric generator, the steam turbine is a standout amongst the most critical methods for delivering electric power on the planet. The current steam turbine may have three phases. The high-pressure segment has small blades. They are little on the grounds that the approaching steam has high energy at high temperature. After the steam goes through the high-pressure segment it is sent back to the reheater. The steam is then sent to the following segment of the turbine, called the intermediate pressure segment. The blades here are bigger than those in the high-pressure segment. In the wake of going through this area, the steam is sent to the low-pressure segment of the turbine. Since the vast majority of the energy was already extracted from the steam, the blades here are the biggest in the turbine. The steam leaves the turbine through the base, where it is consolidated over into water. From that point it is sent back to the boiler drum, to be made into steam once more. The steam turbine is regularly utilized as a part of a consolidated heat and power generation process where the turbine drives a machine in the meantime: steam separated from the machine is utilized to supply locale heating as well as process steam systems.

A solitary unit of steam turbine can create control running from 1 MW to 1000 MW. The thermal efficiency of a present-day steam power plant over 120 MW is as high as 38% to 40%. Water (steam) is the working fluid for most vapor power cycles. Water works over a wide scope of temperatures and pressure have a substantial heat capacity, and it is steady and safe. The energy sources used to create steam incorporate gas, coal, oil, and atomic sources.

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

The most normally utilized vapor power cycle is the Rankine cycle. Despite the fact

Steam Turbine Performance:

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

that a depiction of the Rankine cycle can be found in any thermodynamics reading material. The basic Rankine cycle comprises of four stages. The working fluid is pumped to a high pressure and passed through the boiler drum. The liquid is bubbled at a consistent pressure in the boiler drum after which the high-pressure vapor delivered is passed through a turbine, accordingly removing work from it. The vapor leaving the turbine is consolidated in a condenser by dismissing thermal energy to a cooling liquid. A few adjustments to the Rankine cycle are utilized to accomplish better efficiencies. These incorporate superheating, reheating and recovery. A hefty portion of the difficulties related to the Carnot cycle can be overcome by superheating the steam in the superheater and consolidating it totally in the condenser,

The cycle that outcomes are the Rankine cycle, which is the perfect cycle for vapor power plants. The perfect Rankine cycle does not include any internal irreversibility and comprises of the accompanying four process: isentropic compression in a pump, consistent pressure heat addition in a boiler, isentropic expansion in a turbine, and consistent pressure heat dismissal in a condenser. Every one of the four segments of the

CONDENSER

Steam condensers are devices in which the exhaust steam from the steam engines or steam turbines is condensed by methods for cooling water. The condensate (dense steam) consequently shaped together with the air and other non-condensable gasses is persistently evacuated by pumps or comparative different devices. The essential work of a condenser is to keep up a low back pressure on the exit side of the rotor of the steam turbine. This empowers the steam to extend to a more noteworthy degree which brings about an increase in available energy for changing over into mechanical work.

The secondary requirement of a condenser is to supply to the boiler unadulterated and hot feed water, as the condensed steam which is released from the condenser and, can be utilized over again as feed water for the boiler drum. Hence by fitting a condenser, the thermal efficiency of the steam power plant can be significantly expanded and furthermore the capacity, without expanding size. The condensed steam delivers a fractional vacuum by reason of the colossal decrease in the volume of the low-pressure steam or water vapour.

In the event that a kilogram of dry steam at 1.01325 bar and with a volume of 1.6729 m3, contained in a steam tight vessel, is condensed into water at a temperature of 100°C, the fluid would occupy just 0.001044 m. The volume of steam would be thus $\frac{1}{1644}$ part of the space inside the vessel, and the pressure would fall to 0.2 bar. This means the back pressure in the steam turbine can be lowered from 1.01325 to 0.2 bar or even less. Steam turbines can, in any case, take advantage of low vacuum since they are consistent flow machines and have huge exhausts outlets through which the steam might be released after expansion. Steam turbines may grow steam to a pressure of 25 mm of mercury absolute or even less, i.e. around 0.034 bar.

Rankine cycle is steady state steady-flow devices. The potential and kinetic energy impacts can be disregarded. The boiler drum and the condenser don't include any work, and the pump and the turbine are thought to be isentropic.

Types

Condensers might be arranged comprehensively into two noteworthy groups as indicated by the way in which the cooling water cools and condenses the exhaust steam; these are:

- Jet condensers, in which cooling water comes in direct contact with the exhaust steam and the steam thus is condensed. The cooling water is normally showered into the fumes steam so that fast condensation of the steam happens.
- Surface condensers, in which the cooling water and steam don't really blend the cooling water goes through various tubes while the steam passes over the external surfaces of the tubes.

The most widely recognized is a surface condenser which has the considerable favourable position that the condensate (dense steam) is not thrown to waste but rather is come back to the evaporator through feed water system. A stream condenser is a substantially less complex and less exorbitant mechanical assembly than a surface condenser. The jet condenser ought to be introduced where a shabby wellspring of evaporator feed water is accessible.

FEED PUMP

Boiler feed pumps are also referred to as feed pumps and designed as multistage radial flow pumps. They serve to feed a steam generator such as a boiler or a nuclear reactor or HRSG with a quantity of feed water corresponding to the quantity of steam emitted. Today, all boiler feed pumps are centrifugal pumps. The design of boiler feed pumps regarding power input, material, type of pump and drive is largely governed by the developments which have taken place in power station technology. The trend in fossil-fueled power stations is towards larger and larger power station units (> 1000 MW in 2011). This has led to boiler feed pumps with a drive rating of 30 - 50 MW.

The mass flow rates of centrifugal pumps rose rapidly in conjunction with the rise of unit outputs in power stations. Today's full load feed pumps for conventional 800 to 1100 MW power station units are constructed with four to six stages with stage pressures of up to 80 bar. Feed pumps for 1600 MW nuclear power stations are of the single-stage type.

3.2Description of thermodynamic cycle of Combined Cycle Power Plant

Regular combined cycle power plant comprises of the mix of Brayton and Rankine cycle framing a standout amongst the most proficient cycles utilized for power generation today. In the Brayton Cycle, there is the gas turbine cycle, additionally called the topping cycle and the Rankine Cycle is the steam turbine cycle, likewise depicted as bottoming cycle. The thermal efficiency of the combined cycle plants is to some degree higher today and surpasses 60%.

Gas-turbine cycles ordinarily work at impressively higher temperatures than steam cycles. The greatest liquid temperature at the turbine entry is around 620°C for present day steam power plants, however more than 1425°C for gas-turbine power plants. It is more than 1500°C at the burner exit of turbojet engines. The utilization of higher temperatures in gas turbines was made conceivable by late advancements in cooling the turbine sharp blades and covering the moving blades with high-temperature safe materials, for example, ceramic.

In view of the higher normal temperature at which thermal energy is provided, gas-turbine cycles have a more prominent potential for higher thermal efficiencies. Notwithstanding, the gas-turbine cycles have one natural detriment: the gas leaves the gas turbine at high temperatures (for the most part over 500°C), which deletes any potential picks up in the thermal effectiveness. The circumstance can be enhanced to some degree by utilizing recovery, yet the change is restricted. It bodes well to exploit the exceptionally alluring qualities of the gas-turbine cycle at high temperatures and to utilize the high temperature exit gasses as the energy hotspot for the bottoming cycle, for example, a steam power cycle.

The outcome is a combined gas–steam cycle. In this cycle, energy is recouped from the exhaust gasses by exchanging it to the steam in a heat exchanger that fills in as the boiler.

By and large, (at least one) gas turbine is expected to supply adequate heat to the steam. Also, the steam cycle may include recovery and in addition reheating. energy for the reheating procedure can be provided by consuming some extra fuel in the oxygen-rich exhaust gasses. The gas turbine pipe gas temperature is inside the range 450-650 °C. The energy contained in the pipe gas is a measure of the fuel energy that is not changed

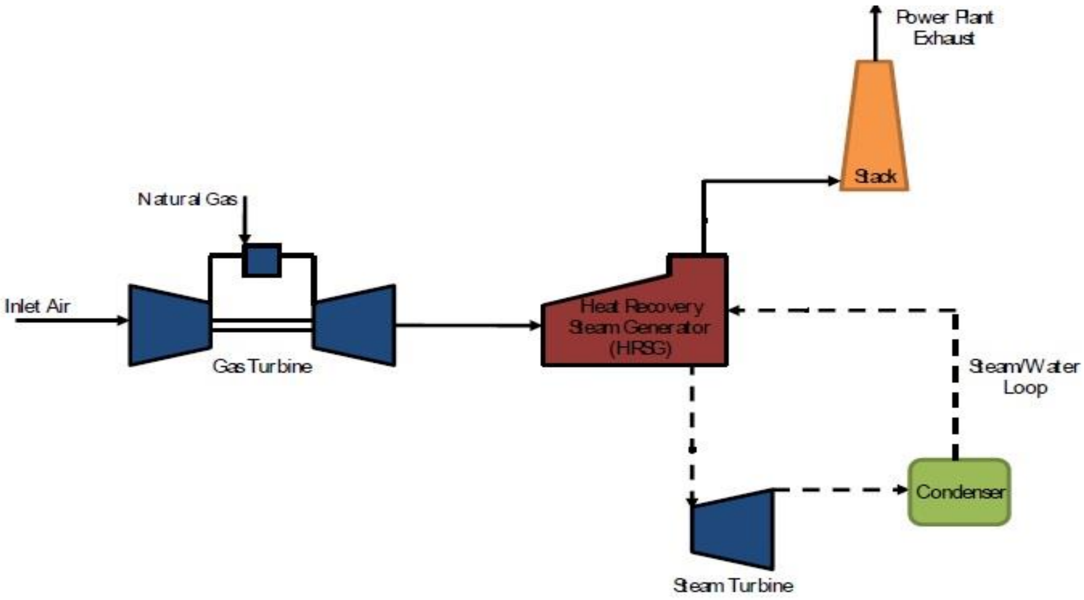

Fig 3-7 Representing Complete Schematic of Combined Cycle Power Plant

over to power by the gas turbine. This energy is utilized to raise steam and to create power by the steam turbine. Contingent on the sort of HRSG the pipe gas temperature is diminished to 80-200 °C, where the lower value is for substantial present day combined cycle consuming a fuel with no or little sulphur. The steam is delivered with a temperature in the range 450-560 °C, and a pressure in the range 30-170 bar.

Steam might be created at various pressure levels. The utilization of super-critical steam pressure (>220.64 bar) was recommended by. Super-critical steam pressure gives the consolidated cycle possibly higher proficiency, yet it relies on upon size of the steam turbine and reliance of pressure on the steam turbine productivity. The cycle working at the higher temperatures is known as the topping cycle, while the cycle reutilizing the energy flux is the bottoming cycle. The Combined Cycle Gas Turbine power plant utilizes the Brayton cycle as the topping cycle while the Rankine cycle goes about as the bottoming cycle.

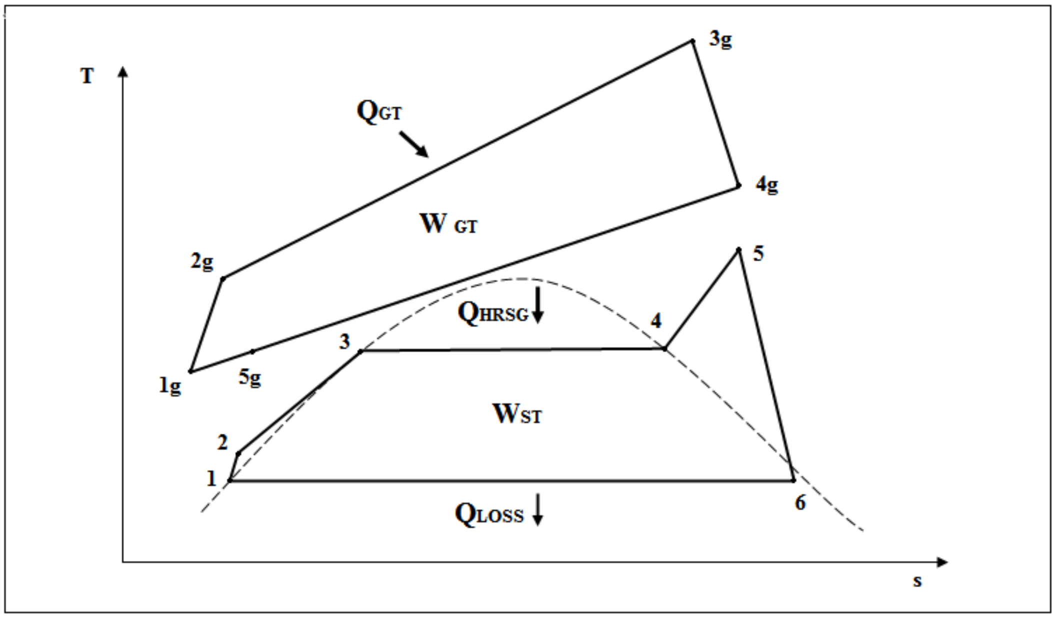

Figure 3-8 T-s Diagram of the Combined Cycle Gas Turbine CCGT Process

CHAPTER 4

4.Thermoeconomic Analysis and – Background

Thermoeconomics investigation is characterized as a branch of engineering that joins energy examination at the system component level into the monetary laws, so as to give valuable data to the engineer or administrator to cost-effectively outline or work the system. It ought to be noticed that this data couldn't be acquired utilizing regular energy or exergy investigation, or potentially economic examination independently.

Thermoeconomics lays on the thought that energy is the main sound reason for doling out fiscal expenses to the associations that a system encounters with its environment and to the sources of thermodynamic wasteful aspects inside it. Thermoeconomics, being a broader term and describing any blend of a thermodynamic and economic examination, may likewise be utilized rather than the term exergoeconomics (yet not the other way around).

The second law of thermodynamics consolidated with economic matters speaks to an effective device for the efficient review and optimizing of energy systems. This blend frames the premise of the generally new field of thermoeconomics

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

4.1Energy Analysis

An energy investigation depends on the first law of thermodynamics. The first law of thermodynamics is all the more usually known as the law of energy protection. The fist and basic law of thermodynamics demonstrate that energy can now be made nor destroyed, and it can just change starting with one form then onto the next. This law characterizes internal energy as a state work, and gives a formal proclamation of the protection of energy. The most common energy systems, such as power generation and refrigeration systems, are open systems (systems in which mass flows through the various components). The typical components of power and refrigeration systems are boilers, turbines, evaporators etc., all of which have inlets and outlets. The expression of the first law of thermodynamics for open systems is:

$$\begin{bmatrix} Rate\ of\ Internal \\ Energy\ change \\ within\ Control\ Volume \end{bmatrix} = \begin{bmatrix} Net\ Rate \\ of\ Heat \\ Addition \end{bmatrix} - \begin{bmatrix} Net\ Rate \\ of\ Heat \\ Work\ out \end{bmatrix} + \begin{bmatrix} Rate\ of \\ Energy\ Addition \\ with\ mass \end{bmatrix} + \begin{bmatrix} Rate\ of \\ Energy\ removal \\ with\ mass \end{bmatrix} \qquad 4\text{-}1$$

The mathematical equation for the first law of thermodynamics for an open system, or any component in an open system, is:

$$\left(\frac{dE}{dtr}\right)_{cv} = \dot{Q} - \dot{W}_{cv} + \sum \dot{m}_i \left(h_i + \frac{1}{2}{V_i}^2 + gz_i\right) - \sum \dot{m}_e \left(h_e + \frac{1}{2}{V_e}^2 + gz_e\right) \qquad 4\text{-}2$$

where $\left(\frac{dE}{dtr}\right)_{cv}$ is the time rate of change of the total energy stored within the given control volume, $\dot{Q}$ is the net rate of heat addition, $\dot{W}_{cv}$ is the net rate of the non-flow work out; and $\sum \dot{m} \left(h + \frac{1}{2}V^2 + gz\right)$ is the rate of the energy h is the specific enthalpy, $\frac{1}{2}V^2$is the specific kinetic energy, and gz is the specific potential energy addition or removal due to mass flowing ($\dot{m}$) into (i) or out of (e) the given control volume. For almost every typical component in an energy system, the velocity and height differences for the working fluid flows can be ignored, so that the rates of energy addition and removal are only associated with the enthalpy, such that:

$$\begin{bmatrix} Rate\ of\ Energy \\ Addition\ with\ Mass \end{bmatrix} = m_i h_i \qquad 4\text{-}3$$

$$\begin{bmatrix} Rate\ of\ Energy \\ Removal\ with\ Mass \end{bmatrix} = m_e h_e \qquad 4\text{-}4$$

An energy investigation is normally utilized as a part of assessing the performance of a segment or a system and can be utilized to decide the first law effectiveness (η_{th}) for a power production cycle.

Notwithstanding, an energy balance gives no data about the course in which procedures can suddenly happen as well as the reversibility of the thermodynamic procedures. The main law can't give data about the powerlessness of any thermodynamic procedure to change over warmth completely into mechanical work, or any understanding into why blends can't suddenly isolate themselves. [53]

4.2Economic Analysis

Preceding the financial treatment, First and Second Law of Thermodynamic examination are made. In the First Law examination, estimations of energy, fuel utilization and productivity of each bit of hardware are resolved; in the Second Law investigation, irreversibility era in each control volume is ascertained. In this way, it is conceivable to evaluate, both subjectively and quantitatively, the utilization of exergy in each purpose of the plant. The financial investigation legitimate evaluates the expenses of the results of the plant through the equalizations of exergetic and money related expenses. Exergetic expenses are doled out relatively to the exergy of each stream. In the figuring of the money related cost, expenses of fuels, makeup water, speculation, upkeep, and plant operation are considered. In this manner, data about the execution of the entire plant can be gotten. Economic analysis has the following objectives:

- to recognize the area, size and genuine sources of thermodynamic losses (destroyed exergy and exergetic losses);
- to ascertain the cost related to exergetic losses and destroyed exergy in any plant segment; and
- to investigate the cost development of every item independently, in those thermal systems which produce more than one item.

The essential target of each venture is to be gainful. Thusly, an appropriate design for any financially savvy thermal system requires an assessment of the project cost. The Economic examination was performed with the end goal of this target.

The economic model considers the yearly cost related to owning and working each plant segment and the yearly cost related with exergy decimation. The expenses of the parts incorporate amortization and upkeep, and the cost of fuel utilization. Keeping in mind the end goal to characterize a cost capacity that relies on upon the streamlining parameters of intrigue, segment costs must be communicated as elements of thermodynamic variable

CHAPTER 5

5.Mathematical modelling

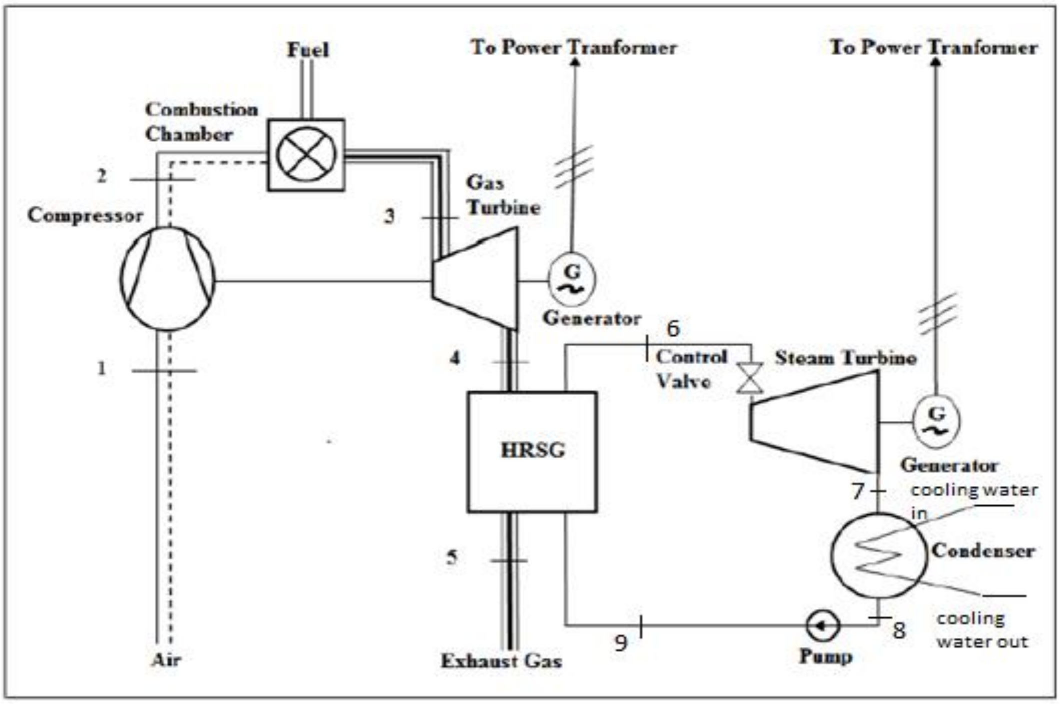

Figure 5-1 Schematic combined cycle gas turbine topping cycle and bottoming cycle

Table 5-1 Representing all points of schematic of combined cycle power plant

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

5.1FORMULATION OF EQUATIONS

5.1.1ENERGY EQUATIONS

5.1.1.1Gas turbine (Brayton cycle) circuit

The assumptions and parameters selected for the thermodynamic analysis of the Gas turbine (Brayton cycle) circuit are tabulated in Table 5-1.

Table 5-1 Gas turbine parameters

Gas turbine (Brayton cycle) circuit PARAMETERS	values
“Ambient air pressure” [bar]	1.013
“Ambient air temperature” [k]	293-313
Gas Turbine isentropic efficiency [%]	90
Compressor isentropic efficiency [%]	88
Compression ratio [-]	2-30
Air-fuel ratio [–]	50-130
Lower calorific value of fuel[kJ/kg]	43500
Heat capacity ratio for air (γ_a)[49]	1.4
Heat capacity ratio for gas (γ_g)[49]	1.33

5.1.1.1.1Compressor

By applying the energy balance for compressor, the following relation is obtained:

Theoretical Temp. at first compressor exit

$$T_2 = T_1 . r^{\frac{\gamma_a - 1}{\gamma_a}} \quad 5\text{-}1$$

Theoretical work absorbed by compressor

$$\dot{W}_c = (\dot{m_a}) C_{pa} (T_2 - T_1) \quad 5\text{-}2$$

Actual temp. at first compressor exit

$$T_{2s} = T_1 + \frac{(T_2 - T_1)}{(\eta_{comp})_{isen.}} \quad 5\text{-}3$$

Actual work absorbed by compressor

$$\dot{W}_{c_{actual}} = (\dot{m}_a) C_{pa} (T_{2s} - T_1) \quad 5\text{-}4$$

5.1.1.1.2Combustion chamber (Combustor)

By applying the energy balance for combustion chamber, the following relation is obtained:

$$\dot{m}_a C_{p_a} T_{2_s} + \dot{m}_f CV \eta_{comb} = (\dot{m}_a + \dot{m}_f) C_{p_g} T_3 \quad 5\text{-}5$$

$$AFR = \frac{\dot{m}_a}{\dot{m}_f} \quad 5\text{-}6$$

Temp. at combustion chamber exit

$$T_3 = \frac{[(AFR \times C_{pa} \times T_{2s}) + CV\eta_c]}{(AFR+1) C_{pg}} \quad 5\text{-}7$$

5.1.1.1.3Gas Turbine

The parameters of the gas turbine are listed in Table 5-1 [38]. By applying the energy balance for Gas Turbine, the following relation is obtained:

Theoretical Temp. at turbine exit

$$T_4 = \frac{T_3}{r^{\frac{\gamma g - 1}{\gamma g}}} \quad 5\text{-}8$$

Theoretical work obtained from gas turbine

$$\dot{W}_{gt} = (\dot{m}_f + \dot{m}_a) C_{pg} (T_3 - T_4) \quad 5\text{-}9$$

Actual temp. at turbine exit

$$T_{4s} = T_3 - (\eta_{gt})_{isen.} (T_3 - T_4) \quad 5\text{-}10$$

Actual work obtained from gas turbine

$$\dot{W}_{gt_{actual}} = (\dot{m}_f + \dot{m}_a) C_{pg} (T_3 - T_{4s}) \quad 5\text{-}11$$

5.1.1.2Heat Recovery Steam Generator HRSG

HRSG is placed after the gas turbine. Hot exhaust gases from gas turbine is conducted through the heat recovery steam generator heaters to generate steam for the steam turbine. The HRSG in this study consists of three sectional heat exchangers

- section 1. superheater
- section 2. evaporator
- section 3. economizer

The assumptions and parameters of the HRSG selected for the analyses are tabulated in Table 5-2.

Table 5-2 Main characteristics and assumptions of the of HRSG (initial input data)

Parameters	Value
The pinch point temperature difference [°C]	13
Pressure in HRSG [bar]	10
“Live steam temperature at the inlet of steam turbine” [°C]	535
Minimum stack temperature [°C]	150
Feed water temperature [°C]	55

Analysis

The investigation of HRSG unit depends on the pinch point. The pinch point is basically characterized as the difference between the saturation temperature of water and the gas temperature before passage to the economizer. The pitch point is meant as PP. The more prominent the pinch point, the littler the surface zone accessible for thermal energy exchange from the hot exiting exhaust gas to the steam. This results in a higher HRSG exit temperature. The water-steam properties were derived from the standard “IAPWS” 97 [47]. The properties of the gas turbine exhaust gases, which are the combustion products of the specified fuel, were calculated according to Baehr and Diederichsen [48]. The process of heat exchange HRSG is similar to a heat exchanger. The flue gases enter the HRSG at temperature T_{4s} and passes through different sections in HRSG and then leaves at T_5. The feed water is heated from T_{9s} to the saturation temperature in the economizer section; at this temperature the feed water enters the evaporator to generate the steam and leaves at the same saturation temperature The steam is superheated to temperature in the superheater section.

Applying the first law of thermodynamics (energy balance equations) to the heat recovery steam generator parts (economizer, evaporator, superheater), yields a system of equations. The steam mass flow rate is calculated by solving these equations.

$$(\dot{m}_f + \dot{m}_a)C_{pg}(T_{4s} - T_5) = \dot{Q}_{econ} + \dot{Q}_{evap} + \dot{Q}_{sup} \qquad 5\text{-}12$$

$$\dot{Q}_{econ} = \dot{m}_w C_{p_w}(T_{sat} - T_9) \qquad 5\text{-}13$$

$$\dot{Q}_{evap} = \dot{m}_w \times LH \qquad 5\text{-}14$$

$$\dot{Q}_{sup} = \dot{m}_w C_{p_{steam}}(T_6 - T_{sat}) \qquad 5\text{-}15$$

From equation 5-12, 5-13, 5-14, 5-15 we can calculate mass flow rate of steam

$$\dot{m}_w = \frac{(\dot{m}_f+\dot{m}_a)C_{pg}(T_{4s}-T_5)}{C_{p_w}(T_{sat}-T_9)+LH+C_{p_{steam}}(T_6-T_{sat})} \qquad 5\text{-}16$$

5.1.1.3Steam Turbine (ST)

The assumptions and parameters selected for the thermodynamic analysis of the steam turbine are tabulated in Table 5-3.

Table 5-3 Main assumptions of the steam turbine

Parameters	Value
The isentropic efficiency of steam turbine	90
The isentropic efficiencies of water pump	82
Minimum dryness fraction of steam at steam turbine outlet	0.88
Low-pressure steam turbine outlet (condenser pressure) [bar]	0.07
The inlet cooling water temperature in condenser [°C]	20

By applying the energy balance for steam turbine levels, the following relation is obtained:

Theoretical work obtained from steam turbine

$$\dot{W}_{st} = \dot{m}_w(h_6 - h_7) \tag{5-17}$$

Actual work obtained from steam turbine

$$\dot{W}_{st_{actual}} = \dot{m}_w(h_6 - h_7) \times (\eta_{st})_{isen.} \tag{5-18}$$

5.1.1.4Condenser

Applying the first law of thermodynamics (energy balance equations) to the Condenser, yields a system of equations. The cooling water mass flow rate is calculated by solving these equations.

$$\dot{m}_{cw} C_{p_{cw}}(\Delta T_{cw}) = (\dot{m_w})(h_7 - h_8) \tag{5-19}$$

$$\dot{m}_{cw} = \frac{(\dot{m_w})(h_7 - h_8)}{C_{p_{cw}}(\Delta T_{cw})} \tag{5-20}$$

Theoretical work absorbed by pump

$$\dot{W}_p = \dot{m}_w(h_9 - h_8) \tag{5-21}$$

5.1.1.5Pump

Similarly, to the steam turbine calculations, the work required by the pump $\dot{W}_{p_{actual}}$ is obtained by applying the energy balance around the pump:

$$\dot{W}_{p_{actual}} = \frac{\dot{m}_w(h_9 - h_8)}{(\eta_{pump})_{isen.}} \tag{5-22}$$

5.1.1.6Performance Assessment Parameters

The performances of combined cycle power plant, including Brayton, Rankine and the overall combined cycle power output and thermal efficiency are calculated as given below:

Net work in Brayton circuit

$$\dot{W}_{net_{brayton}} = \dot{W}_{gt_{actual}} - \dot{W}_{c_{actual}} \tag{5-23}$$

$$W_{Net} = (\dot{m}_f + \dot{m}_a)C_{pg}(T_3 - T_{4s}) - (\dot{m}_a)C_{pa}(T_{2s} - T_1) \tag{5-24}$$

Brayton circuit Efficiency

$$\eta_{brayton} = \frac{\dot{W}_{net_{brayton}}}{\dot{m}_f \times CV \times \eta_{comb}} \qquad 5\text{-}25$$

Net work obtained in Rankine circuit

$$\dot{W}_{net_{rankine}} = \dot{W}_{st_{actual}} - \dot{W}_{p_{actual}} \qquad 5\text{-}26$$

Rankine circuit Efficiency

$$\eta_{rankine} = \frac{\dot{W}_{net_{rankine}}}{\dot{m}_w (h_6 - h_9)} \qquad 5\text{-}27$$

Combined cycle power output

$$\dot{W}_{net_{combined}} = \dot{W}_{net_{brayton}} + \dot{W}_{net_{rankine}} \qquad 5\text{-}28$$

Combined cycle efficiency

$$\eta_{combined} = \frac{\dot{W}_{net_{brayton}} + \dot{W}_{net_{rankine}}}{\dot{m}_f \times CV \times \eta_{comb}} \qquad 5\text{-}29$$

5.1.2ECONOMIC EQUATIONS

$$Selling\ cost\ per\ Unit = ₹C \tag{5-30}$$

$$Air\ cost\ per\ cycle = AFR \times ₹a \tag{5-31}$$

$$Fuel\ cost\ per\ cycle = ₹f \times (m_f) \tag{5-32}$$

$$\frac{Cycles}{sec} = \frac{Plant\ Capacity(KW)}{Net\ work(\frac{KJ}{cycle})} \tag{5-33}$$

$$Installation\ Cost = ₹i/Megawatt \tag{5-34}$$

$$Total\ Installation\ Cost = Plant\ Capacity \times i₹ \tag{5-35}$$

5.1.2.1Depreciation

The decline in the estimation of the power plant equipment and working because of steady utilize is known as devaluation. On the off chance that the power station hardware was to keep going for ever, then enthusiasm on the capital speculation would have been the main charge to be made. In any case, in real practice, each power station has a useful life extending from fifty to sixty years. From the time the power station is introduced, its equipment relentlessly crumbles because of wear and tear so that there is a slow diminishment in the estimation of the plant. This lessening in the estimation of the plant each year is known as yearly deterioration. Because of deterioration, the plant must be supplanted by the new one after its helpful life. In this manner, the appropriate sum must be put aside consistently so that when the plant resigns, the gathered sum by a method for deterioration squares with the cost of substitution. It winds up plainly clear that while deciding the cost of creation, yearly devaluation charges must be incorporated

$$\begin{bmatrix} Depreciation \\ per\ second \end{bmatrix} = \left[\frac{\begin{pmatrix} Total \\ Installation\ Cost \end{pmatrix} \times \begin{pmatrix} \text{Depreciation} \\ \text{Factor} \end{pmatrix}}{\begin{pmatrix} \text{Life of} \\ \text{Plant} \end{pmatrix} \text{(in seconds)}}\right] \times \begin{bmatrix} overhead \\ factor \end{bmatrix} \times \begin{bmatrix} Rate\ of \\ Interest \end{bmatrix} \tag{5-36}$$

$$Depreciation\ per\ cycle = \frac{Depreciation\ per\ second}{Cycle/second} \tag{5-37}$$

$$\frac{Total\ Production\ Cost}{cycle} = Air\ cost + Fuel\ cost + Depreciation\ Cost \tag{5-38}$$

$$\frac{Total\ Production\ Cost}{Unit} = \frac{Cost/cycle}{KWh} \qquad 5\text{-}39$$

Plant factor

The net capacity factor of a power plant is the ratio of its actual output over a period of time, to its potential output if it were possible for it to operate at full capacity continuously over the same period of time. To calculate the capacity factor, take the total amount of energy the plant produced during a period of time and divide by the amount of energy the plant would have produced at full capacity. Capacity factors vary greatly depending on the type of fuel that is used and the design of the plant.

Therefore, a higher load factor usually means more output and a lower cost per unit, which means an electricity generator can sell more electricity at a higher spark spread.

Plant factor refers to the capacity factor of an entire generating facility including all available generating units.

$$Plant\ Factor = \frac{Selling\ Cost}{Total\ Production\ Cost} \qquad 5\text{-}40$$

Plant load factor

PLF decides the correct load ability of the thermal power plant. By implication, it gives the working performance of the power plant. On the off chance that the PLF is 100%, it implies plant is running on full load according to introduced limit. As the PLF approaches 100%, the working performance of the power plant additionally increments. Among different points of interest of higher PLF, one of the real one is lessening in the arranged or constrained blackouts, that at long last optimizes auxiliary power utilization. On higher load, all the individual additional auxiliaries likewise keep running on full load, which brings about use of different auxiliaries at higher proficiency. At last life of the auxiliary additionally increments. Henceforth PLF is observed to be the key marker for the examination of working performance of any power plant.

$$Plant\ Load\ Factor = \frac{1}{Plant\ Factor} \qquad 5\text{-}41$$

Table 5-4 Economic assumptions, prices and coefficients

PARAMETER	VALUE
Life of Plant (N) [year]	35
Fuel cost per unit[₹kg^{-1}]	6
Selling price of electricity[₹kWh^{-1}]	
Overhead (Labour + Maintenance cost) Factor	1.2
Interest rate (i) [-]	0.1
Depreciation Factor	0.8

CHAPTER 6

6.RESULTS AND DISCUSSION

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

6.1. Result from Energy Analysis:

6.1.1. Effect of Pressure ratio at different AFR on Efficiency

Figure 6-1 and Figure 6-2 illustrates the efficiency of Gas turbine power plant as the function of pressure ratio at various air fuel ratios. Pressure ratio was varied from 2 to 30 in a step of 2 while Air fuel ratio was varied from 50 to 130. As the values of air fuel ratio is increasing, the magnitude of gas turbine efficiency is decreasing. But for particular air fuel ratio, gas turbine efficiency keeps on increasing with the increase in pressure ratio up to a definite value of air fuel ratio. After that certain value of air fuel ratio, the gas turbine efficiency first increases then decrease with the increase in pressure ratio. This is due to, at higher air fuel ratio the work done on compressor increases more rapidly as work done by turbine. More over when pressure ratio is increased on these values of air fuel ratio, it shows fall in gas turbine efficiency.

6.1.2. Effect of Ambient Temperature at different Pressure ratio on efficiency

Figure 6-3 demonstrates the efficiency of gas turbine power plant as a function of ambient temperature at various pressure ratios. Ambient temperature was varied from 293 K to 315 K in a step of 1. The Pressure ratio was varied from 5 to 30.

Gas turbine power plant efficiency decreases with increase in ambient temperature because the difference between the highest temperature in the cycle and ambient temperature is decreasing which results in decrement of efficiency at a particular value of pressure ratio.

But as the value of pressure ratio increases, the magnitude of efficiency of gas turbine power plant increases too but shows same nature with ambient temperature. At a particular ambient temperature, the marginal increase in net work done decreases with increase in pressure ratio that’s why the curves come closer at higher values of pressure ratios.

6.2. Result from Economic Analysis

In the present analysis, energy is quantified as cost and this cost is associated with the exergy loss or exergy destruction.

6.2.1. Effect of Pressure Ratio for Different Values of Air Fuel Ratio On

6.2.1.1. Number of Units of Electricity Produced (Per Cycle)

Figure 6-4 depicts the number of units of electricity produced (per cycle) as a function of the pressure ratio for different values of Air Fuel ratio. Pressure ratio varies from 2 to 30 in steps of 2 units along the x axis while a curve a generated for a particular value of AFR which is varying from 50 to 130 in steps of 10.

In general, for a particular value of pressure ratio, the Kwh value decreases as the AFR increases from 50 to 130. This is due to the fact that the compressor and the turbine work both are increasing because of increased amount of air supplied, however the increase in the compressor work is greater than the turbine work in magnitude and hence the net work or the Kwh extracted from the cycle reduces and thus the trend.

We see that for a particular AFR the marginal increase in the number of units produced goes on to decrease as the pressure ratio increases i.e. the rise in the units produced for lower pressure ratio values is greater than that at higher pressure ratio values. From figure 6-8 we see that this effect arises more due to the Brayton cycle compared to Rankine cycle. This is because at high pressure ratios the difference in the turbine and the compressor work becomes more or less stagnant.

Also as the pressure ratio increases, the curves tend to diverge meaning that the difference in the net work output is much more pronounced at higher R_P values again because the turbine and the compressor work is large so the difference between them is also amplified.

6.2.1.2. PLANT FACTOR

Figure 6-5 tells us about the value of Plant factor as the pressure ratio is varied for different values of AFR. The pressure ratio varies from 2 to 30 in steps to 2 units while curves are plotted for AFR values ranging from 50 to 130 with 10 units' difference. The fuel cost was taken to be Rs.6 per Kg.

Although in general, we see, as the pressure ratio increases the net work output also increases (From Fig 6-4) but we must note that the fuel costs (From Fig 6-7) as well as the depreciation cost are also increasing which escalates our input cost per cycle. (Given **Plant Factor= Selling Cost/Input Cost**) Therefore, we observe that the plant factor first increases up to a certain R_P value (4 in this case i.e. optimum value) as the increase in the net work output more than compensates for the increase in the input costs and then, Plant Factor starts to decrease in a gradual manner as the fuel costs start to dominate.

Also note that this trend is only valid for lower values of AFR i.e. from 50 to 90. While for higher AFR values the curves are always decreasing in nature as the net work output of the cycle falls in general and the mass of fuel required for the sustenance of the cycle is also high. This also explains the fact that the plant factor is decreasing for higher values of AFR.

We can see that for lower AFR values the divergence between curves is more that is a greater difference between the plant factor values, given a particular value of R_p. This difference gradually decreases as the AFR value increases. This is because the numerator (Kwh) is decreasing and at the same time the denominator (Input Cost) is increasing.

6.2.1.3. PLANT LOAD FACTOR

Figure 6-6 illustrates the value of Plant Load Factor as the pressure ratio is varied for different values of AFR. The pressure ratio varies from 2 to 30 in steps to 2 units while curves are plotted for AFR values ranging from 50 to 130 with 10 units difference. The fuel cost was taken to be Rs.6 per Kg.

We know that Plant Load Factor is the inverse of Plant Factor, so the analysis would be on similar lines to figure 6-5.

Although in general, we see, as the pressure ratio increases the net work output also increases but we must note that the fuel costs (From Fig 6-7) as well as the depreciation cost are also increasing which escalates our input cost per cycle and thus the load factor value. (Given **Plant Load Factor= Input Cost/Selling Cost**) We observe that the plant load factor first decreases up to a certain R_P value (4 in this case i.e. optimum value) as the increase in the net work output more than compensates for the increase in the input costs but after that, Plant Load Factor starts to increase in a gradual manner as the fuel costs/input costs start to dominate.

Also note that this trend is only valid for lower values of AFR i.e. from 50 to 90. While for higher AFR values the curves are always increasing in nature as the mass of fuel required for the sustenance of the cycle is high and the net work output of the cycle falls in general. This also explains the fact that the plant load factor is increasing for higher values of AFR.

We can see that for lower AFR values the divergence between curves is more that is a greater difference between the plant load factor values, given a particular value of R_P. This difference gradually decreases as the AFR value increases. This is because the numerator (Input cost) is Increasing and at the same time the denominator (Kwh) is decreasing.

6.2.1.4. MASS OF FUEL (IN KG)

Figure 6-7 depicts the value of mass of fuel (in Kg) used in the combined cycle versus the pressure ratio which is varied from 2 to 30 in steps of 2 units. We know that the total fuel consumed is the sum of m_f . While m_f remains fixed at 1 Kg.

Also note that the fuel requirement escalates as the AFR value increases. This is because more amount of heat is required to increase the temperature of greater mass of gases.

Also as the pressure ratio increases, the curves tend to diverge meaning that the difference in the mass of fuel used is much more pronounced at higher R_p values.

6.2.2. Comparison Drawn Between Brayton, Rankine and Combined (Brayton + Rankine) Cycles

6.2.2.1. Comparison of Net Work Outputs of Cycles at Different Pressure Ratio

Figure 6-8 is a comparison drawn between the net work outputs of Brayton, Rankine and combined (Brayton + Rankine) cycles as the pressure ratio is varied. The AFR value is chosen to be 65 while the pressure ratio changes from 2 to 30 in steps of 2 units. From the graph we see that the work output for Rankine cycle remains more or less the same only. Initially there is a dip from R_p 2 to 10 followed by a very gradual increase up to R_p=30. Now, we know that the work output of Rankine cycle depends primarily on the mass of fuel used in the Brayton cycle as well as the exhaust temperature from the turbine. From the table we see that the mass of fuel used increases consistently as the Pressure ratio is increased. But the turbine outlet temperature decreases suddenly for R_p 2 to 8 and then gradually from 8 to 12 and from there on starts to increase. The combined effect of both the factors can now be visualized from figure 6-8. As far as the Brayton cycle is concerned the net work done shows an increasing trend always. We see that the marginal increase in the number of units produced goes on to decrease as the pressure ratio increases i.e. the rise in the Net work done for lower pressure ratio values is greater than that at higher pressure ratio values. This is because at high pressure ratios the difference in the turbine and the compressor work becomes more or less stagnant and thus the slope decreases. Also note, the net work output of Brayton cycle equals that of Rankine cycle between pressure ratios 6 and 8. It is not necessary that the combined net work be the highest for this point as can be seen from the graph.

6.2.3. Net Work Done Versus the Combined Cycle Efficiency for AFR=65

Figure 6-9 depicts the value of the net work done versus the combined cycle efficiency for AFR=65 which is considered to be the ideal air fuel mixture. Also note that irrespective of the efficiency, a higher value of net work done signifies that the plant is operated at a high pressure ratio. From the graph it can be seen that the maximum net efficiency occurs for a pressure ratio value of 8 and the maximum value is 30.35% (Given that least possible value of AFR=65) for which the net work done comes out to be 7.066 x 10^4 KJ. For pressure ratio above or below 8 the efficiency figure drops.

6.2.4. Net Work Done Versus the Combined Cycle Efficiency at Rp=12

Figure 6-10 depicts the value of the net work done versus the combined cycle efficiency for Rp=12 which is considered to be the ideal pressure ratio. Also note that irrespective of the efficiency, a higher value of net work done signifies that the plant is operated at a low AFR.

From the graph it can be seen that the maximum net efficiency occurs for an AFR value of 50 and the maximum value is **34.05%** for which the net work done comes out to be 8.064 x 10^4 KJ. This gives us the maximum possible efficiency of the plant given the minimum possible AFR value is 50. The minimum efficiency exists for the highest value of AFR i.e. 130 and the value is 16.4

6.2.5. Net Work Done Versus the Combined Cycle Efficiency for A Few Favourable Value of Pressure Ratios

Figure 6-11 depicts the value of the net work done versus the combined cycle efficiency for a few favourable value of pressure ratios i.e. from 10 to 14 in steps of 1. The AFR ratio is varied from 50 to 66 in steps of 4 units. Also note that irrespective of the efficiency, a higher value of net work done signifies that the plant is operated at a low AFR. This implies that the rightmost point corresponds to the least AFR i.e. 50 in this case and leftmost point represents AFR=66.

From the graph it is quite evident that maximum efficiency occurs for the least value of AFR i.e. 50 in this case. However, it is not necessary that the plant be operated at this value as it may not prove viable. But assuming we operate at the lowest AFR value, the efficiencies are given for different Rp values. From the graph it can be inferred that the maximum efficiency occurs for Rp 12 and 13 both & the maximum value is **34.05 %.**

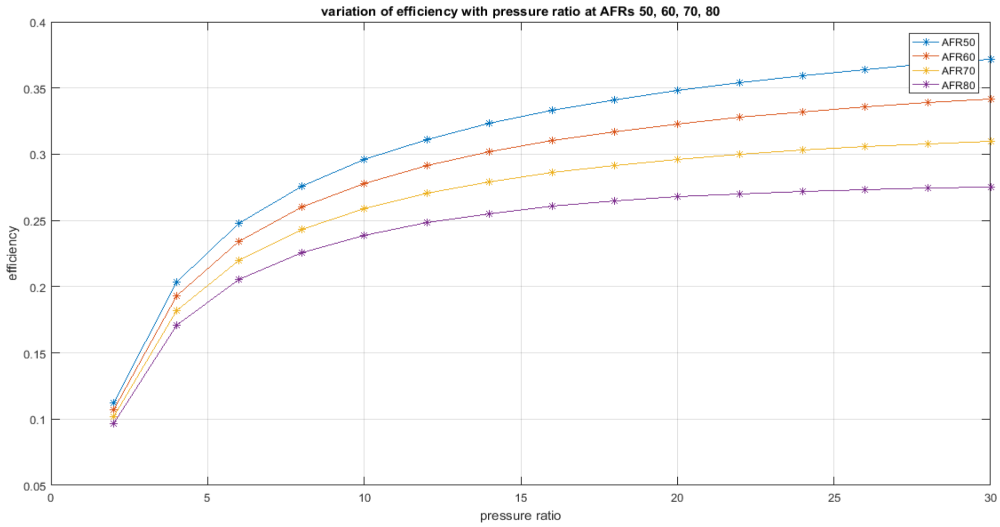

Figure 6-1 Variation of efficiency with pressure ratio at different AFR

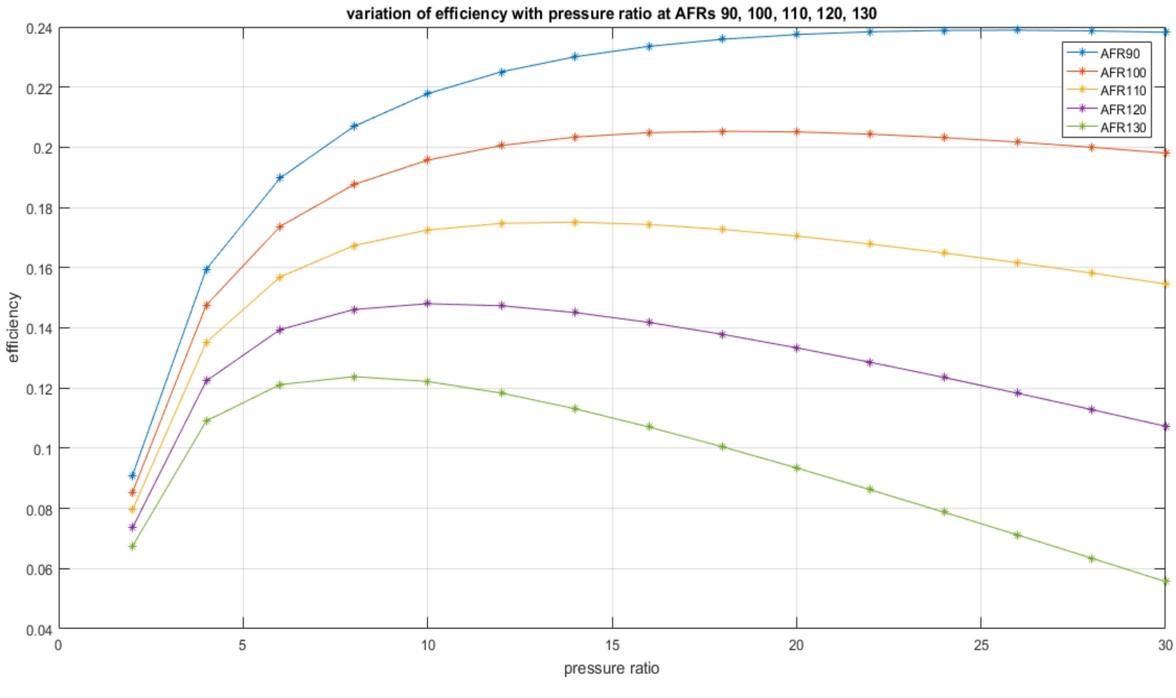

Figure 6-2 Variation of efficiency with pressure ratio at different AFR

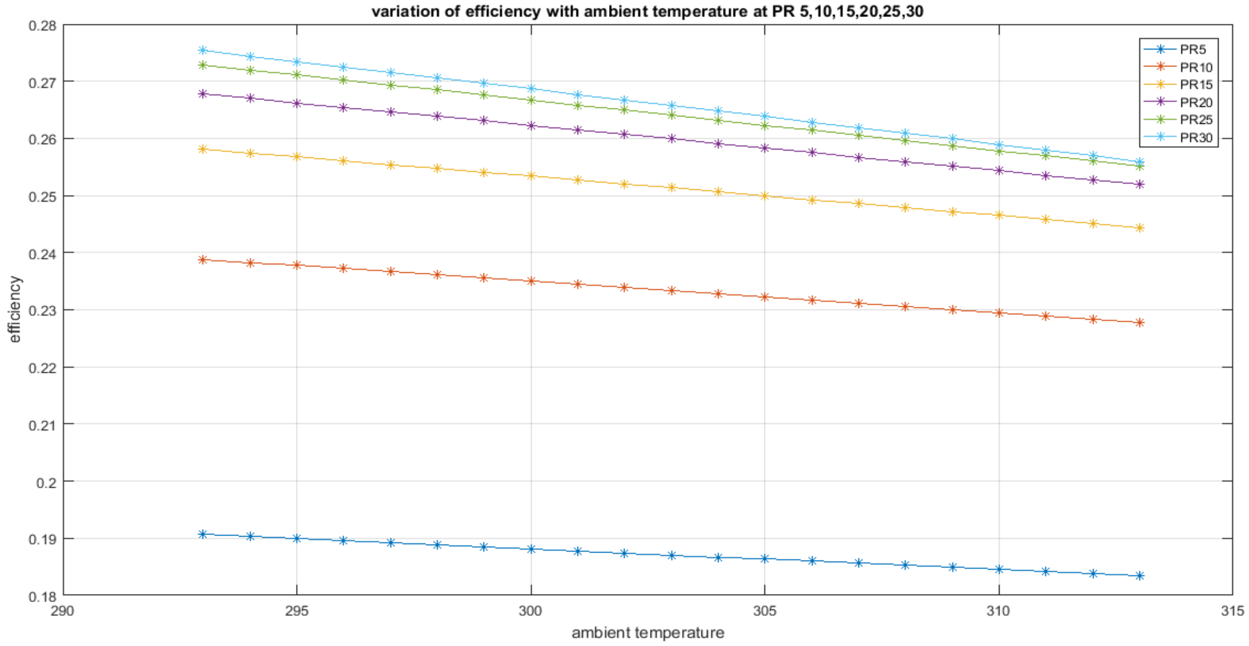

Figure 6-3 Variation of efficiency with Ambient Temperature at different Pressure ratio

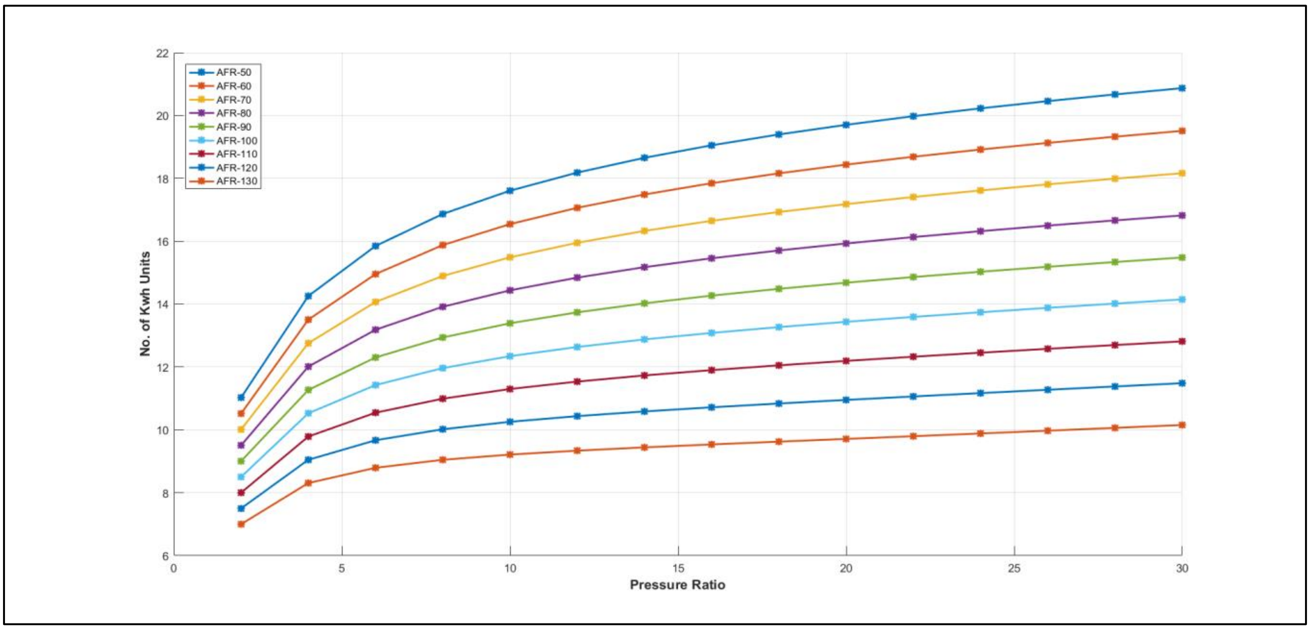

Figure 6-4 variation of number of units of electricity produced with pressure ratio for different values of air fuel ratio

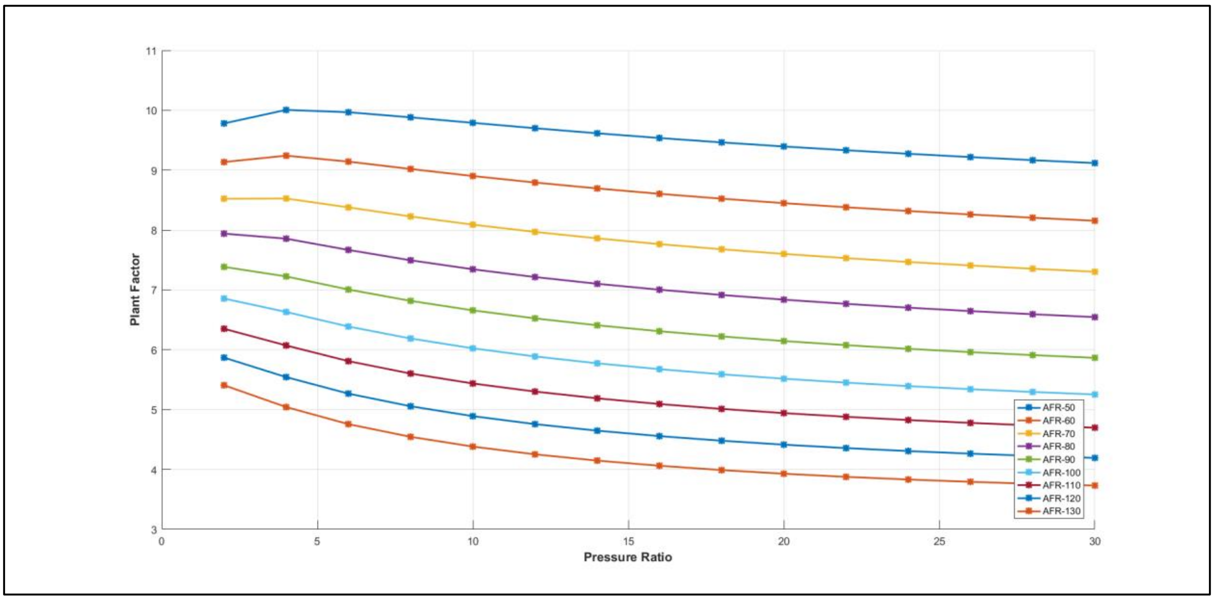

Figure 6-5 variation plant factor with pressure ratio for different values of air fuel ratio

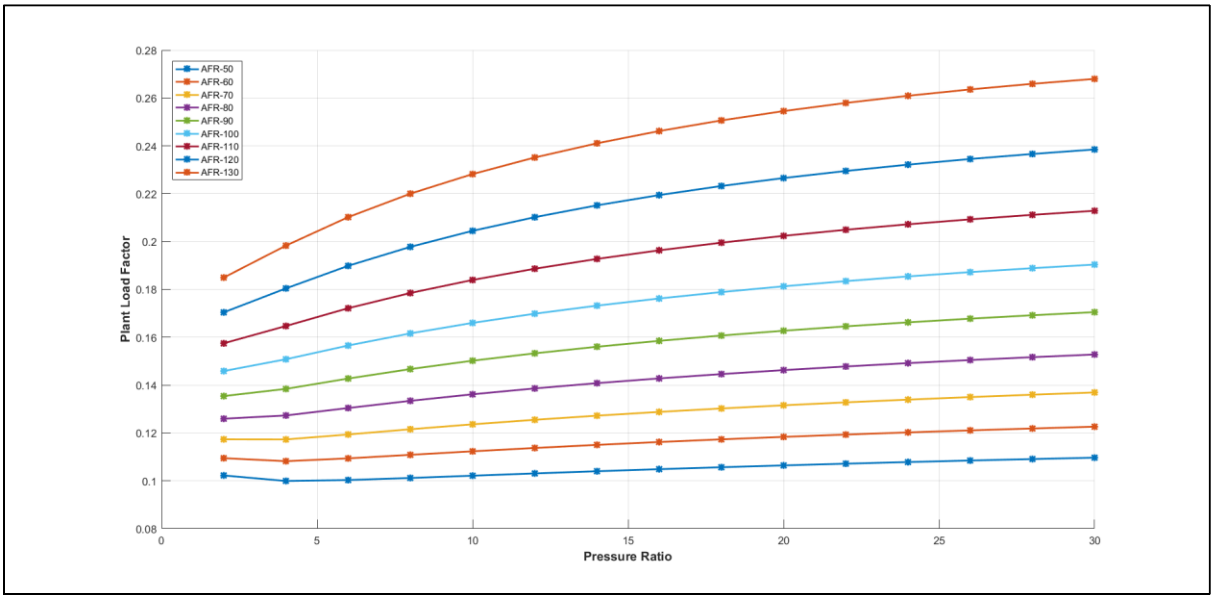

Figure 6-6 variation plant load factor with pressure ratio for different values of air fuel ratio

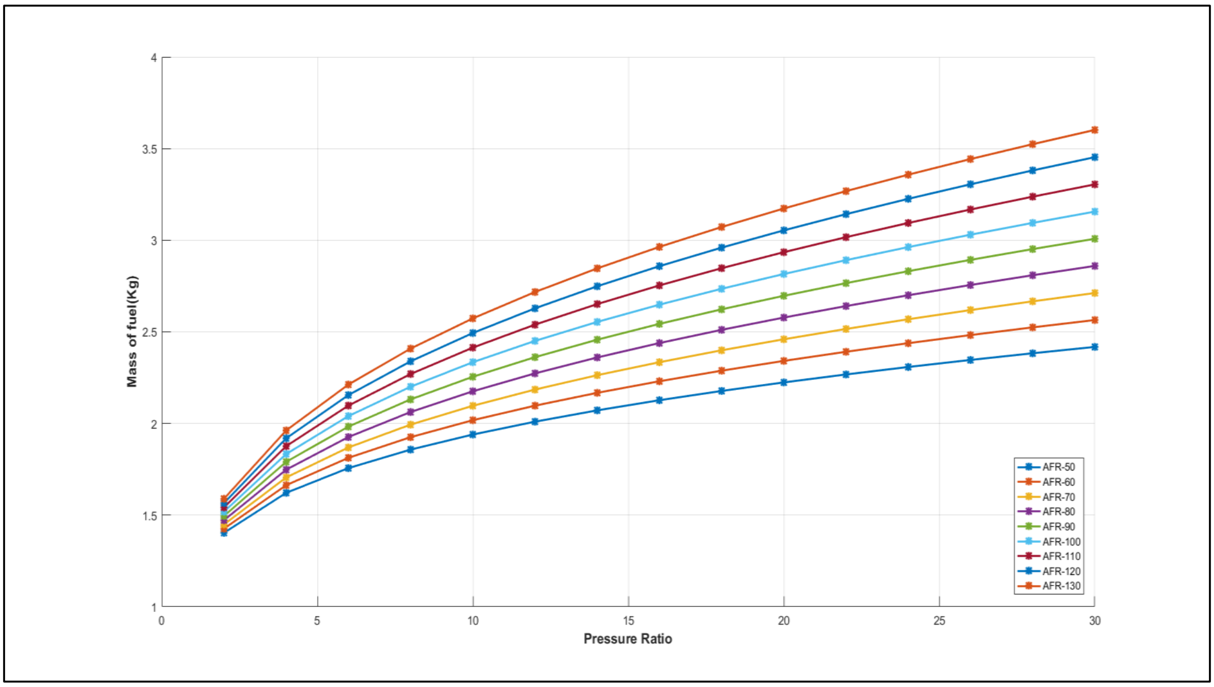

Figure 6-7 variation mass of fuel with pressure ratio for different values of air fuel ratio

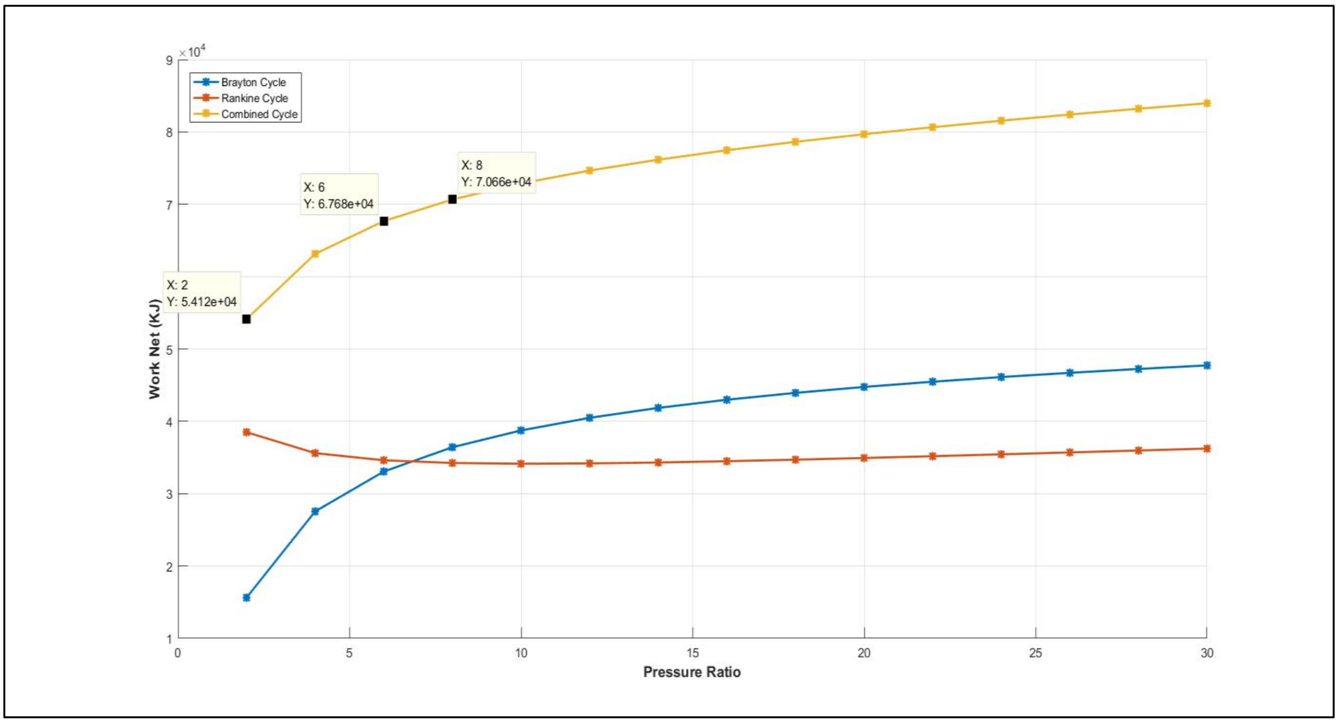

Figure 6-8 variation work net with pressure ratio

1	Mass Of Fuel Used (Kg)	Outlet Tempeture From Turbine T8 (k)	Mass of Fluid Circulated (Kg)(Rankine)	Net Work Rankine* 10^4 (KJ)
2	1.4376	882.7419	22.6063	3.8498
3	1.6844	846.5876	20.9059	3.5603
4	1.8408	833.9118	20.3279	3.4618
5	1.9596	828.7335	20.1074	3.4243
6	2.0572	826.9343	20.0474	3.4141
7	2.1411	826.9307	20.0723	3.4183
8	2.2154	827.9772	20.1466	3.431
9	2.2824	829.6767	20.2513	3.4488
10	2.3437	831.7987	20.3755	3.4699
11	2.4005	834.201	20.5125	3.4933
12	2.4535	836.7915	20.658	3.518
13	2.5034	839.5089	20.809	3.5438
14	2.5506	842.3104	20.9636	3.5701
15	2.5954	845.1664	21.1204	3.5968
16	2.6382	848.0552	21.2784	3.6237

Table 6-1: values of different parameters

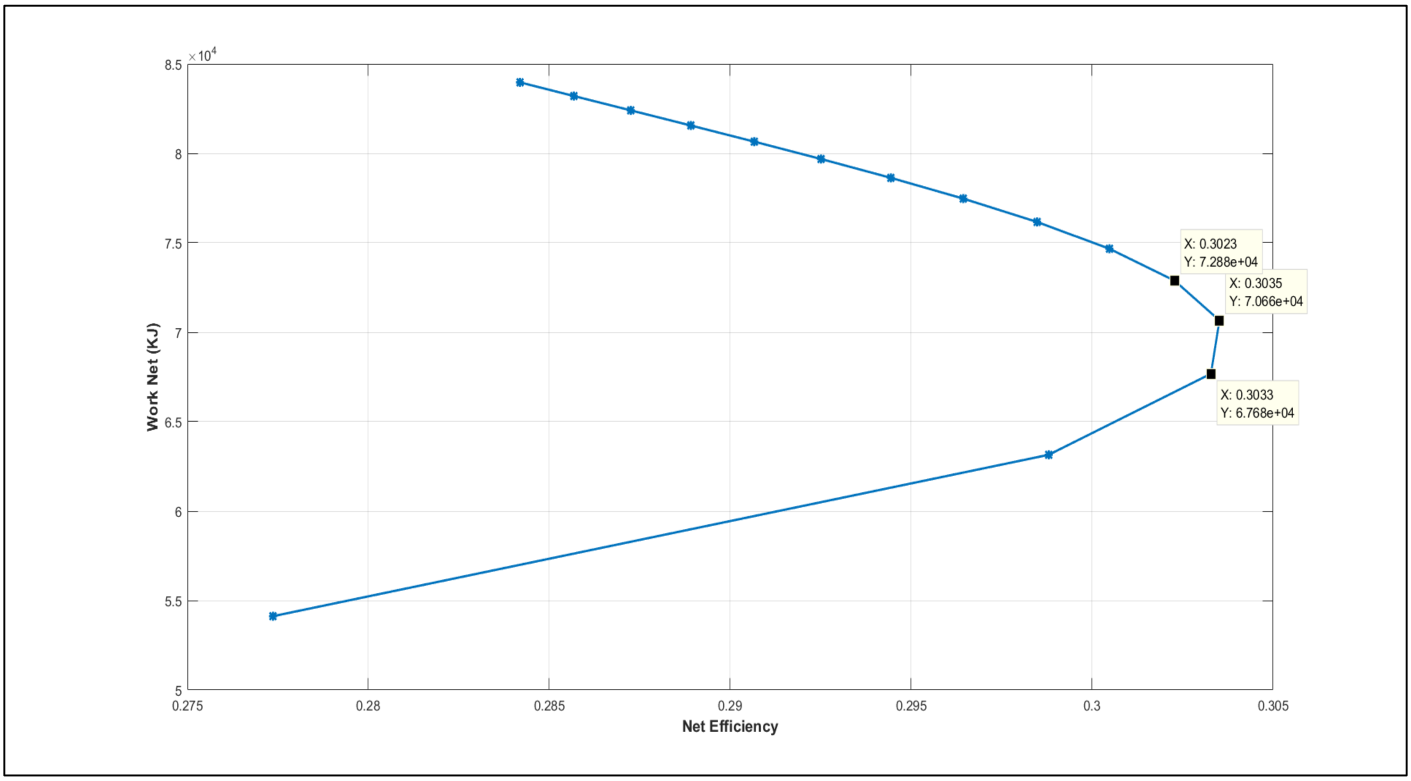

Figure 6-9 variation work net with net efficiency

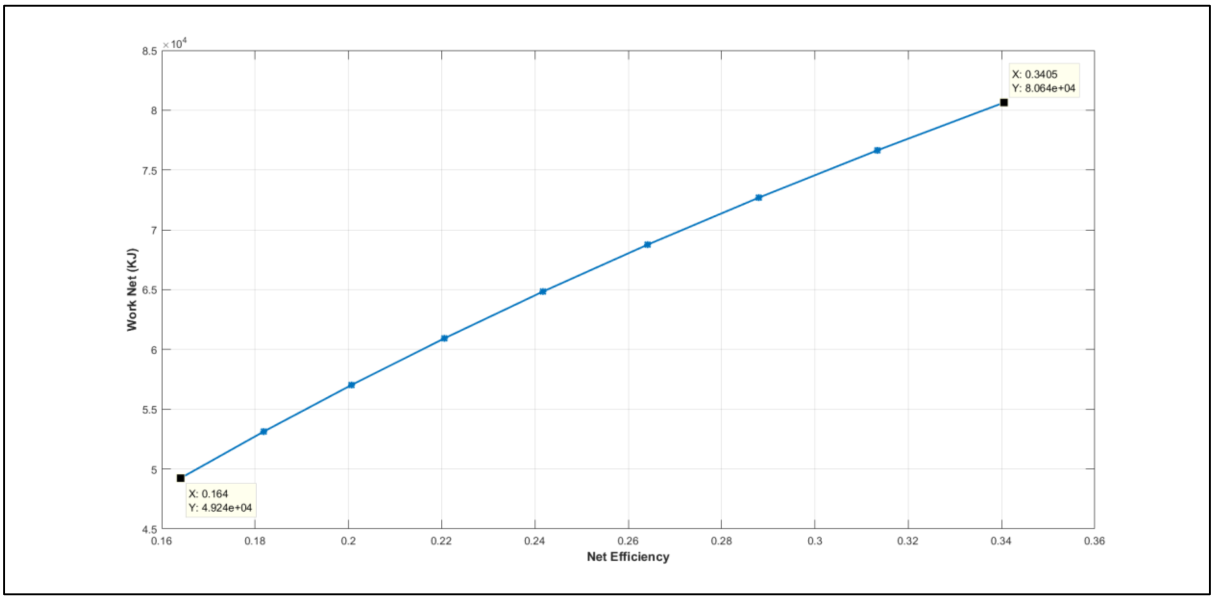

Fig 6-10 variation of net work done versus combined cycle efficiency

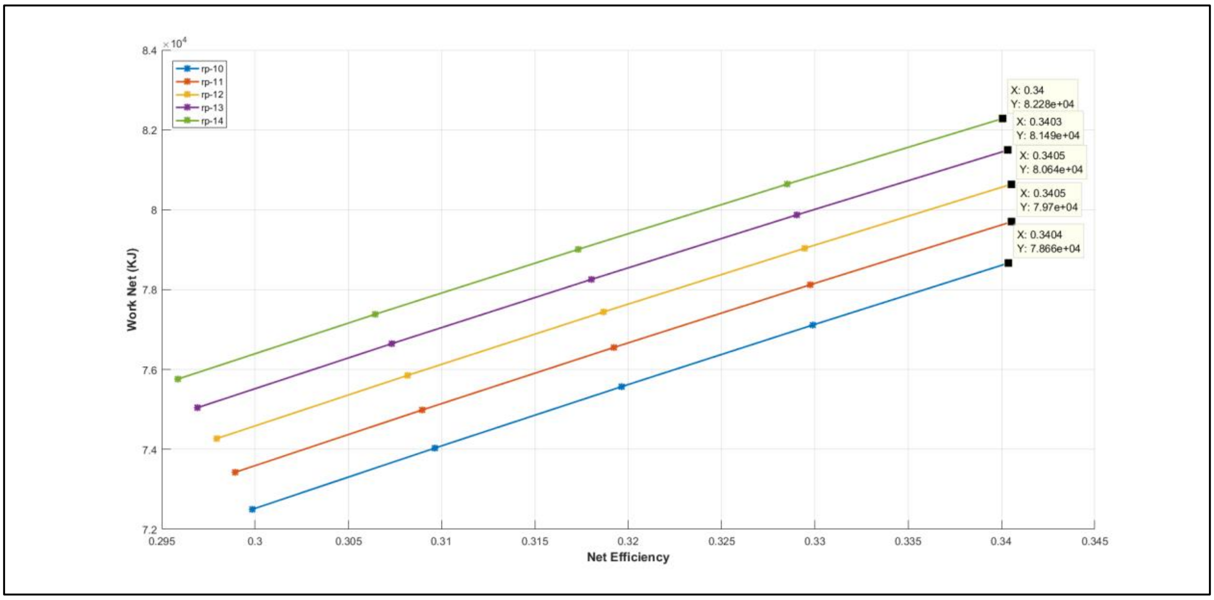

FIG 6-11 Net work done versus the combined cycle efficiency for a few favourable value of pressure ratios

CHAPTER 7

CONCLUSION AND RECOMMENDATIONS

In this thesis, with studying the first and second law of thermodynamics for combined cycle power plant attempted Thermodynamically Energy and Economic analysis is performed for different points of cycle by simultaneous equations. Now the mentioned cycle is Thermoeconomically analysed, by help of these results and writing cost Balance for individual components, which finally has Thermoeconomic optimization with the Exergy flow rates for various parts of the cycle. That its outcome is Finding of Important outline parameters including pressure ratio and airflow rate of compressor for limit the aggregate cost and boost the aggregate power yield.

The thermoeconomic examination comes about demonstrate that the combustion chamber has the maximum cost of energy loss of all parts, and that increasing the Gas Turbine Inlet Temperature diminishes the CCPP cost of energy loss.

In addition, with energy analysis, economic analysis also done in the present work. The impact of different parameters on plant financial aspects has been watched and the accompanying conclusions have been accomplished. The total operating cost, net present value of plant, revenue earned, costs of fuel, maintenance, labour, insurance and pumping with escalation rates observed to be higher than the case without escalation. Also energy is quantified as cost and this cost is associated with the energy loss or exergy destruction. Thus this cost shows a loss value which means energy loss shows a monetary loss and can be minimized using the minimization in the exergy destruction

Conclusions

This study reveals that the CCPP system has the highest efficiency among the Brayton and Rankine cycle individually used

- As the values of air fuel ratio is increasing, the magnitude of gas turbine efficiency is decreasing. But for particular air fuel ratio, gas turbine efficiency keeps on increasing with the increase in pressure ratio up to a definite value of air fuel ratio. After that certain value of air fuel ratio, the gas turbine efficiency first increases then decrease with the increase in pressure ratio.

- Gas turbine power plant efficiency decreases with increase in ambient temperature because the difference between the highest temperature in the cycle and ambient temperature is decreasing which results in decrement of efficiency at a particular value of pressure ratio. But as the value of pressure ratio increases, the magnitude of efficiency of gas turbine power plant increases too but shows same nature with ambient temperature.
- the number of units of electricity produced (per cycle) as a function of the pressure ratio for different values of Air Fuel ratio. Pressure ratio varies from 2 to 30 in steps of 2 units along the x axis while a curve a generated for a particular value of AFR which is varying from 50 to 130 in steps of 10
- the value of Plant Load Factor as the pressure ratio is varied for different values of AFR. The pressure ratio varies from 2 to 30 in steps to 2 units while curves are plotted for AFR values ranging from 50 to 130 with 10 units difference. The fuel cost was taken to be Rs.6 per Kg
- the value of mass of fuel (in Kg) used in the combined cycle versus the pressure ratio which is varied from 2 to 30 in steps of 2 units. We know that the total fuel consumed is the sum of m_f . While m_f remains fixed at 1 Kg.
- the value of the net work done versus the combined cycle efficiency for AFR=65 which is considered to be the ideal air fuel mixture. Also note that irrespective of the efficiency, a higher value of net work done signifies that the plant is operated at a high pressure ratio. From the graph it can be seen that the maximum net efficiency occurs for a pressure ratio value of 8 and the maximum value is 30.35%
- From the graph it can be seen that the maximum net efficiency occurs for an AFR value of 50 and the maximum value is **34.05%** for which the net work done comes out to be 8.064 x 10^4 KJ. This gives us the maximum possible efficiency of the plant given the minimum possible AFR value is 50. The minimum efficiency exists for the highest value of AFR i.e. 130 and the value is 16.4

Recommendations

The recommendations for future research are given below.

- The working fluid considered in this study is natural gas. It is characterized by relatively high efficiency as compared to the other organic working fluids. While this characteristic is the main reason for selecting this working fluid, it is recommended to examine other organic fluids to explore higher efficiency operation. It is also possible to examine binary organic fluids.
- The main component of CCPP is HRSG more optimization can be done on varying various parameters of HRSG
- In this study, a specific pressure range was selected based on some references from the literature. The study reveals that the pressure change has a significant effect on the performance. However, changing the pressure ratio with inlet temperature to the compressor may show a more significant effect on the performance
- The emissions from the CCPP system can be reduced by, for example, using a more efficient Stack after HRSG or switch to the biomass fuel. However, by considering the carbon natural cycle, the CO2 emissions from the biomass fuel could be considered having lesser environmental impact.

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APPENDICES

APPENDIX A: Thermodynamic properties of steam

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

180	194.44	5.1431	2777.4	6.5857
185	197.42	5.0653	2790.7	6.6148
190	200.34	4.9916	2803.5	6.6427
195	203.20	4.9212	2816.0	6.6695
200	206.02	4.8539	2828.3	6.6955
210	211.56	4.7268	2852.2	6.7456
220	216.98	4.6087	2875.5	6.7934
230	222.31	4.4983	2898.4	6.8393
240	227.56	4.3944	2920.9	6.8836
250	232.75	4.2965	2943.1	6.9265
260	237.88	4.2038	2965.1	6.9681
270	242.96	4.1159	2986.9	7.0087
280	248.01	4.0322	3008.6	7.0482
290	253.01	3.9524	3030.2	7.0868

t, °C	**1.0 MPa (t_s = 179.878 °C)**			
	v	ρ	h	s
300	257.99	3.8762	3051.6	7.1246
310	262.94	3.8032	3073.0	7.1616
320	267.86	3.7333	3094.4	7.1979
330	272.76	3.6662	3115.7	7.2335
340	277.64	3.6018	3136.9	7.2685
350	282.50	3.5398	3158.2	7.3029
360	287.35	3.4801	3179.4	7.3367
370	292.18	3.4225	3200.7	7.3700
380	297.00	3.3670	3221.9	7.4028
390	301.81	3.3133	3243.2	7.4351
400	306.61	3.2615	3264.5	7.4669
410	311.39	3.2114	3285.8	7.4984
420	316.17	3.1629	3307.1	7.5294
430	320.94	3.1159	3328.5	7.5600
440	325.69	3.0704	3349.9	7.5902
450	330.45	3.0262	3371.3	7.6200
460	335.19	2.9834	3392.8	7.6495
470	339.93	2.9418	3414.3	7.6786
480	344.66	2.9014	3435.8	7.7075
490	349.39	2.8621	3457.4	7.7360
500	354.11	2.8240	3479.1	7.7641
520	363.54	2.7507	3522.6	7.8196
540	372.95	2.6813	3566.2	7.8740

560	382.35	2.6154	3610.1	7.9273
580	391.74	2.5527	3654.2	7.9796
600	401.11	2.4931	3698.6	8.0310
620	410.47	2.4362	3743.2	8.0815
640	419.82	2.3820	3788.0	8.1312
660	429.16	2.3301	3833.1	8.1800
680	438.50	2.2805	3878.5	8.2281
700	447.83	2.2330	3924.1	8.2755
720	457.15	2.1875	3970.0	8.3221
740	466.47	2.1438	4016.1	8.3681
760	475.78	2.1018	4062.5	8.4135
780	485.08	2.0615	4109.2	8.4582
800	494.38	2.0227	4156.1	8.5024
820	503.68	1.9854	4203.3	8.5460
840	512.97	1.9494	4250.8	8.5890
860	522.26	1.9147	4298.5	8.6315
880	531.55	1.8813	4346.5	8.6735
900	540.83	1.8490	4394.8	8.7150
920	550.11	1.8178	4443.3	8.7560
940	559.39	1.7877	4492.1	8.7965
960	568.67	1.7585	4541.1	8.8366
980	577.94	1.7303	4590.4	8.8763
1000	587.21	1.7030	4639.9	8.9155
1100	633.54	1.5784	4891.4	9.1056
1200	679.83	1.4710	5148.9	9.2866
1300	726.10	1.3772	5411.9	9.4593
1400	772.34	1.2948	5680.0	9.6245
1500	818.57	1.2216	5953.0	9.7830
1600	864.78	1.1564	6230.3	9.9351
1800	957.19	1.0447	6796.7	10.222
2000	1049.6	0.952 78	7376.8	10.489

APPENDIX A: Thermodynamic properties of Water

p, MPa	t, °C	Density,kg/m3		Enthalpy,kJ/kg		Entropy,kJ/(kg·K)		Volume,cm^3/gm	
		ρ_L	ρ_v	h_L	h_v	s_L	s_v	v_L	v_v
0.0070	39.0	992.55	0.048722	163.35	2571.7	0.55903	8.2745	1.0075	20524.