THERMODYNAMIC ANALYSIS OF COMBINED CYCLE POWER PLANT WITH BYPASSING AND REHEATING

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Submitted by:

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DECLARATION

I hereby declare that the work presented in the major thesis **"THERMODYNAMIC ANALYSIS OF COMBINED CYCLE POWER PLANT WITH BYPASSING AND REHEATING"** submitted to Delhi Technological University in partial fulfillment for the award of the degree of Master of Technology in "Thermal Engineering" is an authentic record of my own work carried out under the supervision of Prof. B.B. ARORA, Department of Mechanical Engineering. I have not submitted this dissertation for any other purpose than the award of a degree or diploma.

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CERTIFICATE

This is to certify that CHOUDHURY ANKIT KUMAR SARAN, (2K20/THE/08), student of M.Tech, "THERMAL ENGINEERING," Delhi Technological University, has submitted the dissertation titled **"THERMODYNAMIC ANALYSIS OF COMBINED CYCLE POWER PLANT WITH BYPASSING AND REHEATING"** under my supervision as partial fulfilment of the requirements for the award of the degree of Master of Technology.

Supervisor

Prof. B.B. Arora

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ABSTRACT

The fundamental aim of the presented research is to demonstrate how different parameters of a combined cycle power plant, such as overall efficiency, overall work done, steam mass flow rate, and so on, alter with a change in the Brayton cycle pressure ratio when bypassing and reheating are performed. A combined cycle power plant is made up of gas turbine along with steam turbine and heat exchanger. For reaching the optimal result, we basically combine the Brayton and Rankine cycles. The calculations required for the thermal analysis are performed using MATLAB software, which considers genuine variable ranges of the main operating factors such as pressure ratio, temperature, air fuel ratio along with mass flow rate of air. These parameters' influence on system performance are explored. The optimization findings show that varied values of optimization methods can increase thermodynamic performance.

Key words: Thermodynamic Optimization Exergy, Combined cycle, Heat Recovery Steam Generator,

Scientific field: Thermal Engineering, Mechanical Engineering

Narrow scientific field: Thermal Engineering.

NOMENCLATURE

C_p	Specific heat at constant pressure	kJ/kg
$C_{p\ steam}$	Specific heat at constant pressure for steam	kJ/kg
g	Gravity Acceleration	m/s^2
h	Specific Enthalpy	kJ/sec
H	Enthalpy	kJ
LH	Latent Heat	kJ
$C_{p\ gas}$	Specific heat at constant pressure for exhaust gases	kJ/kg
CV	Calorific Value	kJ
LHV	Latent Heat of Vaporization	kJ
$\dot{m}$	Mass flow Rate	kg/sec
p	pressure	bar
$C_{p\ air}$	Specific heat at constant pressure for air	kJ/kg
$\dot{Q}$	Heat transfer rate	kJ/sec
PP	Pinch point	K
Q	Heat Supplied or Rejected	kJ
q	Specific heat supplied or rejected	kJ/kg
S	Entropy	kJ/K
s	Specific Entropy	kJ/kgK
T	Temperature at various points	K
TIT	Inlet temperature of gas turbine	K
v	Specific Volume	m^3/kg
S_{gen}	Entropy Generation	kJ/kg
CCPP	Combine Cycle Power Plants	-
HRSG	Heat Recovery Steam Generator	-
r_p	Pressure Ratio	-
AFR	Air Fuel Ratio	-

Greek Symbols

γ_a	Heat Capacity Ratio of Air	-
γ_g	Heat Capacity Ratio of Exhaust Gases	-
η	Efficiency	-
ρ	Density	kg/m^3

Subscripts

CCGT	Combine Cycle Gas Turbine
CV	Control Volume
w	Water
c	Compressor
p	Pump
i	Inlet
o	Outlet
a	Air
f	Fuel
evap	Evaporator
econ	Economizer
sup	Super heater
sat	Saturated
cw	Cooling Water
gt	Gas Turbine
st	Steam Turbine
ip	Intermediate Pressure
loss	Losses
lp	Low Pressure
net	Net
out	Outlet
surr	Surroundings
sys	System

th	Thermal
comb	Combustion

Superscripts

ΔT	Temperature difference
ΔP	Pressure difference

CONTENTS

CHAPTER 1

1. INTRODUCTION:

Modern life is reliant on renewable energy supplies due to the sudden replenishment of nonrenewable energy, which is critical for attaining sustainable development. As a result, it is clear that clean, ecologically sound, and cost-effective energy solutions must be implemented as soon as possible. To ensure that everyone has access to energy, all basic and available energy services must be provided via a wide scale of different energy resources and cutting-edge conversion technologies for reduction in emission of greenhouse gases, unfavorable consequence on people along with environmental concerns at regional and national level. When electricity demand peaks, new combined cycle power plants ought to be approached internationally as a backup to different energy sources. Because of the new issues that the fossil fuel sector is facing, as well as recent breakthroughs in the field, energy technology, specifically combined cycle technology, is utmost efficient direction for enhancing effectiveness while bringing down the emissions along with the junk generated per unit of electricity. The combined cycle power plant utilizes burning gas in a gas turbine cycle which creates not only electricity, but also hot exhaust gases, which can be turned into electrical power by a linked generator. Natural gas is utilized in the combined cycle power plant owing to its cleanest properties. Rapid start-ups, short shut-down times along with quick and effective adaptation to shifting load needs are all possible thanks to its efficiency and operational flexibility. This property is particularly well suited to systems that must accept varying levels of energy supply. The mixed cycle power plant is more efficient than single-cycle power plant i.e. the turbine cycle power plant along with the steam power plant when used separately. Combined cycle power plant. When nuclear power facilities are turned off and renewable energy integration is slowed, a combined cycle power plant can provide a bridge to sustainability.

The thermodynamic factors of the combined cycle power plant characteristics is discussed in current study article. It is an outline overview of the operational flexibility of modern combined cycles, which includes a novel bypass valve component in the topping cycle's combustion chamber. So, a proposal is proposed where in this cycle system with gas turbine coupled with steam turbine at topping and bottoming cycle is designed along with efficiency of system, network and exergy losses due to topping cycle exhaust gases visually displayed.

1.1. THE PURPOSE OF THE PROJECT:

The goal of the project is to create a practical methodology for achieving thermodynamic optimization in a combined cycle power plant. As a result, major goal of this study is to propose an optimization approach using a computer software called "MATLAB" to enhance overall efficiency with and without bypassing, reheating, and reheating combined with bypassing. It would be feasible to:

1. Provide information on energy losses as well as their location using this technology.

2. Determine the most realistic operational parameter values that result in the highest feasible power output, efficiency, and steam mass, and compare the results with and without bypassing and reheating.

3. Calculate the mass of steam as a function of pressure ratio with and without bypassing and reheating in the Combined Cycle Power Plant.

1.2. Outline for Thesis:

Seven chapters make up the thesis. Thermodynamic Analysis of CCPT with Bypassing and Reheating at Different Pressure Ratios is introduced in this chapter. The study's significance and purpose are briefly explained.

Chapter 2 contains a comprehensive overview of the literature on areas connected to this research, such as thermodynamic analysis and optimization.

Chapter 3 describes about the mixed cycle gas turbine power plant system. It focuses on coupled cycle thermodynamics and its constituents.

Chapter 4 describes the theoretic underpinning of energy interpretation and thermodynamic optimization, as well as the mathematical model expression and equation are covered. Furthermore, it discusses how the optimization method employed within paper was developed.

Chapter 5 The base plants modelling chapter includes a full detailing of plant as well as an energy analysis. It explains the mathematical steps involved in solving the optimization problem.

Chapter 6 The outcomes for every step of process are presented. It also covers how the operational parameter affects the performance parameters. There were additional similarities made between the cases.

Chapter 7 describes about the study results.

Chapter 8 wraps up with the references.

<u>**CHAPTER 2**</u>

2. LITERATURE REVIEW:

A quick analysis for available literature is conducted for sensing the current strategical progress gas turbine power plants. This research focuses entirely on a literature evaluation of optimization techniques used on diverse thermal systems. Some of the study conduct the optimization of the steam plant (bottoming cycle) based on certain gas turbine while other insights into suitable method of optimization for complete power plant. The methods for optimization is studied from a respective of thermodynamic standpoint using the first and second laws.

2.1 REVIEW OF THE TOPPING CYCLE'S ANALYSIS AND OPTIMIZATION:

2.1.1 THERMODYNAMIC ANALYSIS AND OPTIMIZATION:

Ambient conditions along with compressor pressure ratio along with turbine inlet temperature are operational characteristics in the gas turbine that influence combined cycle gas turbine performance.

2.1.1.1 IMPACT OF AMBIENT CONDITIONS:

The ambient conditions consisting of ambient temperature along with atmospheric pressure plus relative humidity of air, are one of the parameters that influence gas turbine performance. During operation, these characteristics have an impact on the yield of electric power along with the rate of heat produced. The geographical region of a power plant influences its performance significantly. After compression, the ambient air goes through the compressor and is routed to a combustion chamber, where it grows hotter.

Arora and Rai [1] According to the combined cycle power plant, the compressor along with combustion chamber plus gas turbine, HRSG, boiler, steam turbine, and heat generator are all included. The steam turbine inlet temperature was found 540°C, the exhaust pressure is atmospheric. Input temperature is limited, and this cycle efficiency is roughly 40% due to design considerations. In gas turbine, the inlet temperature of turbine is around 1100℃ but its exhaust temperature can be brought down to roughly 500℃ to 600℃, the effectiveness of turbine comes around 33%. The steam turbine can be powered by the gas turbine's exhaust, outcoming as 60 percent efficiency.

Ibrahim and Rahman [2] While studying the combined cycle gas turbine, such as air fuel ratio, isentropic compressor and efficiency, compression ratio, and gas turbine peak temperature ratio, the thermodynamic analysis parametrically was carried out and the impacts of the operational parameters were studied. The results show that the air-to-fuel ratios, compression ratios, and isentropic efficiencies all have an impact on the overall thermal effectiveness of this cycle power plant. The total efficiency of the plant improves when the compressor ratio, turbine efficiency, and isentropic compressor improve. The overall thermal efficiency varies only slightly at lower compression ratios, but the isentropic compressor and turbine efficiencies differ dramatically at higher compression ratios. When comparing the efficiency of both the gas turbine along with the steam turbine separately, the combined cycle power plant's overall efficiency is significantly higher. Around 63 percent efficiency was discovered. The thermal efficiency & power output of a combined cycle power plant grows and declines linearly as the compression ratio increases while keeping the turbine inlet temperature constant.

Ameri and Hejazi [3] The 170 gas turbine units in Iran lose 25% of their rated capacity when the ambient temperature changes. The power output was found to be reduced by 0.76 percent for each 1°C increase of ambient temperature for five different gas turbines wherein the variance aspect of both the ambient temperature along with the ISO conditions was moderately 11.6 °C. As a result, it was concluded that the compressor intake air temperature should be cooled for improvement in efficiency of gas turbine.

Hosseini et al [4] showed that for constant air volume flow the gas turbine compressor is designed for a certain mass flow rate, electric power production is affected by ambient temperature. He also said that its compressor outlet pressure is reduced due to the rise of ambient temperature thus reducing the effectiveness of gas turbine whereas the enhancement of air density increases its specific fuel consumption and reduces the gas turbine heat rate. The electric power of a gas turbine reduces by 0.5 percent to 0.9 percent, and by 0.27 percent for a combined cycle, each and every 1°C increase in ambient air temperature.

2.2. REVIEW OF ANALYSIS AND BOTTOMING CYCLE OPTIMIZATION:

2.2.1 . THERMODYNAMIC EVALUATION AND OPTIMIZATION:

The efficiency of a steam power plant using the Rankine Cycle can be enhanced by raising live steam and reheat steam characteristics, along with incorporation of high-efficient and low-loss turbine blade shapes. For optimization of the steam parameters, we must choose the right stuff for the components that work with live steam and reheat steam. Development of steels with the improved creep properties have been made possible for the appropriate manufacture of the key components through the Collaborative European Programs. In addition to that, major improvements in the overall has been made possible through the optimization of the geometries and the blade profiles. For the increase in temperature of 10°C, live steam and reheat steam, the improvement in efficiency is found out to be 0.50% and for the increase in pressure of 10 bar, the improvement in efficiency is found out to be 0.20%. Furthermore, the effectiveness of this power plant is influenced by the design and optimization effectiveness of the Heat Recovery Steam Generator - HRSG for the bottoming cycle.

Mohagheghi and Shayegan [5] Thermodynamic optimization of the heat exchanger and design variables in HRSG for a combined cycle power plant was carried out utilizing the genetic algorithm. This optimization was carried out using nonlinear equations and their simultaneously employed solutions, which are used in sophisticated combined cycle steam cycle modelling. The placement of heat exchangers in the HRSG was optimized once they were distributed throughout the different parts. Here they are considering a gas turbine and for the recovery boiler model the inlet parameters are same as the parameters for the outlet gas stream for the gas turbine cycle like temperature, the mass flow rate and gas stream chemical composition. Finally for the different HRSG, the maximum power output from the steam cycle was analyzed after the optimization process.

Woudstra et al [6] Using the same gas turbine, but with distinct steam bottoming cycles carried out the analysis of thermodynamic for this plant. The results elucidated that the loss of energy is reduced as a result of flue gas exhausting onto the stack and the heat transfer losses in the HRSG will reduce due to the result of the increased number of pressure level in the steam production. Furthermore, it was found out that from the exergy point of view triple pressure reheat in the bottoming cycle was the best option.

Mansouri et al [7] has shown how the pressure magnitude for steam production at the HRSG vary with the HRSG, bottoming cycle, and combined cycle power plant's exergetic and energetic efficiency. The findings revealed that a soar in exergy efficiency of the HRSG and CCPP occurs due to a rise in the pressure levels of steam generation at the HRSG. Aside from that, the exergy destruction in the HRSG lowers down as the pressure levels rise owing to heat transfer, and the HRSG's energetic efficiency rises as the steam generation; pressure rises and reheat is added to the cycle.

Bracco and Silvia [8] investigated and then created a mathematical version of this cycle plant for calculation of ideal steam pressure reading for HRSG on the basis of many objective functions. The transfer of heat between the stream and exhaust for HSRG is depended on energetic analysis. In order to assess the impact of the temperature of gas for the HRSG inlet and the temperature variability at the pinch point on the objective functions under consideration. In addition, a mathematical evaluation was conducted. The presence of special limits for the power plant's operating characteristics was taken into account in their mathematical model. For the mathematical modelling, other essential constraints such as steam standard at the turbine outlet, blade height for the turbine of steam and HRSG outlet exhaust gas temperature were taken into account.

Xiang and Chen [9] A three-pressure HRSG combined cycle powered by a GE PG9351FA gas turbine was considered in which the effectiveness of the combined cycle was maximized by the HRSG's operating parameters should be optimized in which the loss of energy was minimized. The impact of the temperature of the HRSG incoming gas for bottoming cycle efficiency was being highlighted and the effect of the temperature of the HRSG inlet gas on the efficiency of the steam bottoming cycle was studied by them. It was discovered that raising the HRSG inlet temperature above 600°C had a lower influence on steam efficiency.

2.3. REVIEW OF OPTIMIZATION OF WHOLE CCPP:

Tyagi and Khan [10] They looked at the impacts of ambient temperature, exhaust temperature of gas turbine plus stack temperature for total effectiveness of the power plant while managing the value of gas turbine plus steam turbine efficiency constant. After researching the impact on the Power Plant total efficiency, it was set that the gas turbine exhaust temperature should be the highest and the stack temperature should be the lowest. Out of the three elements mentioned above, the stack temperature is most significant for enhancing the efficiency of power plant.

Boonnasa et al [11] studied about an established gas cycle power plant situated in Bangkok. It consists of each with two gas turbines of 110.76MW power output and a steam turbine of power output 115.14MW in ISO condition. In order to chill the intake air of one of the two gas turbines to 15°C, an absorption chiller is used by the CCPP. Besides that they have thermal energy storage tank used to meet the varying daily cooling load by storing the sensible heat of the stored water. With the help of the thermal storage an absorption chiller is for satisfying a maximum load of 7049.58kW, it is powered by low pressure steam generator. Hence total power output of the CCPP is improved by 6.24% and the cooled gas turbine's power output has been enhanced by 10%.As a result of connecting the absorption chiller to the HRSG unit directly, steam is generated. The steam turbine was being powered, it was reported by the authors that the steam turbine power output was being reduced by 2.87%.The loss in steam turbine power result may have been prevented if a boiler uses heated waste energy from stack after the HRSG unit.

Bassily [12] The impact of Pinch Point altering the gas turbine's inlet temperature on the dual pressure reheat cycle's performance were discussed. The irreversibility of the Heat Recovery Steam Generator of both Brayton cycle and Rankine cycle was also reduced. The parametric methodologies reveal that optimizing and lowering the irreversibility of cycle can boost their efficiency by 2 to 3%. When we use gas reheat, the moderate temperature of heat delivery and the generated power both rise. In order to improve cycle efficiency, we must employ gas recuperation, which allows the gas turbine outlet to boost efficiency of gas. For the gas reheat and gas recuperator combination cycles, stainless steel recuperated heat exchanger are built to endure these conditions. The optimized results are then compared to the triple reheat combined cycle as it is normally configured.

Sipeng Zhu [13] did a theory-based study of the thermodynamic method for bottoming Rankine cycle for recovery of heated engine waste and found that the main factors influencing the design and execution of this plant were evaporating pressure, superheated temperature, and working fluid properties. Under typical operating conditions, the global heat recovery efficiency was less than 0.14.

Wenguo XIANG et al [14] In China, researchers looked into how to improve the combined cycle gas turbine's efficiency. A discovery reported that when the input temperature exceeds 590°C, the steam cycle efficiency improves the least. They discovered that when the Heat Recovery Steam Generator (HRSG) optimization is paired with heat recovery, combined cycle efficiency rises to 59.05 percent. Few tools are illustrated for evaluating potential designs of combined cycle plants.

Ghazikhani et al. [15] the injection of steam in the gas turbine modelled with the Air Bottoming Cycle along with the performance of the Air Bottoming Cycle is investigated via calculation of the irreversibility in the systems related devices. They propose two new cycles in which steam injection is used in the Air Bottoming Cycle. Steam Injection Gas Turbine used with Air Bottoming Cycle (STIG-ABC) and The Evaporating Gas Turbine used with Air Bottoming Cycle are the two cycles (EGT-

ABC). The results of these two cycles reveal a higher air inlet mass flow rate and better energy recovery, which translates to an increase in effectiveness and turbine work output. When compare to the STIG-ABC cycle, the EGT-ABC cycle has higher work output and lesser irreversibility due to better heat recovery in the regenerator resulting in lower exhaust temperature. As a result, it was determined that with the same ratio of pressure and Turbine Inlet Temperature. EGT-ABC cycle enhances effectiveness of the plant.

Ghamami et al. [16] by completing an energetic and exergetic study of the Power Plant, proposed a strategy to upgrade the effectiveness and output of the gas turbine They discovered that the gas cycle's energy and exergy efficiency, as well as its net power output, have improved, while air pollution levels have decreased.

Chmielniak et al. [17] discussed that in the gas turbine air bottoming cycle what type of air heat exchanger should be preferred. To enhance the effectiveness of the gas turbine air bottoming cycle bigger size of the air heat exchanger should be preferred.

Polyzakis et al. [18] The power plant, which has four separate gas turbine cycles, was first analyzed and subsequently optimized. Simple cycle, inter cooled cycle, reheated cycle, and reheated and inter cooled cycle are the four types of cycles. The reheated gas turbine was discovered to be the most attractive among these cycles, resulting in a more effective combined cycle power plant.

V. Macian et al. [19] The bottoming cycle, which is considered as waste heat recovering system in the vehicles, was researched and then performed parametric analysis through the method of optimization. They walk two specific examples through the process for assessing the preliminary energetics of the bottoming cycle implementation. The fundamental reason for our preference for a gas turbine-based power plant over a steam-based power plant is its lower relative cost. As a result, one of the most essential technologies currently in use is the combined cycle power plant.

Iacopo et al. [20] A thermodynamic analysis designed to match a vapour cycle efficiently was introduced by them and then a study was performed with a second law analysis and based on the research findings it was revealed that there has been an improvement in overall efficiency of combined cycle power plant to about 12%.

Adrian Tica et al. [21] The efficient approach was used to study a physical CCPP model that is built for replicating the optimization model. The study's method centered on the smooth model's formulation of the plant represented with discontinuity sources.

Kehlhofer et al.[22]The gas turbine plant was considered an important component of power plant, and it was discovered that increasing gas turbine effectiveness will not result in the combined cycle power plant being more efficient.

El-Masri et al.[23] identified the irreversibility's that cause the combined cycle power plant to lose work production, thermal efficiency, and overall efficiency of the steam turbine. He also discovered that inter cooling the compressor increases specific work production while lowering thermal efficiency. As the turbine inlet temperature rises, the balance of increased turbine blade cooling losses and lower combustion exergy losses shifts, resulting in the combined cycle power plant's efficiency being dominated.

Kail et al.[24]It is not possible to convert the efficiency and advantages at the output into lower electrical power prices while conducting reheating in the gas turbine. The increased maintenance costs and further investments outweigh the thermodynamic gains. For the design and quality of cooling system and its steam, steam cooled turbine blades place stricter conditions on the materials of the blade. Simple gas turbine power plants or simple steam power plants are considered the best facilities from an economic standpoint since they achieve the lowest cost of electrical power.

Rufli et al.[25]Using simple thermodynamic calculations, we examined the combined cycle power plant first thereafter turbine cycle and its steam. The maximum temperature of gas turbine was regulated between 900°C and 1400°C while the gas turbine pressure ratio is controlled in between 8 and 22. Rufli's clear and simple calculations yielded the total heat transfer area and power plants' thermal efficiency. Rufli has illustrated a methodology for choosing the best factors for steam running in a combined power plant cycle under any provided gas turbine functions circumstances.

Cerri et al. [26] analyzed the cycle plant and put forward thermodynamic factors or indices for assessing the plant's performance. The utmost inlet temperature of the gas turbine was varied between 800°C and 1500°C while the pressure ration of the gas turbine was varied between 2 and 25. Aside from that, they have thought about what they will do after the fire is out. He calculated the cycle's thermal effectiveness and output of the specific concluding that the turbine pressure ratio has no impact on the plant's thermal efficiency, but the thermal efficiency may change somewhat if influenced slightly by the high steam pressure. If the inlet temperature of the turbine is crucially less and an afterburner is gradually added, the result will have positive impact on the thermal efficiency of the combined cycle power plant.

Bhinder and Mango et al. [27] The thermodynamic analysis was used to investigate the combined cycle power plant's performance. They arrived to the conclusion that the combined plant efficiency was considerably more as compared to effectiveness of gas or the steam turbine. It has been discovered that a combined cycle power plant with overall efficiency of roughly 60% is possible. Because many losses were not included in the computation, the combined cycle calculation was determined to be straightforward. It would be impossible to reach a 60 percent overall effectiveness if losses factored into the calculations.

Horloock et al. [28] followed through on a thorough investigation of combined cycle power plants, learning about their early history and describing the latest advancement and future prospective of gas turbine and steam turbine facilities separately. A graphical method was used to forecast the performance of the gas turbine and steam turbine cycles. He also identified gas turbine's ideal pressure ratio for best combined cycle power plant efficiency.

Sarabchi and Polley et al. [29]The influence of operating variables was first analyzed and then evaluated, and then it was compared to simple gas turbine cycle. They altered the ratio of compressor pressure and the pressure of the recovery boiler for each inlet temperature of the gas turbine, then looked at the specific network production and thermal efficiency. They discovered that combined cycle maximum efficiency occurs at pressure ratios that are significantly lower than those required for the matching simple gas turbine to achieve maximum efficiency at the given turbine inlet temperature. Increasing inlet temperature of gas turbine,values of heat recovery boiler pressure and the optimal pressure ratio increases and for the utmost work output the optimum pressure ratio for the combined cycle gas turbine is equivalent to the optimum ratio of pressure for the simple gas turbine.

Bannister, Cheruvu, Little, & McQuiggan et al. [30]conducted the procedures vital to reach energy conversion efficiency of more than 60%. They suggested that the operating parameters for gas turbine plus steam turbine power plants be improved by increasing the inlet temperature of the gas turbine to 1427°C, utilizing heat losses in both cycles through more integration between the two power plants, upgrading component effectiveness and commencing advanced cooling techniques in the gas turbine.

Gülen and Smith et al.[31]did research on the combined cycle's Rankine Bottoming Cycle efficiency. A CC-RBC performance model based on the exergy concept and the second law was developed by them. For a given exhaust temperature of the gas turbine,a CC-RBC The engineer might be able to use a performance model to precisely estimate the Rankine Bottoming Cycle's performance.

Gulen S. C. et al.[32] Gülen and Smith's work was continued in order to exploit the effect of consumption on power plant effectiveness. The heat rejection systems and boiler feed pumps are the two main grantor of auxiliary systems.

Bassily et al.[33]The impact of differing both ambient temperature along with relative humidity on the operation of combined cycle power plant with different input cooling systems employed in the gas turbines was investigated. When compared to refrigeration and absorption gas turbine inlet cooling systems, the results showed that introducing gas turbine inlet cooling approaches is more suitable for hot and humid conditions, with a 1.4 percent increase in power production and a 1.2 percent increase in combined cycle efficiency.

Andreades, Dempsey, and Peterson et al.[34]The topic of ext. heat to plant cycle was explored, as well as the modifications that were required. When the open-air design is employed in reheat air combined cycle power conversion in order to increase revenue for the operators, the option for injection of natural gas or other fuel for boosting power during extreme necessity gives the electric grid with contingency and flexible capacity. Compared to stand-alone natural gas combined cycle and peaking facilities, this combination offers a number of significant advantages.

Da Cunha Alves, et al.[35] introduced the concepts of inter cooling and reheating for gas turbines in a cost-effective manner by utilizing a model derived from turbo machines and blade cooling to assess their influence on engine performance. They came to the conclusion that inter cooling improves overall efficiency more than the simple cycle at higher pressure ratios, whereas reheating is better suited to combination cycles.

Kehlhofer et al.[36] He investigated the gas along with steam turbine in plant and discovered that increasing turbine effectiveness did not always result in the best combined cycle power plant efficiency.

Kaikko et al.[37] He introduced the economical concept of the air bottoming cycle in order to improve the power generation effectiveness of small along with medium sized gas turbines. He presented a thermodynamic analysis for the cogenerative system, in which the fraction of intercooled air from the air bottoming cycle is routed via the reversed brayton cycle to give cold air flow.

Korobitsyn et al.[38]The plant was first examined, followed by research into the twin gas turbine cycle's performance. He analyzed dual gas turbine cycle with various topping gas turbines, then implemented the air bottoming cycle at the gas turbine, resulting in a net power output gain of 20% to 35% above the steam bottoming cycle. The simpler and more study design is one of the setup's distinguishing qualities. In numerous applications, such as an upgrade option for small gas turbines installed at offshore sectors, he used compact and simple bottoming cycle. He assessed the air bottoming cycle's economic and technical viability, which uses hot air from a gas turbine to cool the water. The food processing industry receives turbines.

Bolland et al. [39]The air bottoming cycle was subjected to a thermodynamic study, and the results revealed that it is the most cost-effective option when compared to the other options.

Najjar et al.[40] The researchers discovered that the air bottoming cycle can attain a thermal efficiency of over 60%, which is not possible in ordinary gas turbine cycles.

Wicks et al.[41] The air bottoming cycle notion is acquired from the theory based on ideal fuel burning engines through compare analysis of the engines to the Carnot cycle. The air bottoming cycle can be utilized to utilize the heat rejected by the gas turbine.

Sipheng Zhu et al.[42] A theoretical study of the Bottoming Rankine Cycle was done by him so as to recover the heat of the engine waste by comparing with the five different working fluid on the Mat lab software. The factors that affected the engine performance are the working properties of the fluid and super heating temperature which caused a little change in the overall efficiency of the cycle.

Khan et al.[43] Bypass valves utilized to improve the effectiveness of power plant. The cycle plant network output rose by around 45 percent when the gas turbine's input temperature was raised from 1000 to 1400 degrees Celsius. It resulted, the combined cycle power plant's overall efficiency improved from 15% to 31%. The overall efficiency of the combined cycle power plant could be boosted by providing steam cooling and sequential combustion cooling.

Kumar et al.[44] The two ways of waste heat recovery system of a gas turbine was proposed by him in order to generate electricity. In the first method, he used a single heat recovery boiler in which all the mixed gases were passed through before entering the grid cooler and the pre heater where as in the second method, he allowed the vapour mixture to pass through the steam turbine. After comparing both the methods it was found out that in the first approach 23931kJ/s of heat was recovered with 23.5% power generating efficiency where as in the second approach 21253kJ/s of heat was recovered with 22.2%power generating efficiency. He also created a simulation model for calculating the total efficiency and steam flow rate of a 250MW coal-fired power station. He went on to examine how variations in load affect overall efficiency, and discovered that the overall efficiency of the plant was marginally impacted by the increased load.

Ahmadi et al.[45] He conducted an assessment of solar power technologies for the generation of electric power in the combined cycle power plant. The solar energy can be used for generating electricity by the two methods, one by using the PV model to convert the solar irradiations into the electricity and to harness the thermal energy, the other method is to use Concentrated Solar Power Plants. After performing the two methods it was found out that the PV Model Power Plants are the better options as compared to the Concentrated Solar Power Plants.

Locopo et al.[46] analyzed the three separate cycles using the thermodynamic second law viz.,addition of the cooling water with the engine exhaust, a cycle that uses the exhaust gases produced by the engine and at last with a regenerated cycle. For the cycle without the bottoming, the combined cycle power plant's overall efficiency has been raised by 13%.

Hawaj & Mutairi et al.[47]The pressure ratio of the compressor, the gas turbine's combustion efficiency, the steam turbine's inlet temperature, steam to gas mass flow rate ratio were all examined. In another area, he showed how the combined cycle power plant with absorption cooling outperformed the thermally comparable mechanical vapour compression cooling technology. The reheated cycle is shown to be the best cycle when compared to the intercooled cycle and the single stage cycle.

Butcher & Reddy et al.[48] In accordance to thermodynamics, second law, execution of the waste heat recovery power generating system for various operating situations was evaluated. He also modeled the network output, second law efficiency, entropy generation number, throughout the heat recovery steam generator (HRSG), and other parameters under a variety of operating situations.

Kakaras et al.[49] The three methods of air cooling, namely evaporative cooling along with refrigeration cooling plus evaporative cooling of pre-compressed air, were refined. He got to the conclusion that absorption intake air cooling generates the most additional power, while the evaporative cooler illustrate low cost based incremental electricity production and the shortest payback period based on the investment's economic performance. When we evaluate the pre-compressed air cooling approach, however, the results show that the total cost of incremental.

CHAPTER 3

3. INTRODUCTION:

Amidst the mid-1960s, the eras of the combined cycle power generation systems were introduced that included conventional fired boilers. The adjustments of the conventional steam plants were done through the exhaust gases from the gas turbine exit are used as the boiler's combustion air. As compared with the traditional steam plants,the combined cycle power plant's efficiency was discovered to be 5% to 6% higher. Because the combustion products and steam/water have such a large mean temperature difference, combined cycle power generation systems can afford to employ the exposed tubes in the boiler. The implementation of the heat recuperation type of the combined cycle systems emerged to be the most prevalent as its primary application is in the heat and power applications where the energy to heat proportion is great. Besides this the implementation of the heat recuperated type combined cycles were introduced in the applications of the utility power generations and these frameworks are then utilized into the heat recuperation feed water heating combined cycle as an innovation which is developed for the base load and mid-extend benefit.

Abundance of combined cycle power plants were constructed around 1990, including many built for base load operations. Five years later, the era of massive gas turbines arrived on the scene, bringing with it a combined cycle power output of 350-400MW and a 57-58 percent efficiency. The combined cycle power production then increased to roughly 570MW around 2011, with a 61 percent efficiency.

3.1. Description of the Components of Combined Cycle Power Plant:

3.1.1. COMPRESSOR:

Axial flow and centrifugal flow compressors are different types of gas compressor turbines. Air streams in axial flow compressors are aligned and parallel to axis of rotation and air streams in centrifugal flow compressors are radially directed away from the hub.

The optimal yield of the gas turbines is ensured by the larger pressure ratios. When similar diameters are taken it is found that unlike a centrifugal stage compressor, axial stage compressors do not offer a larger pressure ratio. In order to obtain more prominent pressure ratios, it is possible with the help of the multistage axial compressors. The bigger mass flow rates and the more power output yielded is permitted by the multistage axial compressors with the help of the improved pressure ratios.

The compressor's primary job is to coordinate and quicken mass flow rate of air to flow into the gas turbine. The compressor works on a few stages of rotating blades on the plates and on the stationary vanes. The air is quickened towards its stage's vanes and its trailing edges by each row of the rotating blades. The air is slowed down by the stage's vane and is then guided into the following stages of the blades.

Then, at each level of the compressor, the air is compressed and quickened and then it passes into the diffuser area. At the most extreme compressor pressure, the air has reached its greater speed. The diffuser provides an expanding cross section area throughout its length with its prime purpose is to increase its static pressure and decrease the air’s speed moderately. The extreme value has been achieved by the air at the diffuser outlet and then finally enters into the combustion chamber.

The captious point of view that should be contemplated is the nearby atmosphere and the elevation of gas turbine and density of air drops as height, temperature, and humidity rise, and as a result, the surrounding temperatures can have a major influence on the gas turbine's output yield. The weight of the air at a given volume will be lower with the increase in high temperatures, high elevation and high humidity as a result of which there will be less necessity of the fuel and will produce the lesser yield of the power output.

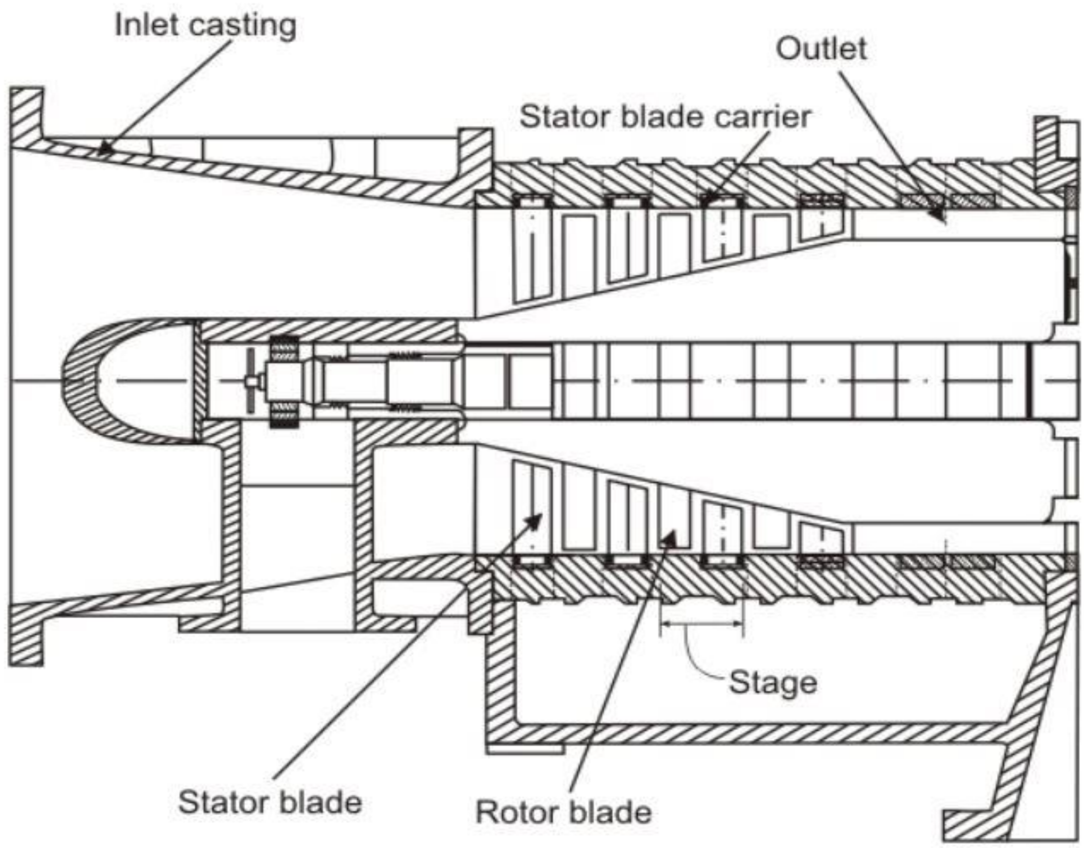

Fig.3-1 AXIAL FLOW COMPRESSOR

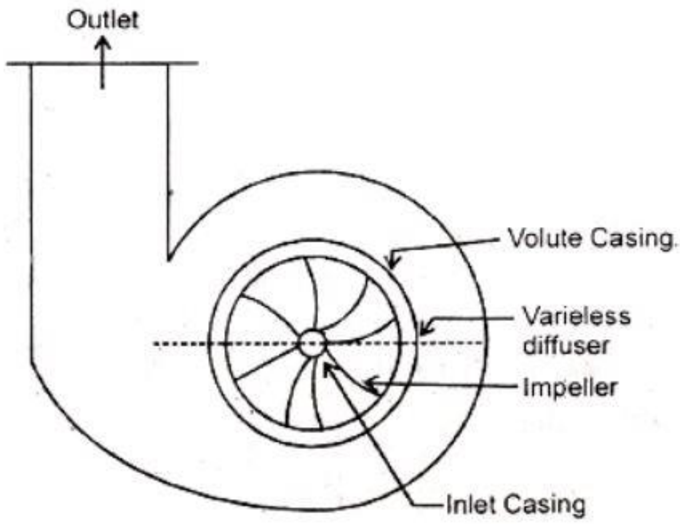

Fig.3-2 CENTRIFUGAL FLOW COMPRESSOR

3.1.2. COMBUSTOR:

The combustor consists of a few segments like ignition plugs, combustion chambers and the fuel nozzles. Combustors consume the fuels, air blends and the mixture of the both and then carrying the burning products to the turbine bay at its design parameters.

In the beginning of the cycle, the combustor starts through the igniter plugs that is located at the burning liners closer to the fuel nozzles. The burning is started by the flames through the igniter and then spreads to the next combustion cans.

The three fundamental sorts of the ignition chambers: can, annular and can-annular are utilized by the combustor. Spray the fuel into the ignition chambers through a simple type of the fuel nozzle that carries either the liquid or the vapor fluid or through the double fuel nozzles that carries the liquids or the gas at different circumstances all the time. Bi-fuels are carried through the combustor in which both the liquid and the gaseous fuels blend at the same time.

It is required to blend air with the fuels from the fuel nozzles that is utilized through the whirled blades. Here the primary air is defined as the air that flows over the twirled blades and the primary constitutes 25% of the aggregate air that is destroyed by the gas turbines. The remaining air is viewed as the secondary air that is utilized to cool the burning gases to the temperatures feasible for the turbine inlet conditions.

3.1.3. GAS TURBINE:

It is a mechanism that transmits mechanical energy or thrust. A vaporous working liquid is used to generate mechanical thrust or power, and this power can be created. The gas turbine operates with a constant flow of working fluid. In compressor, working fluid is compressed, then heated in combustion chamber, and finally expanded in the gas turbine. A turbine, in general, is a machine that generates work.

Here the, energy is directly converted into the mechanical work and some amount from this is utilized for compressor functions whereas the rest work can be used in the plant. The working of the gas turbine relies on working of the different parts i.e. compressor, combustion chamber plus turbine.

The gas turbine driven via Brayton Cycle methodology. As shown in Figure 4, air taken into the compressor at ambient conditions and then compressed into compressor, where it is pressurized up to the pressure P2, thereby increasing the pressure and temperature and allowing the compressor to deploy the work provided by the turbine in the Brayton cycle. After being compressed to a high pressure, the air is directed into the combustion chamber, for ignition of fuel at constant pressure. The additional heat provided to the combustion chamber raises the temperature of the gas from T2 to T3, which raises the temperature of the gas. After passing through the combustion chamber, hot gases enter the turbine, delivering power to the turbine that is pushed by the compressor, resulting in net work that expands the turbine to ambient pressure. The heat is finally rejected to the surroundings after the expansion operations in the turbine. Axial flow compressors are mostly employed in gas turbine power generation. Because the exhaust gases produced in the turbine cannot be recirculated, this system is classified as an open cycle. These gases are expelled. The compressor's pressure ratio and the gas turbine inlet temperature are two most important characteristics.

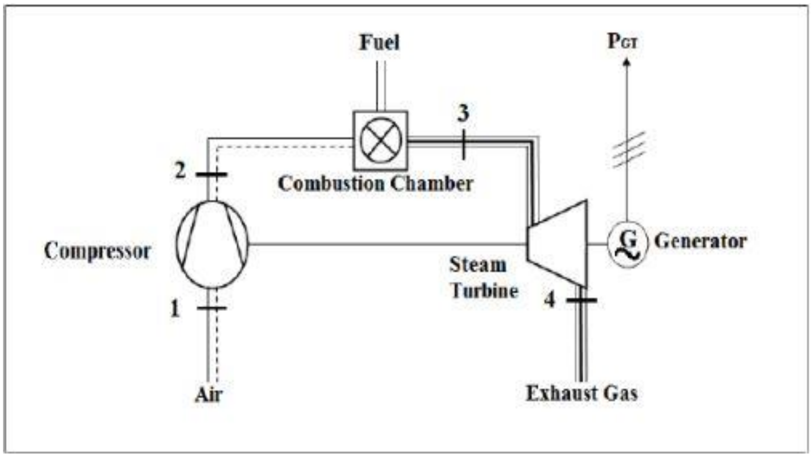

Fig.3-3 The schematic gas turbine

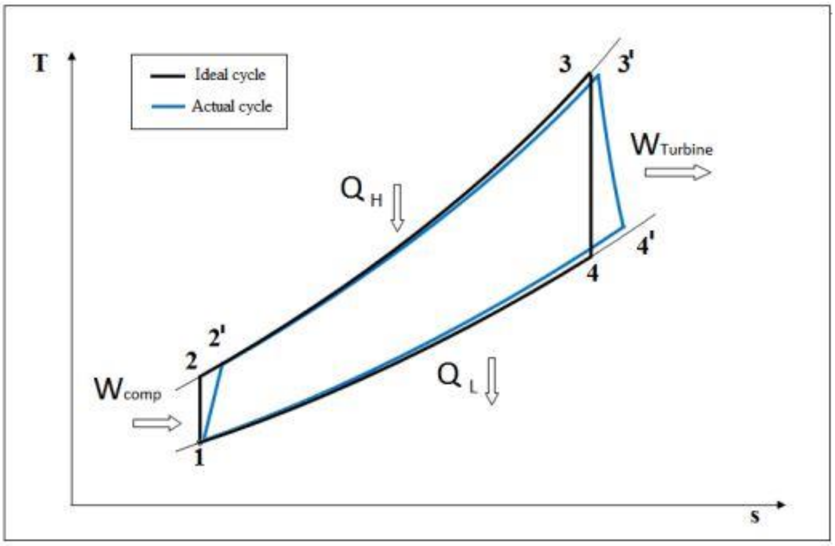

Fig.3-4 The Brayton Cycle

3.1.4. HEAT RECOVERY STEAM GENERATOR:

Gas turbine hot gases is used by heat recovery steam generator for production of steam for the steam cycle. Its efficiency has a higher impact on combined cycle power plants' total efficiency. There are three main types of heat exchangers in Heat Recovery Steam Generator (HRSG). Economizer, evaporator, and super heater are three components. The super heater, evaporator, and economizer all receive the combustion gases. In three processes, heat can be recovered from the gas side and transferred to water steam:

- Feed water must be heated to near saturation temperature in the economizer.
- Evaporation of water should be at constant temperature, pressure and saturated steam in the evaporator.
- It is compulsory to use the heat of the steam from the exhaust in the super heater in order to use the superheated steam from the evaporator, which is then reinforced to the steam turbine.

It can be divided based on the generated steam pressure (see fig.3-5, fig.3-6, and fig.3-7) and the circulation system (see fig.3-8).

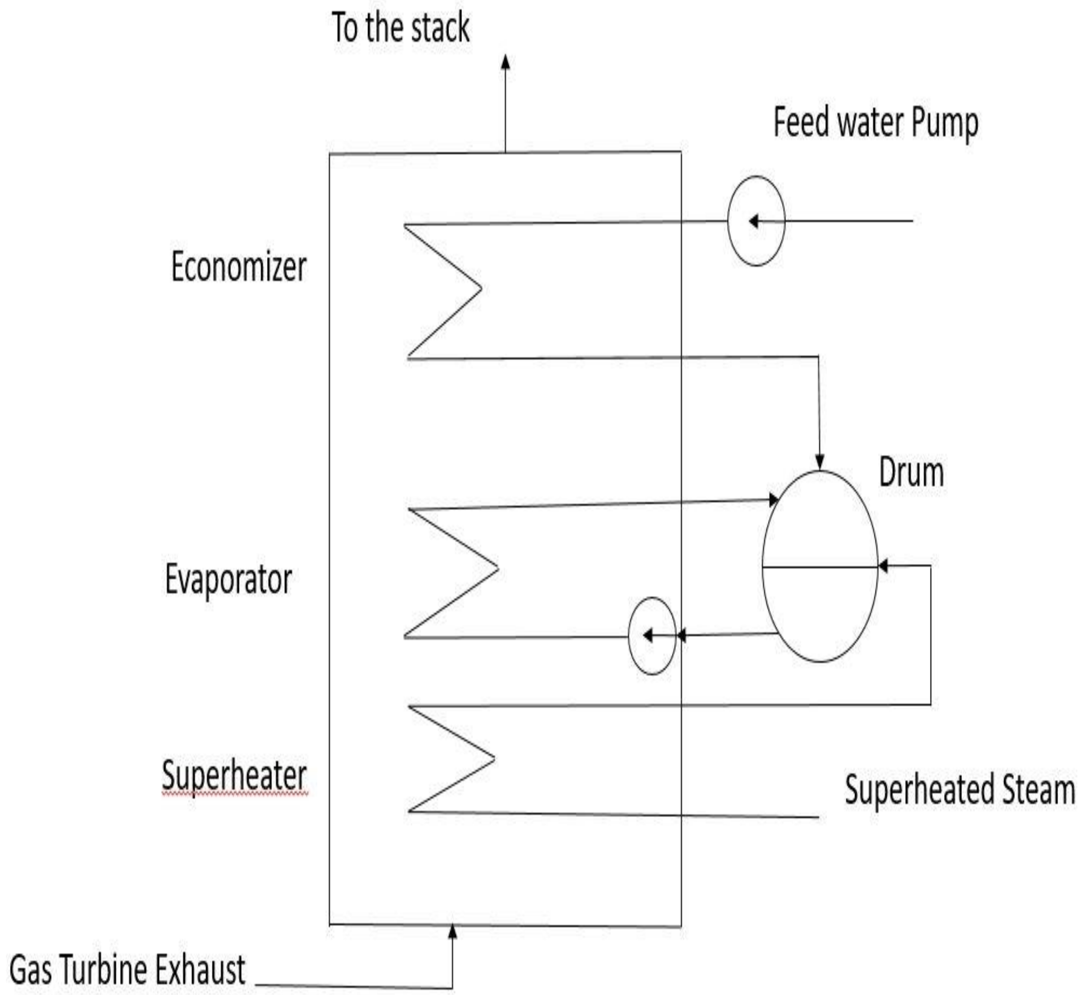

Fig.3-5 Single Pressure HRSG

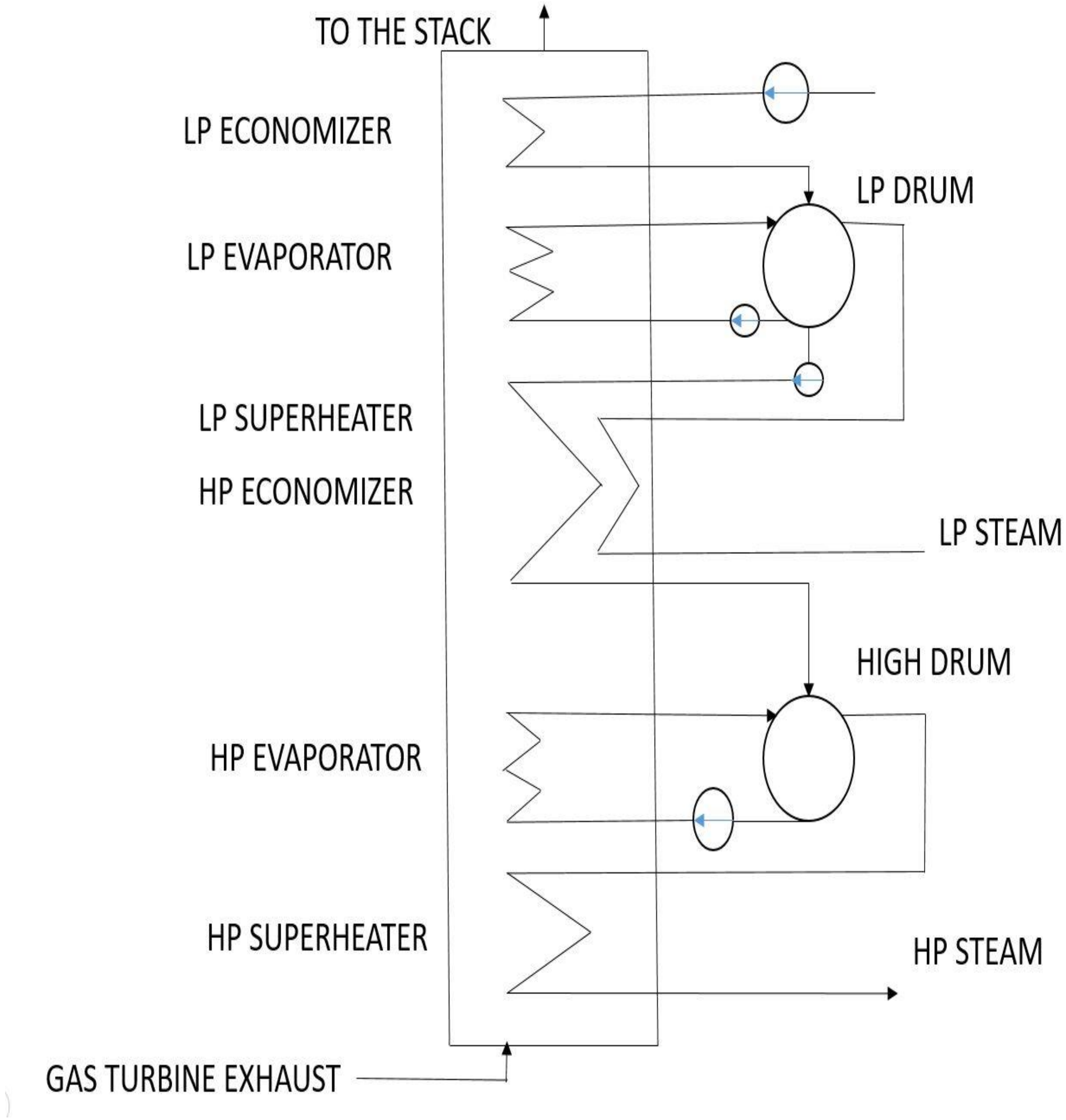

Fig.3-6 Dual Pressure HRSG

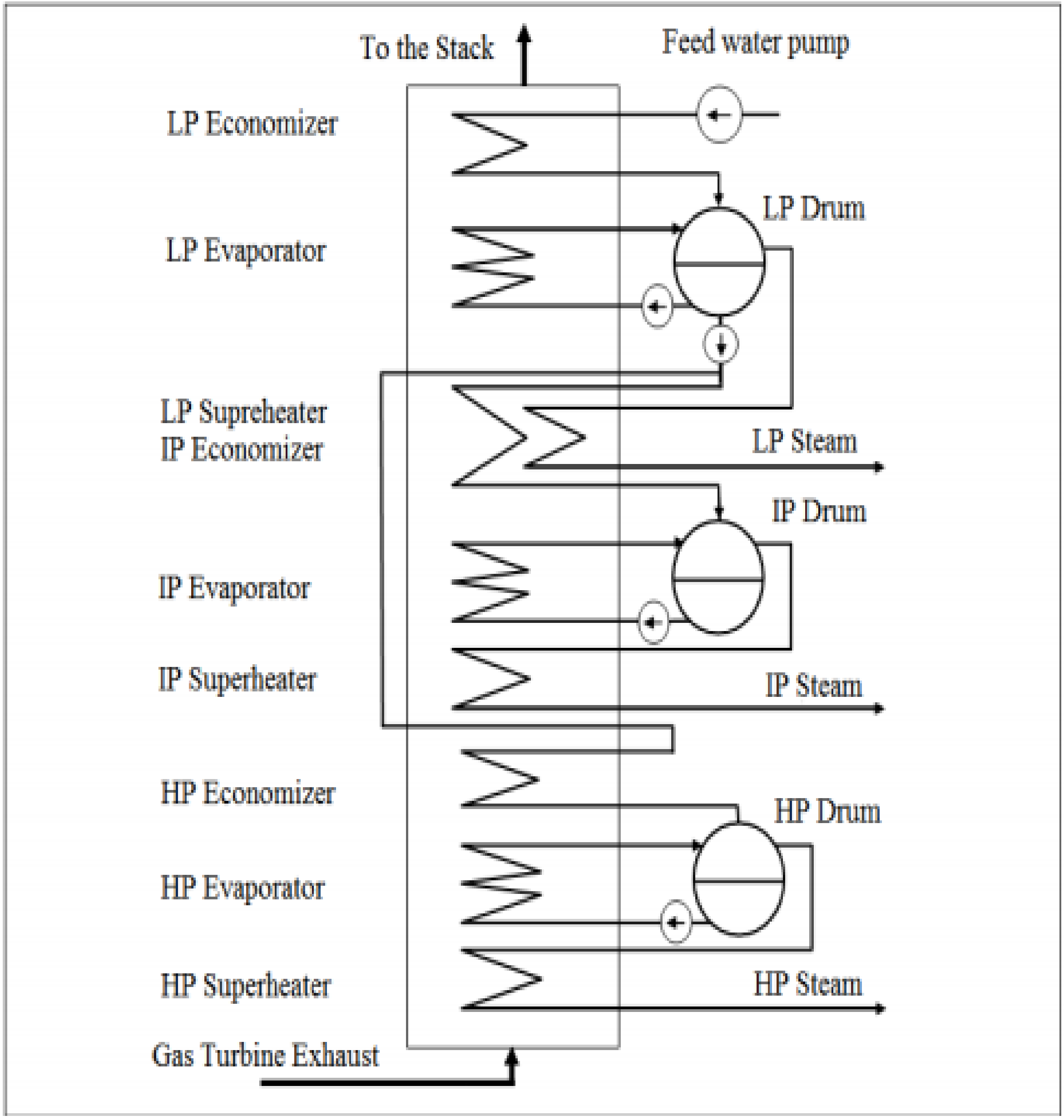

Fig. 3-7 Triple Pressure HRSG

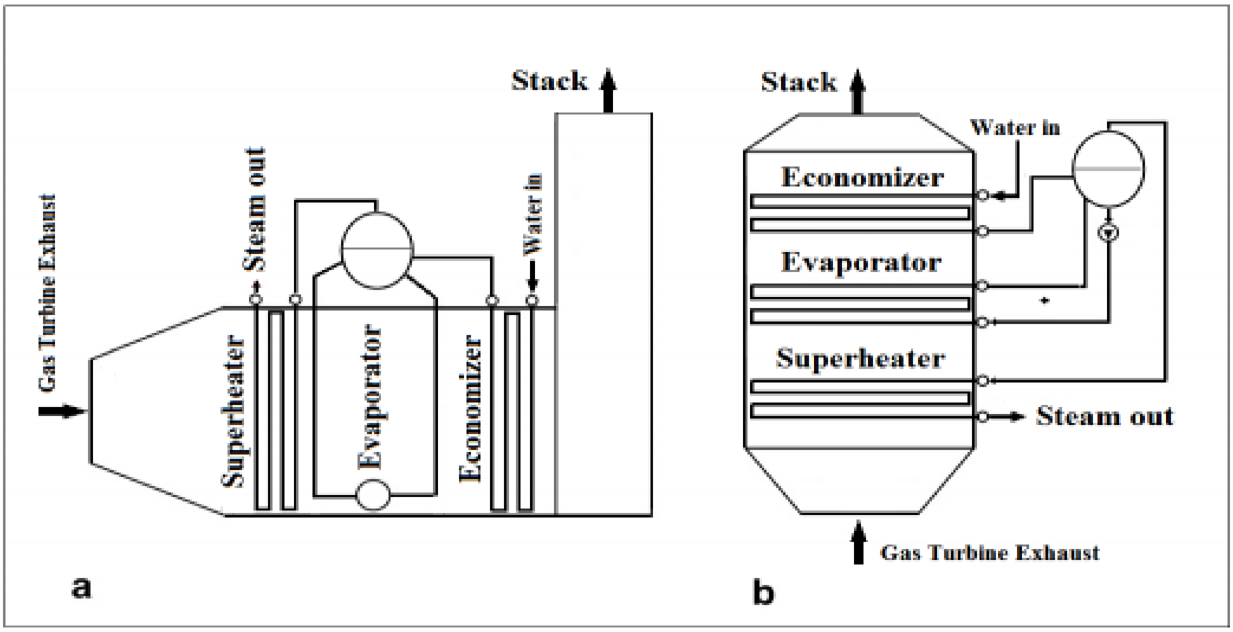

Fig.3-8 (**a) Natural Circulation** **(b)Forced Circulation**

The differential in density of the hot steam water mixed in the evaporator and the cold water in the bottom chamber causes circulation.

Natural circulation is made up of vertical and horizontal tubes that are arranged in a flow pattern as illustrated in the diagram.

HRSGs are analyzed based on horizontal tubes consisting of vertical gas stream flow and pumping for circulation of steam-water mixture.

3.1.4.1. DESIGN PARAMETERS OF HEAT RECOVERY STEAM GENERATOR:

When developing a gas turbine heat recovery steam generator, the given parameters should be considered.

(a)Pinch Point Temperature:

It is the distinction of saturated temperature of the water and temperature of gas leaving evaporator

(b)Approach Point:

The temperature variance of saturated steam along with water entering the evaporator is defined as the approach point. The size of the super heater, evaporator along with the economizer all have an impact on the approach point. As a result, the capital cost will be higher with more material.

(c) Stack Outlet Temperatures:

This is the gases temperature that are released into the atmosphere from the heat recovery steam generator.

(d)Exhaust Temperature of the Super Heater:

This is the variance of the superheated steam generator in the super heater and the gas turbine outlet temperature.

(e) The steam pressure and steam temperature are likely to define the most cost-effective design of the heat recovery steam generator.

3.1.4.2. T-Q Diagram of HRSG:

The T-Q diagram illustrate the profiles for process of transfer of heat .The usage of the T-Q diagram is critical to understand and design a combined cycle power plant. Figure 9 depicts the T-Q diagram for a single-pressure mixed cycle. The pinch point on the cooler side of the evaporator has the smallest temperature differential in the HRSG, but temperature profile of flue gases is formed by top line with a constant slope. The heat recovery steam generator is made up of three components for a single pressure combined cycle. The economizer, which heats the liquid water to saturation temperature, is located in the first portion of the HRSG. It all begins with the smallest temperature difference. To avoid evaporation and water hammering, the output temperature is kept a few degrees below the saturated condition. The 'approach point' is the two point temperature difference. Evaporator is the second segment, in which the water is cooled to a consistent temperature, and the super heater is the third section, in which the superheated steam is evaporated. The temperature and reheat relationship is provided by

$$\dot{Q} \dot{=} \dot{m}C_p\Delta T \quad (1)$$

When no phase change is suffered by the working medium, the preceding equation applies. In the evaporator of the Heat Recovery Steam Generator, the phase changes from water to steam. As a result, equation 1 will be substituted by

$$\dot{Q} \dot{=} \dot{m}\Delta h_{evap} \ldots.\ (2)$$

where Δh_{evap} is the enthalpy of evaporation and $\dot{Q}$ is the heat energy transferred. T-Q diagram consisting of slope of the line in is inversely proportionate to the mass flow rate and specific heat, according to equation (1). The Δh_{evap} is the length of the evaporation line, as determined by the equation (2).

For a combined cycle power plant, the aforementioned equations are critical. The HRSG's economizer can be found in the first part. The HRSG's first and second laws of thermodynamics are both taken into account when designing the HRSG's commencement.. The second law specifies that the flue gases' potential or exergy must be used as efficiently as possible, i.e. just a tiny quantity of entropy should be formed during the operation. To compute this, we must conduct a system exergy analysis, which will reveal the process' flaws. If the temperature difference can be minimized throughout the T-Q diagrams, the process will be able to carry out the least amount of exergy destruction/entropy generation.

A reversible process return the system and its surroundings to their original state without producing any effort, although this process is impossible to achieve in reality. As a result, there should be no temperature differential between the hotter and colder sides of a heat exchanger in order to sustain the reversible process. As a result, it is argued that irreversibility arises as the temperature difference increases. When designing a Heat Recovery Steam Generator (HRSG),

- On X axis, the T-Q diagram should be stretched long as possible to collect as much energy as possible,
- The region between the T-Q diagram lines, i.e. the temperature difference, should be kept to a minimum.

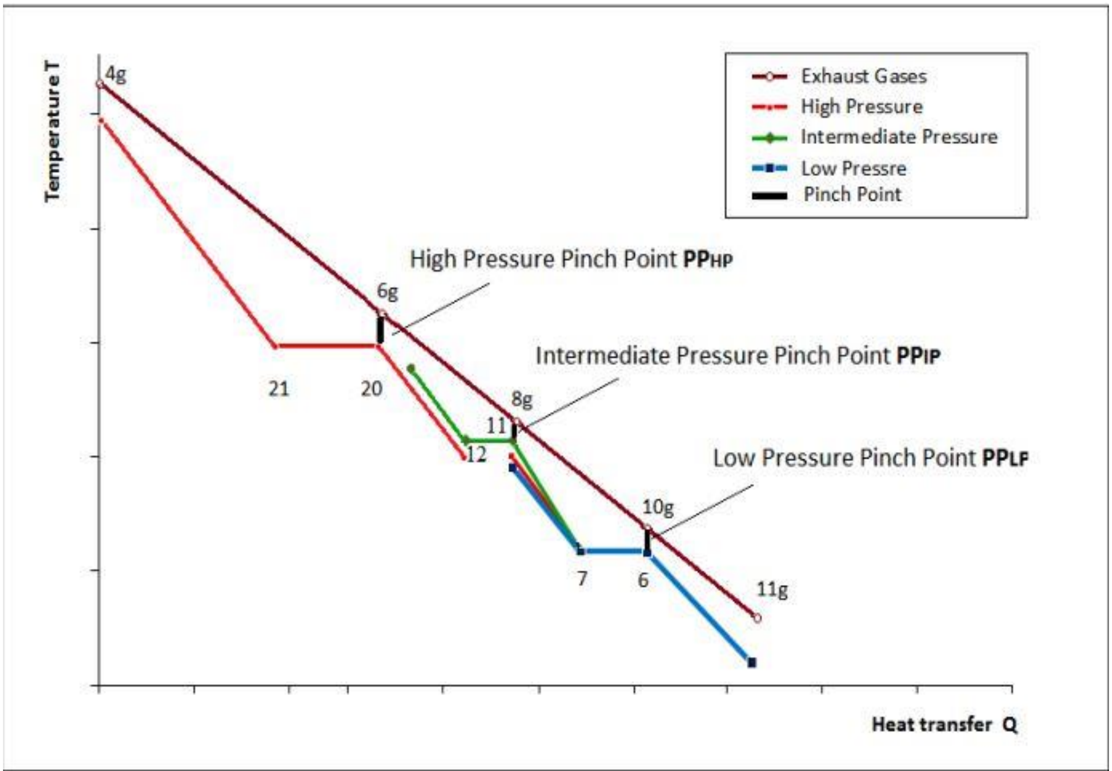

Fig.3-9 T-Q diagram of single pressure HRSG

3.1.4.3. Dew Point of Exhaust Gas and Water

For Sulphur-containing fuels, the Sulphur dew point must be taken into account because natural gas often have less sulphur , some authors set a lower limit for exhaust gas temperatures at (65 °C). The dew point for oil and solid fuels is normally in range of 100-165 °C, depends on Sulphur content.

For avoidance of consumption on metal surfaces in the HRSG and stack by corrosion, the stack temperature must be kept at the Sulfur dew point temperature throughout. The hugeness of the water dew point is that it frames a liquid stage that holds water as it contains corrosive segments.

Sulfuric acid (H_2SO_4) or nitric acid may be present in these components (HNO_3). The last is framed by NO or NO_2 produced by combustion. Corrosion can be caused by being near water.

Gas turbine exhaust gas has a water dew point of roughly 40 °C in most cases. It is unusual for power plants to have bulk exhaust temperatures anything close to that. In a high-efficiency plant, an exhaust gas stack temperature of around 80-100 °C is typical. Regardless of whether the bulk exhaust temperature is at or below the dew point near a cool surface, it is possible that it is below the dew point. With the purpose of avoiding water buildup on the tube surface, a usual business rule is that HRSG feed water must enter at a base temperature of 60 °C.

Corrosion can also be prevented via use of stainless steel or using a non-corroding substance to cover the tubes. Because the two options are extremely costlier in HRSG.

3.1.5. Steam Turbine:

The prime mover is the steam turbine, which converts steam heat energy into mechanical work. Gas turbines, jet engines, diesel engines along with gas engines are examples of notable prime movers. All steam-powered engines have made so such that to concentrate by allowing steam to flow through system. Steam flows through hub and fixed blades in the casing simply by rotating the progressive rings of the moving blades. When a steam turbine and a steam generator are coupled, it is considered one of the most innovative ways to distribute electric power on the world. Steam contains small blades that are approached by high-energy steams at greater temperatures in the high-pressure portions. Steam is subsequently transferred to reheater after passing through the high-pressure portion. After that, steam is transferred to intermediate pressure parts of the turbine. The blades in the intermediate pressure parts are bigger than those in the high pressure segments. The steam is transferred to the lower pressure parts of the turbine after passing through the intermediate pressure portions. The blades in this turbine are the largest because the majority of the energy is already taken from the steam. The steam exits the turbine at the base, where it is mixed with the water before being returned to the boiler drum and converted into steam once more. For most vapour power cycles, water (steam) is employed as working fluid in the steam turbine cycle based on the Rankine cycle. Water has a wide range of pressures and temperatures, as well as a huge heat capacity and is stable and safe. Coal, oil, incorporate gas, and atomic structures are employed as energy sources to make steam.

3.1.5.1. Steam Turbine Capacity:

The combined power output of the tiny turbines and connected generators ranges between 500 and 7500 kW. Large steam turbines can oscillate up to 500MW, and 200MW producing units are becoming more common. The steam turbine's ability to use steam is decided via the steam temperature and pressure at the entry, as well as the exhaust pressure, the number of draining stages, and other factors. In large gas turbines, steam is used at a rate of 3.5 to 5 kg per kWh.

3.1.5.2. Steam Turbine Performance:

Mechanical losses include things like generator bearings, oil pump losses along with bearing losses. The exhaust losses include the pressure drop from the exit of the last stage to the condenser, the loss of kinetic energy when steam exits through the last stage, and so on. The pressure of steam in the steam turbine may be efficiently managed when the security valve is installed in the boiler drum.

The Rankine cycle is the vapour power cycle employed in the steam power plant. This cycle include four process. In this cycle, the working fluid include steam. The working fluid is pumped from low to high pressure before being circulated through the boiler drum. The liquid is bubbled at a constant pressure in the boiler drum until it is turned into superheated high-pressure vapour, which is then allowed to run in the turbine, removing the work. The vapour from the turbine now enters the condenser, where the cooling liquid removes a significant quantity of heat energy.

Little changes are made to the Rankine Cycle to attain higher efficiency, such as superheating, reheating, and recovery. The perfect Rankine cycle, which has no external irreversibility, consists of four processes: constant-pressure heat addition in the boiler, isentropic expansion in the turbine, constant-pressure heat removal in the condenser, and isentropic compression in the pump. Each of the four processes in the Rankine cycle occurs in the steady flow devices and is a steady state process. The impacts of potential and kinetic energy, as well as the work done in the boiler and condenser, are ignored. The Rankine cycle's turbines and pumps are believed to be isentropic.

3.1.6. CONDENSER:

It is the device through which the steam exhaust from the steam turbines or the steam engines is condensed by the method of cooling water. The steam that is condensed is known as the condensate. The condensate formed with the air and the other non-condensable gases are continuously evacuated by the pumps. The main function of the condenser is maintaining a low back pressure on the exit of the rotor of the steam turbine. Because the condensate that is released from the condenser can be used as a feed water for the boiler drum multiple times, the condenser's primary requirement is to give the boiler with pure heat and hot feed water. The condenser can significantly upgrade the thermal effectiveness of a steam power plant. The fractional vacuum is given by condensed steam due to the significant drop in volume of low-pressure steam or water vapour.

At 100°C, 1kg of dry steam with a volume of 1.6735m3 and a pressure of 1.01325bar is contained in a steam tight vessel and condensed into water vapour. It was discovered that the fluid occupies 0.001044m3 i.e. the volume of the steam was found to be (1/1644) part of the space inside the vessel and the pressure will be reduced to 0.20bar. The back pressure in the steam turbine could be reduced from 1.01325 bar to 0.2 bar or less as a result of this. The Rankine cycle is the vapour power cycle employed in the steam power plant. A total four processes are included in this cycle. Steam is used as the working fluid in this cycle. Before circulation to boiler drum, the fluid is pumped from low to high pressure. The liquid is bubbled at a constant pressure in the boiler drum until it is turned into superheated high-pressure vapour, which is then allowed to run through the turbine, removing the work. The vapour from the turbine now enters the condenser, where the cooling liquid removes a significant quantity of heat energy.

3.1.6.1. Elements of a Steam Condensing Plant:

The elements that are necessary for a steam condensing plant are:

- A condenser wherein steam is being condensed.
- Instead of a surface condenser, a pump will circulate the cooling water.
- Condensing the exhaust steam requires injection or cooling water.
- The condenser's condensed steam, air, and uncondensed gases, as well as water vapours, are evacuated using the wet air pump.
- For chilling the flow of water, a cooling pond or a cooling tower is used when the water is limited.
- A hot well from which the condensed steam are discharged and from the hot well the boiler feed water is taken.

3.1.6.2.TYPES OF CONDENSER:

Condenser are classified into the two types in which the cooling water is cooled and the exhaust steam is condensed. They are as follows:

- **Jet Condenser:**

It is a condenser wherein exhaust steam comes in direct contact with the cooling water and hence the steam is condensed. For the fast condensation of the steam to happen, shower the cooling water with the fumed steams.

- **Surface Condenser:**

In this, the cooling water along with steam does not blend and here the cooling water is goes via the various tubes whereas the steam moves over the external surface of the tubes.

Out of the above two condenser it is found out that the surface condenser has the advantage that the densed steam which is also known as the condensate is allowed to come back to the evaporator through the feed water system rather than being thrown out of the waste. As compared to the surface condenser, the complexity of the mechanical assembly is less and less equipment’s are used in the steam condenser. When a shabby well spring of the evaporative feed water is attainable, we can use the jet condenser.

3.1.7. FEED PUMP:

Feed pumps are designed as the multistage radial flow pumps. They are also known as the boiler feed pumps. The main purpose of the feed pump is to feed the steam generators like boilers,HRSG with the quantity of feed water representing the quantity of the steam emitted. Now a days all the feed pumps are called as the centrifugal pumps. The feed pumps are designed based on the various parameters like input power,material,types of pump and drive that is mostly managed by the evolution that has taken place in the power station technology. The ratings of the boiler feed pump is found to be in the range of 30-50MW due to the trends in the larger fossil fueled power stations with the power output ratings greater than the 1000MW.Now a days, the feed pumps at the full loads are constructed with the four to six stage pressures of around 80bar with the total power output of 800 to 1100MW.The feed pumps used in the nuclear power stations are of the single type with the power output of 1600MW.With the rise of the unit outputs in the power stations, the mass flow rate of the boiler feed pumps varies rapidly.

CHAPTER 4

4. SYSTEM DESCRIPTION

The Brayton and Rankine Cycles, which came out on top among all the capable cycles devised for power generation, are both included in this Combined Cycle Power Plant. In this plant, the topping cycle operates on the Brayton Cycle's principle, and the bottoming cycle is a steam turbine cycle functions on the Rankine Cycle's premise. It was discovered that it had a thermal efficiency of more than 60%. In general, The Brayton cycle operates at a greater temperature than Rankine cycle.

Owing to the higher temperature on the end of combustion chamber, the gas turbine has more potency for better thermal effectiveness. However, has demerits that the gas turbine's exit temperature can exceed 600°C, lowering the turbine's thermal efficiency. As a result, the heat recovery steam generator is employed in order to remedy the situation. The heat released from exhaust gases is transferred to steam, which fills the boiler, in the heat exchanger.

As a result, the gas turbine provides sufficient heat to the steam. Furthermore, by consuming the extra fuel in oxygen-sufficient exhaust gases, heat recovery and the addition of steam is incorporated in the steam cycle for reheating process. The temperature of the gas turbine pipe is between 450 and 600 degrees Celsius. The amount of fuel energy not converted to power is measured by the quantity of energy present in pipe gas. It increases the steam and produce the steam turbine's maximum power output. The pipe gas temperature is decreased to roughly 80-200°C with the help of the heat recovery steam generator.

In the steam turbine cycle, steam is provided at temperatures at scale from 500 to 600°C and pressures ranging from 30 to 170 bar. The formation of steam will occur at various pressure levels. The application of a supercritical steam pressure greater than 220.65bar has been made possible by supercritical steam pressure, however this is mostly dependent on the pressure's dependency on the productivity and size of steam turbine. The topping cycle functions at higher temperatures, whereas the bottoming cycle functions at low temperatures and concentrates on the reutilization of energy fluxes.

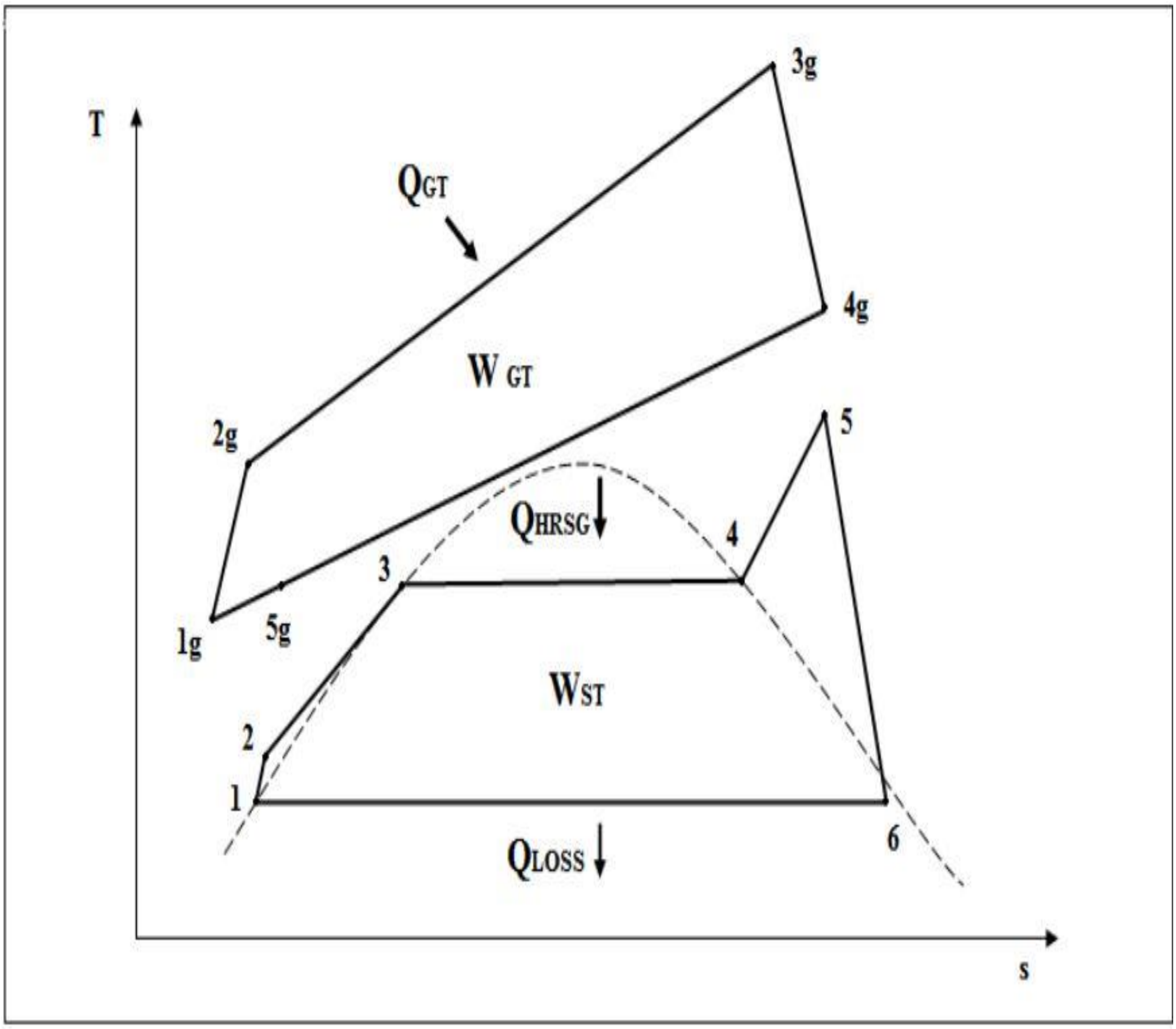

Fig.4-1 T-s diagram of the Combined Cycle Power System

<u>CHAPTER 5</u>

5. METHODOLOGY:

CASE1:

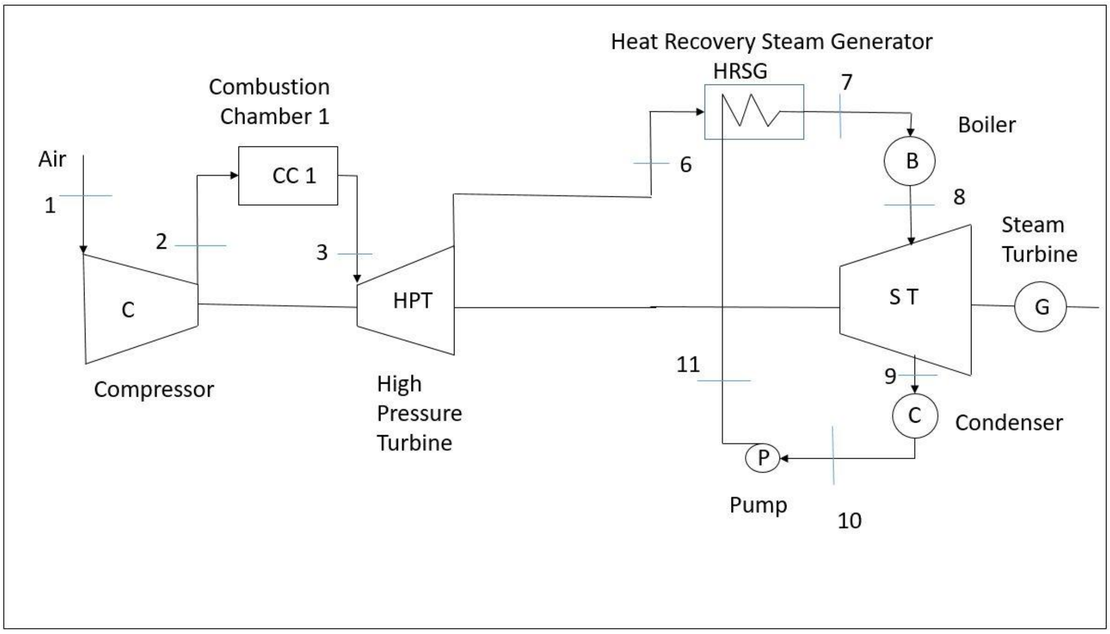

Fig.5-1 Simple Combined Cycle Power Plant

CASE2:

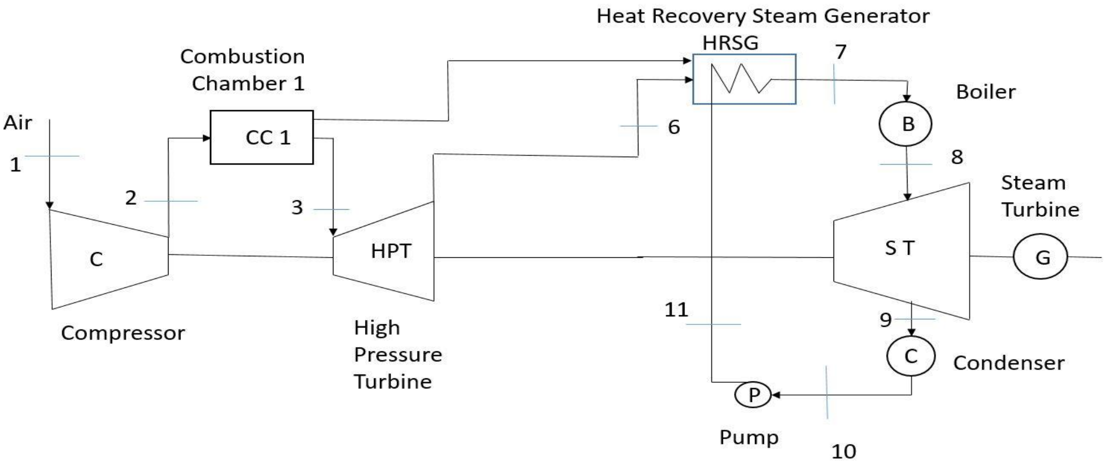

Fig.5-2 Combined Cycle Power Plant with Bypassing

CASE3:

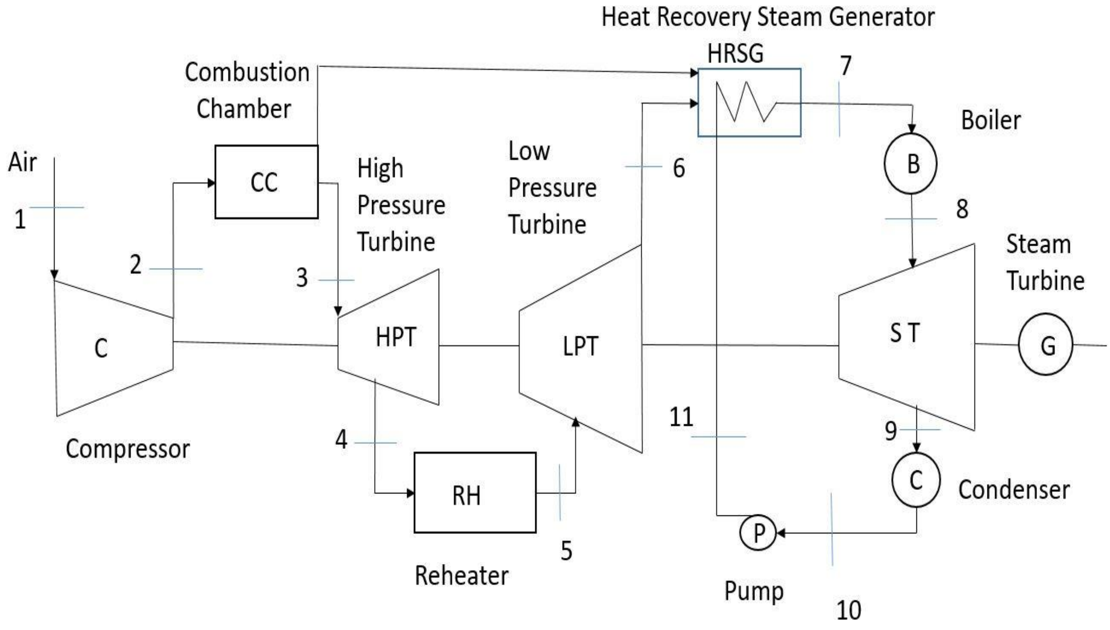

Fig.5-3 Combined Cycle Power Plant with Bypassing and Reheating

TABLE:1 Representing all the points in the schematic diagram

Sl.No.	Specifications
1	Inlet Air Entering the Compressor
2	Outlet Air From Compressor
3	Combustion Gases Exiting the Combustion Chamber
4	Outlet Hot Gases Exiting from the High Pressure Turbine
5	Outlet Hot Gases Entering into the Low Pressure Turbine
6	Outlet Hot Gases Exiting from the Low Pressure Turbine
7	Outlet Gases Exiting HRSG
8	Super-heated steam entering from the gas turbine
9	Outlet Steam from the steam turbine
10	Saturated liquid entering the Pump
11	Supplied water entering the HRSG

5.1. FORMULATION OF THE EQUATIONS:

5.1.1. ENERGY EQUATIONS:

5.1.1.1. GAS TURBINE (BRAYTON CYCLE):

For the gas turbines, the parameters along with assumptions, selected for thermodynamic analysis of the Gas Turbines (Brayton Cycle) are tabulated in the table 2 as shown below.

Table 2:Gas Turbine Parameters

Gas turbine (Brayton cycle) circuit PARAMETERS	Values
Ambient air pressure [bar]	1.01325
Ambient air temperature [K]	298
Gas Turbine isentropic efficiency [%]	88
Compressor isentropic efficiency [%]	92
Lower calorific value of fuel[kJ/kg]	47000
Heat capacity ratio for air (Υ_{air})	1.40
Heat capacity ratio for gas (Υ_{gas})	1.33
Pressure Ratio	12-20
Pressure loss in the combustion chamber	0.96 of P2
Efficiency in the combustion chamber [%]	99.5
Compressor mechanical efficiency [%]	98
Turbine mechanical efficiency [%]	98
Effectiveness of HRSG	0.95

5.1.1.1. COMPRESSOR:

Temperature at first compressor exit (theoretical)

$$T_2 = T_1 * rp^{\frac{\Upsilon a - 1}{\Upsilon a}}$$

The theoretical work done by the compressor

$$\dot{W}_{compressor} = \dot{m}_{air} * C_{pa} * (T_2 - T_1)$$

Actual temperature at wayout of the compressor

$$T_{2a} = T_1 + \left(\frac{(T_2 - T_1)}{\eta_{isentropic\ compressor}}\right)$$

The actual work done by the compressor

$$\dot{W}_{compressor_{actual}} = \dot{m}_{air} * C_{pa} * (T_{2a} - T_1) * \eta_{mech}$$

5.1.1.2. COMBUSTION CHAMBER:

The following relationship is found using the energy balance for combustion chamber:

$$\dot{m}_a * C_{pa} * T_{2s} + \dot{m}_f * LCV * \boldsymbol{\eta}_{comb} = (\dot{m}_a + \dot{m}_f) * C_{pg} * T_3$$

$$AFR = \frac{\dot{m}_a}{\dot{m}_f}$$

Temperature outside the combustion chamber

$$T_3 = \frac{(AFR * C_{pa} * T_{2a} + LCV * \boldsymbol{\eta}_{comb})}{(AFR + 1) * C_{pg}}$$

5.1.1.3. GAS TURBINE:

Without Reheating:

The gas turbine's parameters are presented in Table 2. The following relationship is found using the Gas Turbine Energy Balance.

Theoretical turbine exit temperature

$$T_4 = \frac{T_3}{r_p^{\frac{\gamma g - 1}{\gamma g}}}$$

Theoretical work obtained from gas turbine

$$\dot{W}_{gt} = (\dot{m}_{air} + \dot{m}_f) * C_{pg} * (T_3 - T_4)$$

Temperature at turbine outlet

$$T_{4a} = T_3 - \boldsymbol{\eta}_{gt\ isen} * (T_3 - T_4)$$

Actual task done with a gas turbine

$$\dot{W}_{gt}=\frac{(\dot{m}_{air}+\dot{m}_{f})*Cpg*(T3-T4a)}{\eta_{mech}}$$

With Reheating:

Intermediate Pressure

Pressure Ratio in the High Pressure Turbine

$$r_{p_3}=(r_{p_2})^{n}$$

Temperature at high pressure turbine outlet

$$T_{3a}=\frac{T_3}{(r_{p_3})^{\frac{\Upsilon_g-1}{\Upsilon_g}}}$$

Pressure Ratio in the Low Pressure Turbine

$$r_{p_4}=(r_{p_2})^{1-n}$$

$$r_{p_2}=r_{p_3}*r_{p_4}$$

where n is the exponent varying from 0.2 to 0.8 with a gap of 0.2 for determining the intermediate pressure

Temperature at low pressure turbine outlet

$$T_{3d}=\frac{T_3}{(r_{p_4})^{\frac{\Upsilon_g-1}{\Upsilon_g}}}$$

The following relationship is found using the energy balance for combustion chamber:

$$(\dot{m}_{a}+\dot{m}_{f1})*C_{pa}*T_{3a}+\dot{m}_{f2}*LCV*\boldsymbol{\eta}_{comb}=(\dot{m}_{a}+\dot{m}_{f1}+\dot{m}_{f2})*C_{pg}*T_3$$

Work done by the High Pressure Turbine

$$W_{Hpt}=(\dot{m}_{air}+\dot{m}_{f1})*C_{pg}*(T_3-T_{3a})$$

Work done by the Low Pressure Turbine

$$W_{Lpt}=(\dot{m}_{air}+\dot{m}_{f1}+\dot{m}_{f2})*C_{pg}*(T_3-T_{3d})$$

Total work obtained after reheating

$$W_{total}=W_{Hpt}+W_{Lpt}$$

Total heat supplied into the combustion chamber $=(\dot{m}_{f1}+\dot{m}_{f2})*LCV*\boldsymbol{\eta}_{comb}$

$$\boldsymbol{\eta}_{reheat\ overall}=\frac{W_{Hpt}+W_{Lpt}}{(\dot{m}_{f1}+\dot{m}_{f2})*LCV*\boldsymbol{\eta}_{comb}}$$

5.1.1.4. Heat Recovery Steam Generator (HRSG):

The HRSG comes after the gas turbine. For production of steam, gas turbine hot gases are moved via steam generator heaters.. In this study, the HRSG is made up of three sectional heat exchangers.

- Section 1: super heater
- Section 2: evaporator
- Section 3: economizer

Table 3 lists the HRSG's assumptions and parameters.

Parameters	Values
The Pinch Point Temperature Difference (℃)	13
Pressure in HRSG(bar)	10
Live Steam Temperature at the Inlet of the Steam Turbine(℃)	535
Minimum Stack Temperature(℃)	150
Feed Water Temperature(℃)	55

The HRSG unit's investigation is based on the pinch point. The distinctiveness between the water saturation along with gas temperature before passage to the economizer is known as the pinch point. PP denotes the pitch point. As a result, the HRSG exit temperature rises. The standard "IAPWS" was used to derive the water-steam characteristics. HRSG's heat exchange method is comparable to that of a heat exchanger. The flue gases are drawn into the HRSG.At T4s, it enters HRSG and travels through several sectors before exiting at T5. In the economizer portion, the feed water is heated from T9s to saturation temperature; at this temperature, it enters the evaporator to generate steam and exits at the same saturation temperature. In the super heater portion, the steam is brought up to temperature.

A system of equations is obtained via application of thermodynamics 1st law to the heat recovery steam generator parts (economizer, evaporator, and super heater). Solving these equations yields the steam mass flow rate.

$$(\dot{m}_a + \dot{m}_f) * C_{pg} * (T_{4a} - T_5) = \dot{Q}_{econ} + \dot{Q}_{evap} + \dot{Q}_{sup}$$

$$\dot{Q}_{econ} = \dot{m}_w * C_{pw} * (T_{sat} - T_9)$$

$$\dot{Q}_{evap} = \dot{m}_w * LH$$

$$\dot{Q}_{sup} = \dot{m}_w * C_{psteam} * (\mathrm{T}_6 - \mathrm{T}_{sat})$$

From the above equations we get the mass flow rate of the steam as

$$\dot{m}_w = \left(\frac{(\dot{m}_a + \dot{m}_f) * C_{pg} * (\mathrm{T4a} - \mathrm{T5})}{(C_{pw} * (\mathrm{T}_{sat} - \mathrm{T9}) + LH + C_{psteam} * (\mathrm{T}_6 - \mathrm{T}_{sat}))}\right)$$

5.1.1.5. Steam Turbine:

Table 4 summarizes the assumptions and parameters used in the steam turbine's thermodynamic study.

Parameters	Value
High-Pressure steam turbine inlet (boiler pressure) [bar]	60
High-Temperature steam turbine inlet (boiler pressure)[℃]	540
Low-pressure steam turbine outlet (condenser pressure) [bar]	0.07
Minimum dryness fraction of steam at steam turbine outlet	0.88
The isentropic efficiency of steam turbine	92

The following relationship is established via application of the energy balance to steam turbine levels:

Steam turbine theoretical research

- $\dot{\mathrm{W}}_{st} = \dot{m}_w * (h_5 - h_6)$

Actual work done using a steam turbine

$$\dot{\mathrm{W}}_{st} = \dot{m}_w * (h_5 - h_6) * \boldsymbol{\eta}_{st\ isen}$$

5.1.1.6. Condenser:

Equations are secured via thermodynamics 1st law. Solving these equations yields the cooling water mass flow rate.

$$\dot{m}_{cw} * C_{p_{cw}} * \Delta\mathrm{T}_{cw} = \dot{m}_w * (h_7 - h_8)$$

$$\dot{m}_{cw} = \frac{\dot{m}_w * (\mathrm{h7} - \mathrm{h8})}{C_{p_{cw}} * (\Delta\mathrm{T}_{cw})}$$

Theoretical work absorbed by pump

$$\dot{\mathrm{W}}_p = \dot{m}_w * (h_9 - h_8)$$

5.1.1.7. Performance Assessment Parameters:

The following are the results of a power plant, including Brayton, Rankine, and cycle's production and thermal efficiency:

Net work in Brayton Cycle

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

$$W_{Net} = (\dot{m}_{air} + \dot{m}_{f})*C_{pg}*(T_3-T_{4a}) - (\dot{m}_{a} + \dot{m}_{f}) * C_{pa}*(T_{2a}-T_1)$$

Effectiveness of the Brayton Cycle

$$\boldsymbol{\eta}_{brayton} = \frac{\dot{W}_{net_{brayton}}}{\dot{m}_{f} * LCV * \boldsymbol{\eta}_{comb}}$$

Network in Rankine Cycle

$$\dot{W}_{net_{Rankine}} = \dot{W}_{st_{actual}} - \dot{W}_{p_{actual}}$$

Efficiency of the Rankine Cycle

$$\boldsymbol{\eta}_{rankine} = \frac{\dot{W}_{net_{Rankine}}}{\dot{m}_{w}*(h_6-h_9)}$$

Power Output of the combined cycle

$$\dot{W}_{net_{combined}} = \dot{W}_{net_{brayton}} + \dot{W}_{net_{Rankine}}$$

Efficiency of the combined cycle

$$\boldsymbol{\eta}_{combined} = \left(\frac{\dot{W}_{net_{combined}}}{\dot{m}_{f} * LCV * \boldsymbol{\eta}_{comb} + \dot{m}_{w} * (h_6 - h_9)}\right)$$

<u>CHAPTER 6</u>

6.1. RESULTS AND DISCUSSION:

6.1.1. WITH BYPASSING AND WITHOUT REHEATING:

6.1.1.1 PRESSURE RATIO VS OVERALL EFFICIENCY OF THE COMBINED CYCLE POWER PLANT:

The overall efficiency of the gas turbine has includes both the efficiency of the gas turbine and the Rankine efficiency and with the increase in the pressure ratio the overall efficiency of the CCPP plant increases and reaches to the maxima with the increase in the bypass factor from 10% to 30%.For the bypass factor of 30%,the maximum operating efficiency at the pressure ratio of 16.

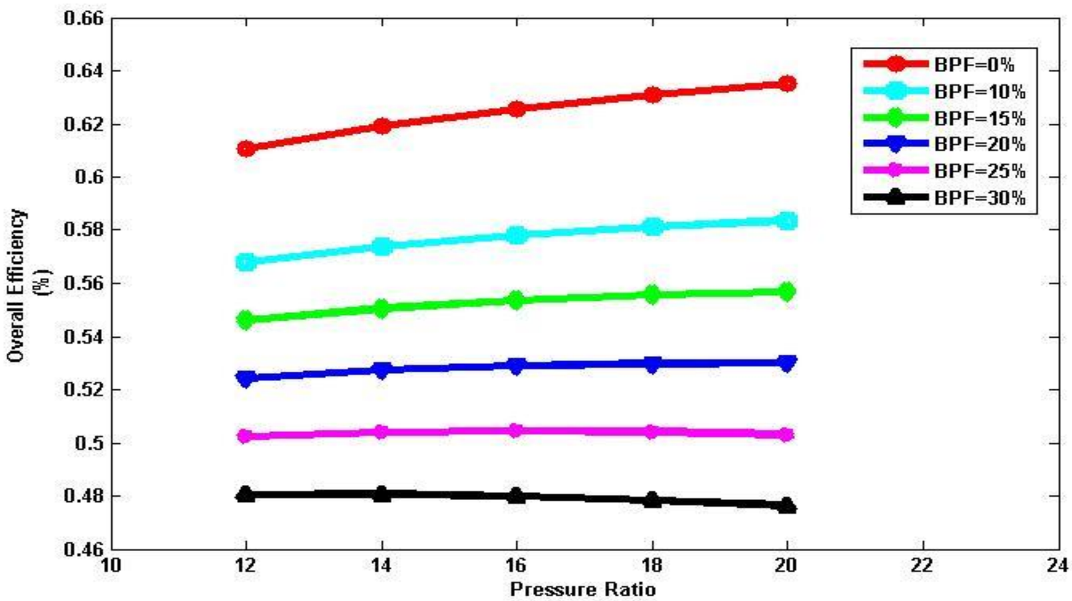

Fig. 6-1 Variation of Pressure Ratio with the Overall Efficiency of the CCPP

6.1.1.2. PRESSURE RATIO VS THE MASS OF THE STEAM OBTAINED IN THE CCPP:

Mass flow rate of the steam is the function of the bypass factor. With the increase in the bypass factor the mass flow rate of the steam increases but with the increase in the pressure ratio the mass flow rate of the steam decreases.

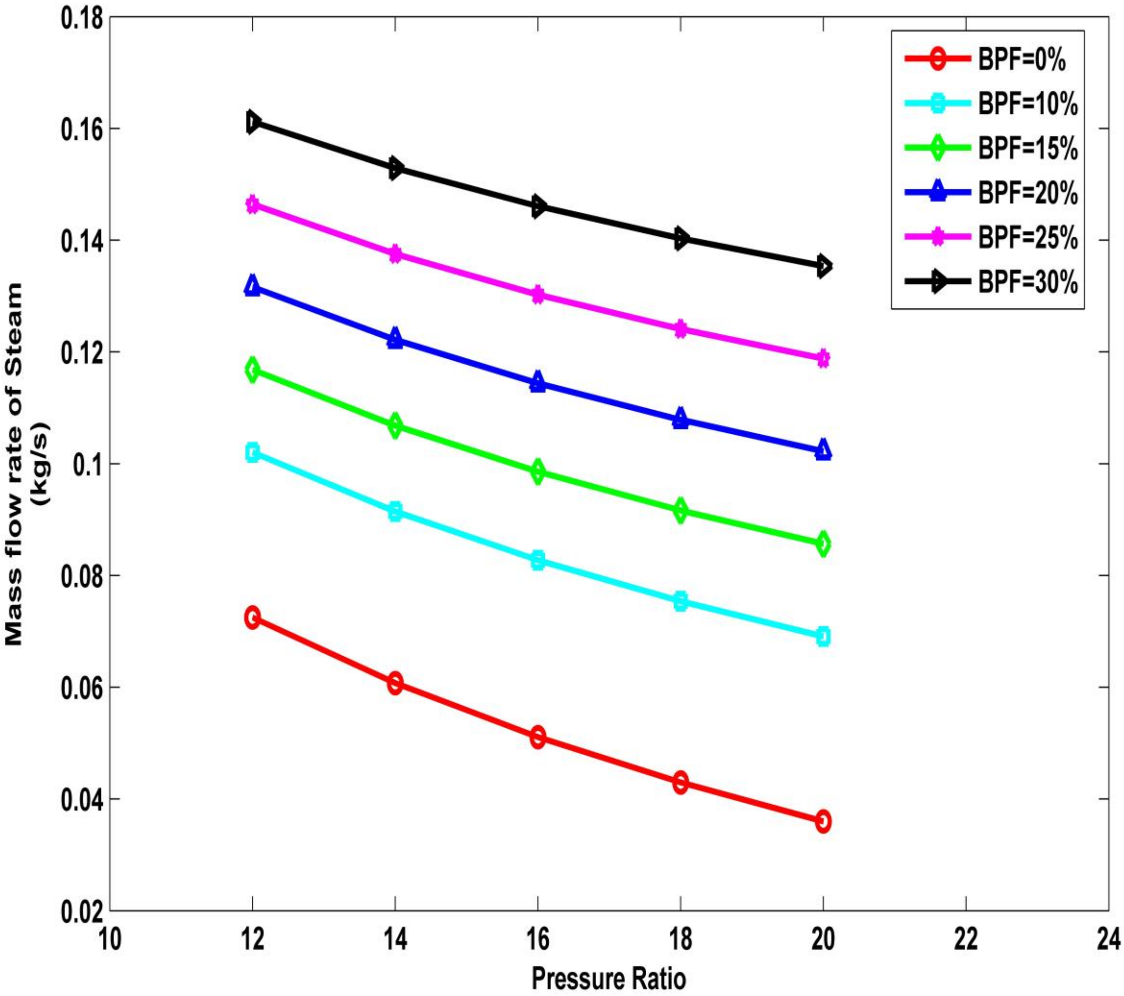

Fig. 6-2 Variation of Pressure Ratio with the Mass of the Steam Produced in the CCPP

6.1.1.3. PRESSURE RATIO VS OVERALL WORK DONE IN THE CCPP:

Overall work done is a function of pressure ratio. With the increase in the pressure ratio, the overall work done in the CCPP decreases and it is observed that with the increase in the bypass factor the work done by the CCPP decreases.

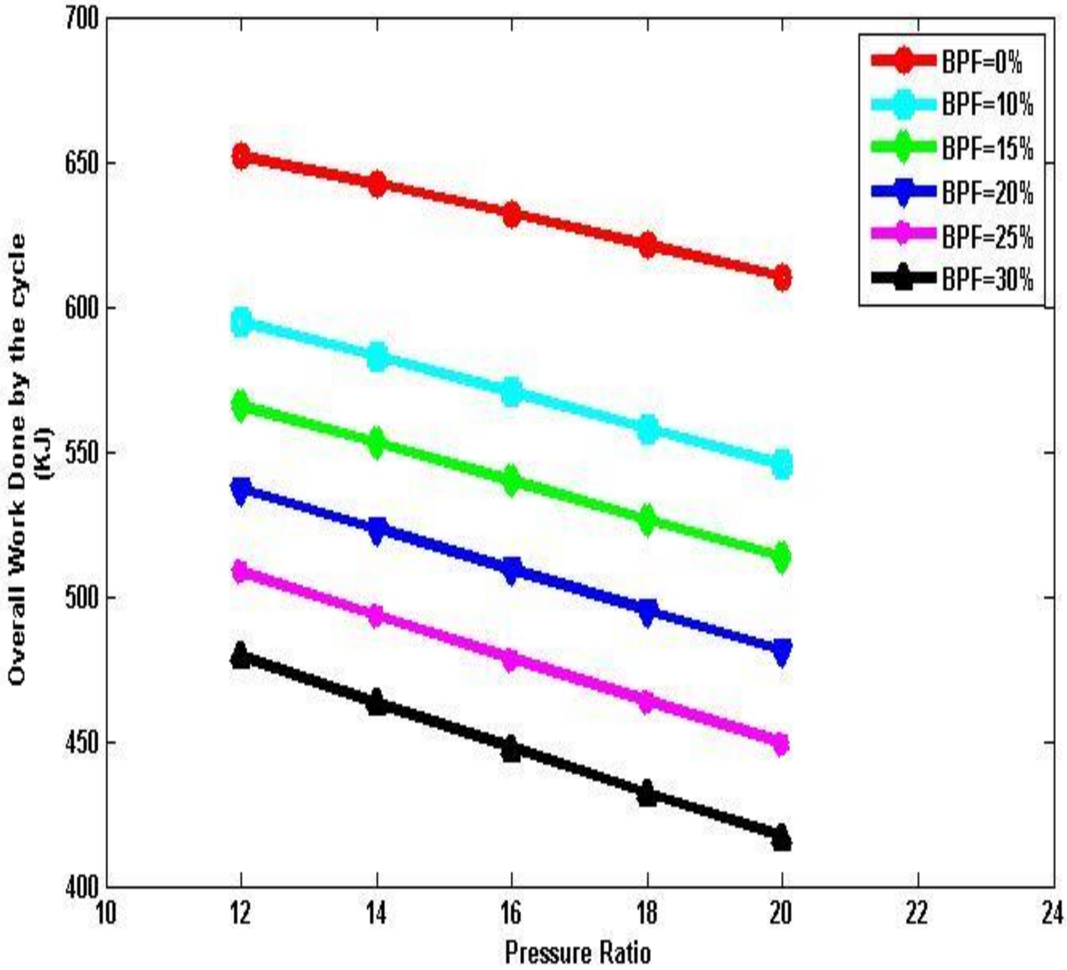

Fig. 6-3 Variation of Pressure Ratio with the Overall Work Done in the CCPP

6.1.1.4. PRESSURE RATIO VS GAS TURBINE EFFICIENCY IN THE CCPP:

With the increase in the pressure ratio, it is found that with the increase in the pressure ratio the gas turbine efficiency increases and its slope i.e. the rate of increase of efficiency decreases and tries to reach the maxima. Gas turbine efficiency is the function of the pressure ratio.

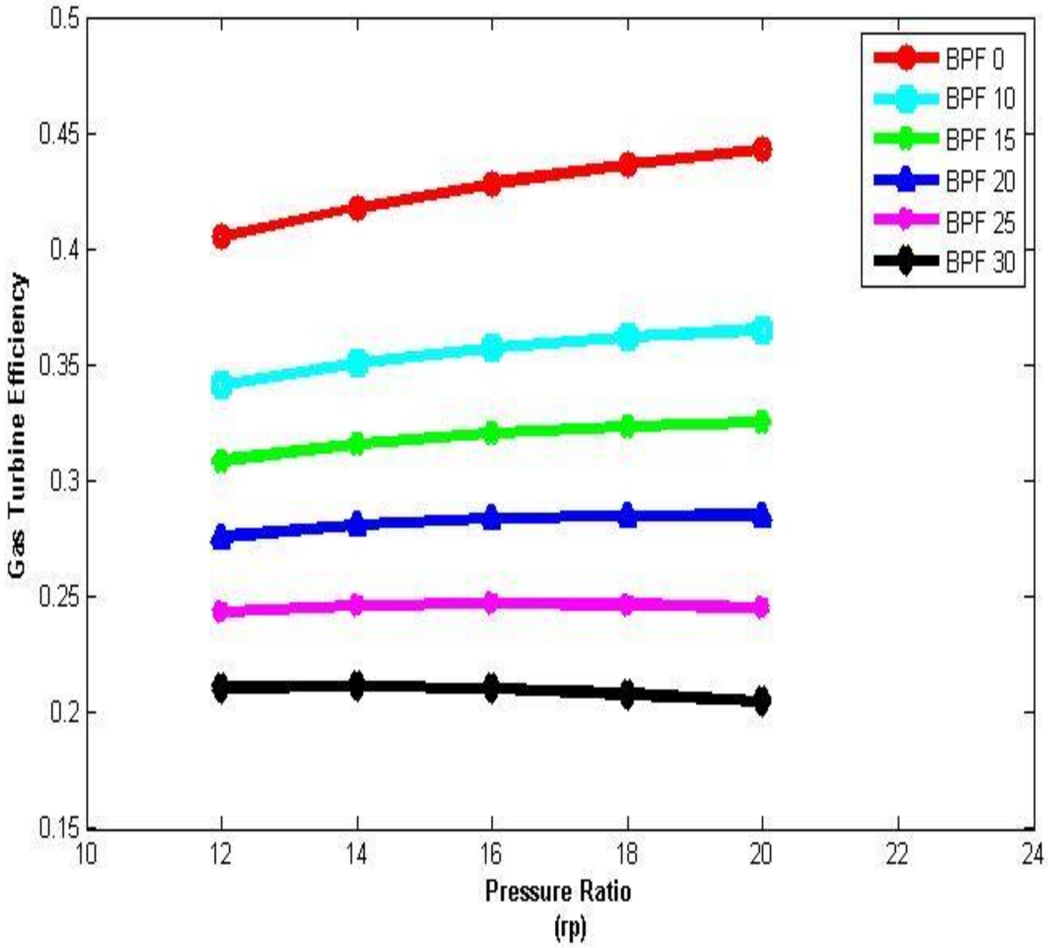

Fig. 6-4 Variation of Pressure Ratio with the Gas Turbine Efficiency in the CCPP

6.1.2. WITH BYPASSING AND WITH REHEATING:

For BPF=0.10

6.1.2.1 PRESSURE RATIO VS GAS TURBINE WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the value of n from 0.2 to 0.8, it is concluded that for n=0.20 and n=0.80,the gas turbine work obtained during the reheating is same and similar is the case for n=0.40 and n=0.60.It is due to the methodology of calculating the intermediate pressure.(shown in methodology chapter).

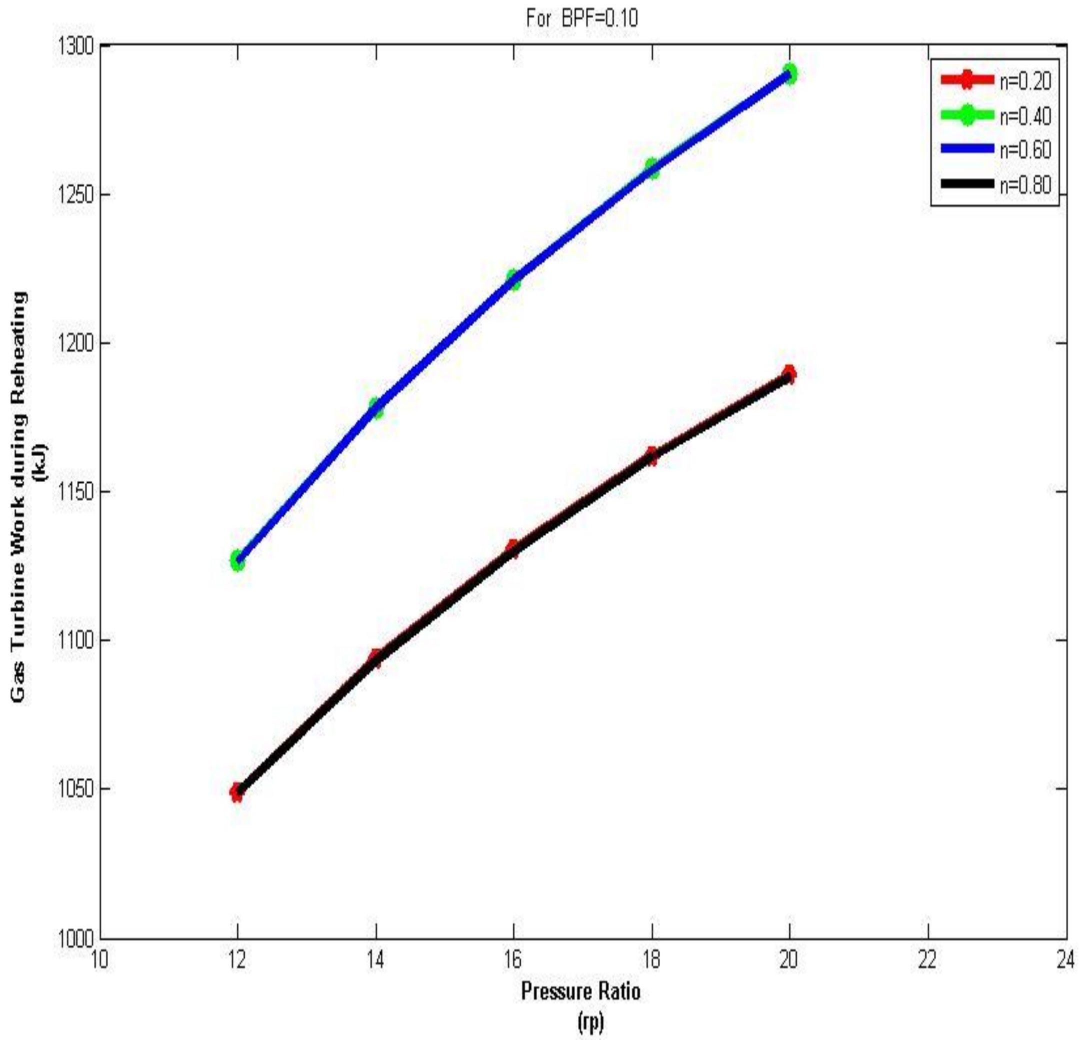

Fig. 6-5 Variation of Pressure Ratio with the Gas Turbine Work during Reheating in the CCPP

6.1.2.2 PRESSURE RATIO VS NET WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

Here the graph is plotted by taking the pressure ratio in the abscissa and the network during reheating of plant in the ordinate for the bypass factor ranging from 10% in the middle of combustion chamber plus high-pressure turbine. The pressure ratio is varies with a common difference of 2 starting from 12 to 20 and then the graph is plotted as shown below.

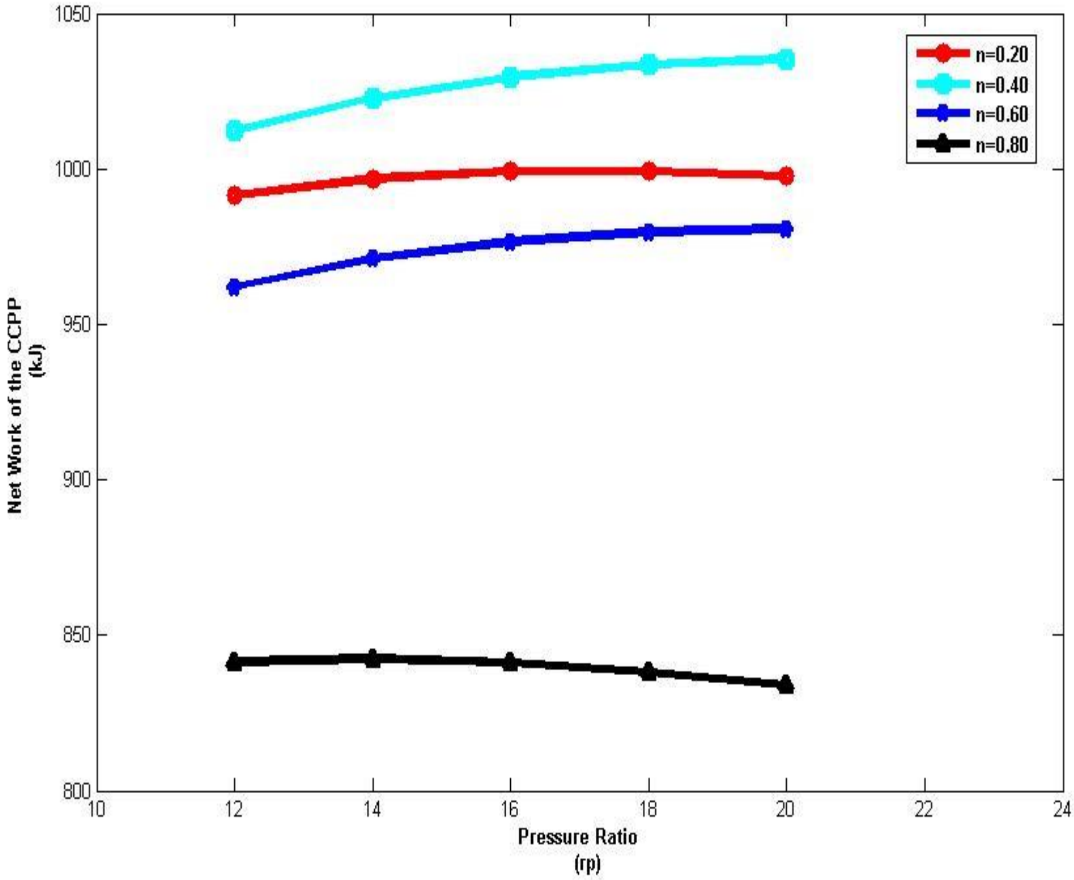

Fig. 6-6 Variation of Pressure Ratio with the Net Work during Reheating in the CCPP

6.1.2.3.PRESSURE RATIO VS REHEATING EFFICIENCY OF GAS TURBINE IN THE COMBINED CYCLE POWER PLANT:

It is found out that with the increase in the pressure ratio, the reheating efficiency of the gas turbine increases and also with the increase in the value of n the reheating efficiency increases. For n=0.80, reheating efficiency is maximum and for n=0.20, the reheating efficiency is minimum.

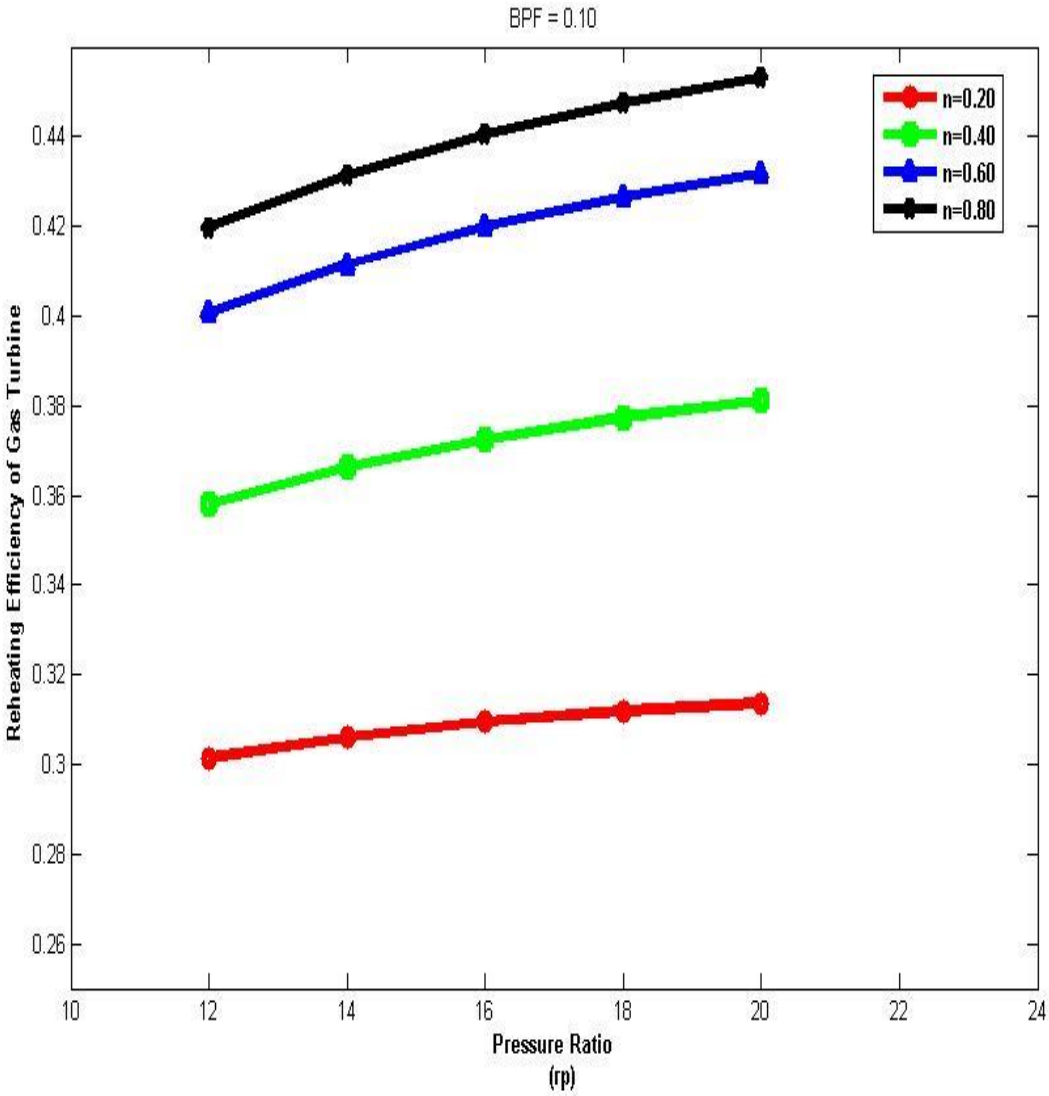

Fig. 6-7 Variation of Pressure Ratio with the Gas Turbine Efficiency with Reheating in the CCPP

6.1.2.4.PRESSURE RATIO VS OVERALL EFFICIENCY OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the n value the overall efficiency increases with the increase in the pressure ratio. Hence for n=0.80, the overall efficiency of the combined cycle becomes maximum.

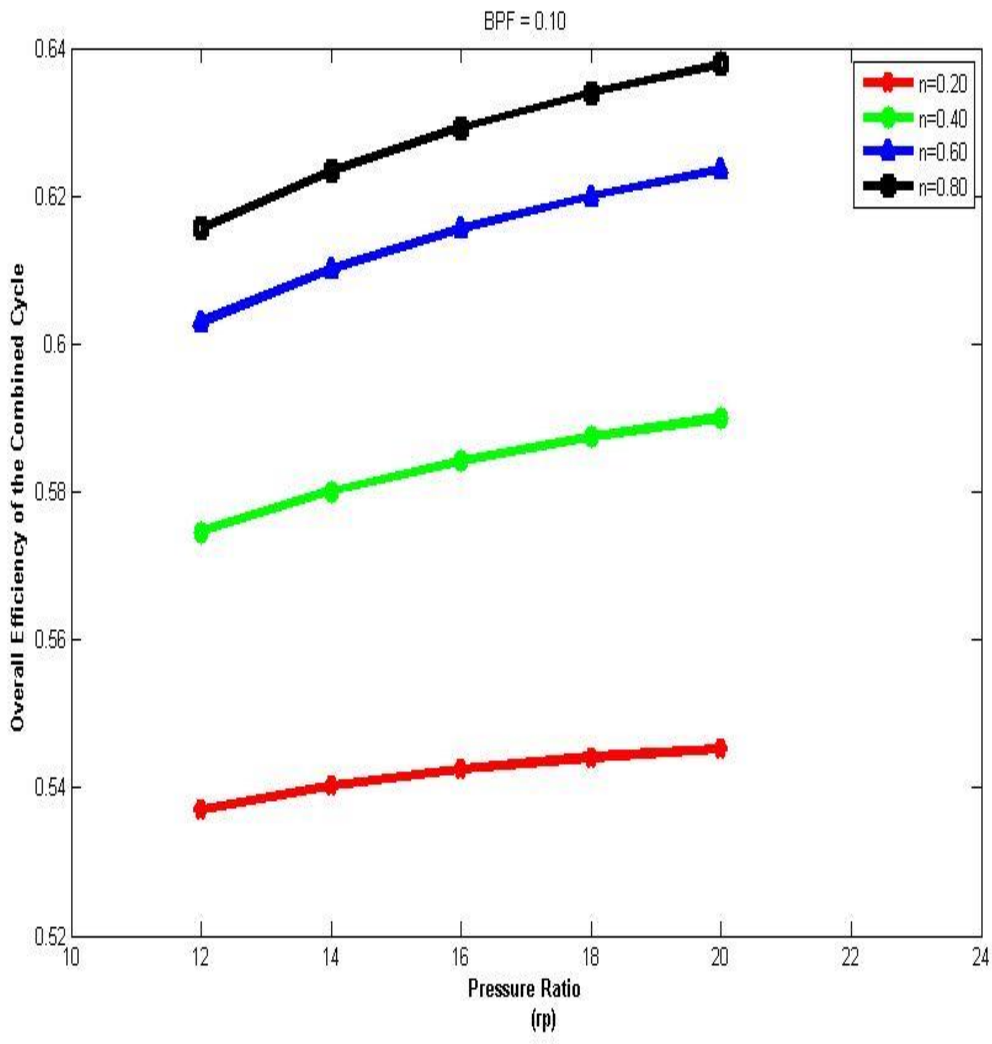

Fig. 6-8 Variation of Pressure Ratio with the Overall Efficiency of the CCPP

For BPF=0.20

6.1.2.5 PRESSURE RATIO VS GAS TURBINE WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the value of n from 0.2 to 0.8, it is concluded that for n=0.20 and n=0.80, the gas turbine work obtained during the reheating is same and similar is the case for n=0.40 and n=0.60.It is due to the methodology of calculating the intermediate pressure. (Shown in methodology chapter).

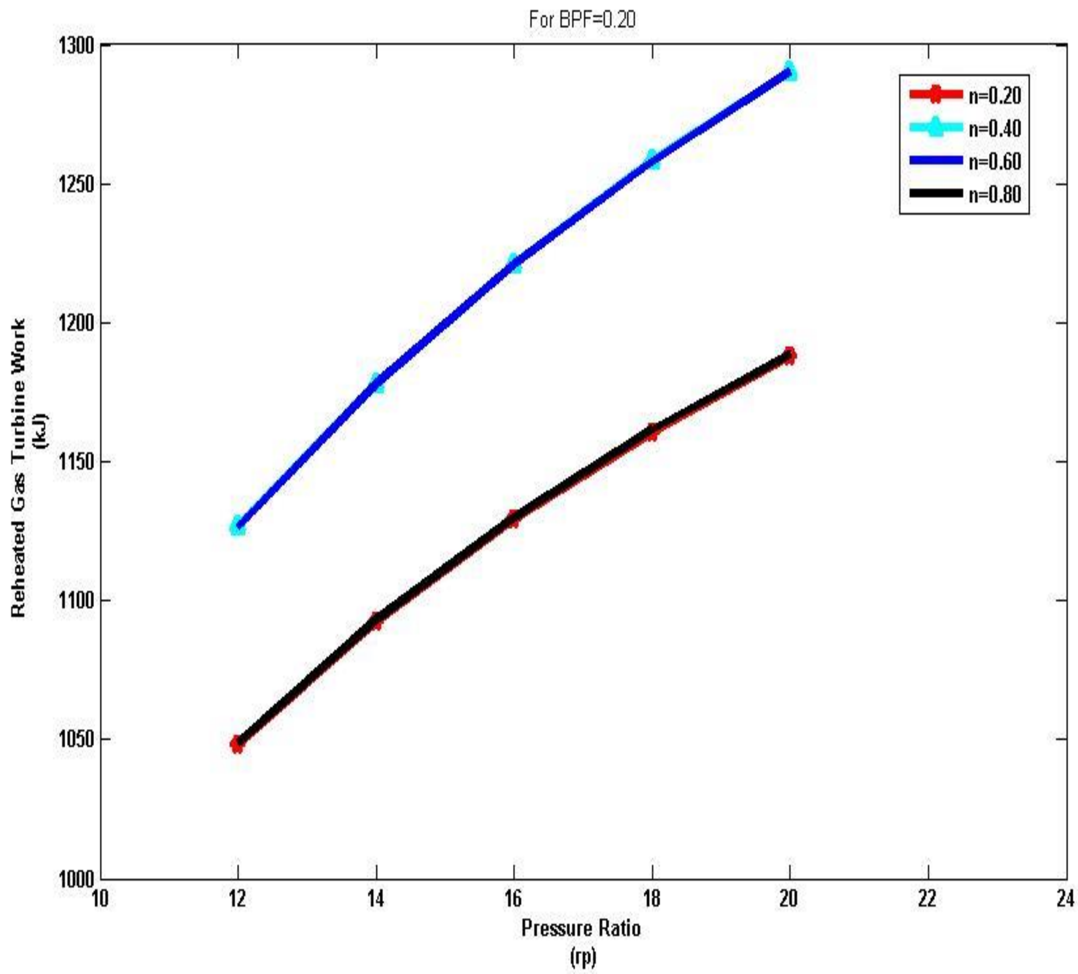

Fig. 6-9 Variation of Pressure Ratio with the Gas Turbine Work during Reheating in the CCPP

6.1.2.6.PRESSURE RATIO VS NET WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

Here at n=0.20 and n=0.60 intersects in between pressure ratio 18 to 20 and at n=0.4 the net work of the combined cycle. becomes maximum. As the n values increases from n=0.20 to n=0.80, the network of the combined cycle graph becomes steeper.

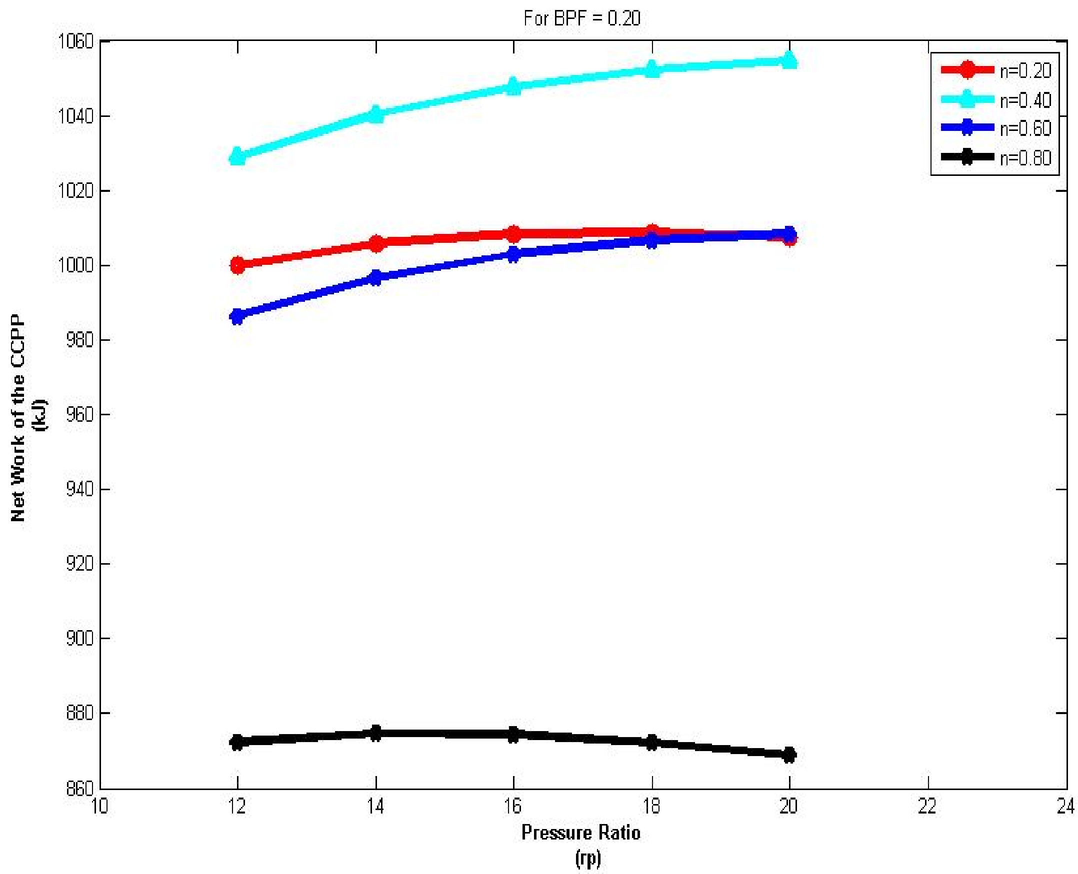

Fig. 6-10 Variation of Pressure Ratio with the Net Work during Reheating in the CCPP

6.1.2.7.PRESSURE RATIO VS REHEATING EFFICIENCY OF GAS TURBINE IN THE COMBINED CYCLE POWER PLANT:

It is found out that with the increase in the pressure ratio, the reheating efficiency of the gas turbine increases and also with the increase in the value of n the reheating efficiency increases. For n=0.80, reheating efficiency is maximum and for n=0.20,the reheating efficiency is minimum.

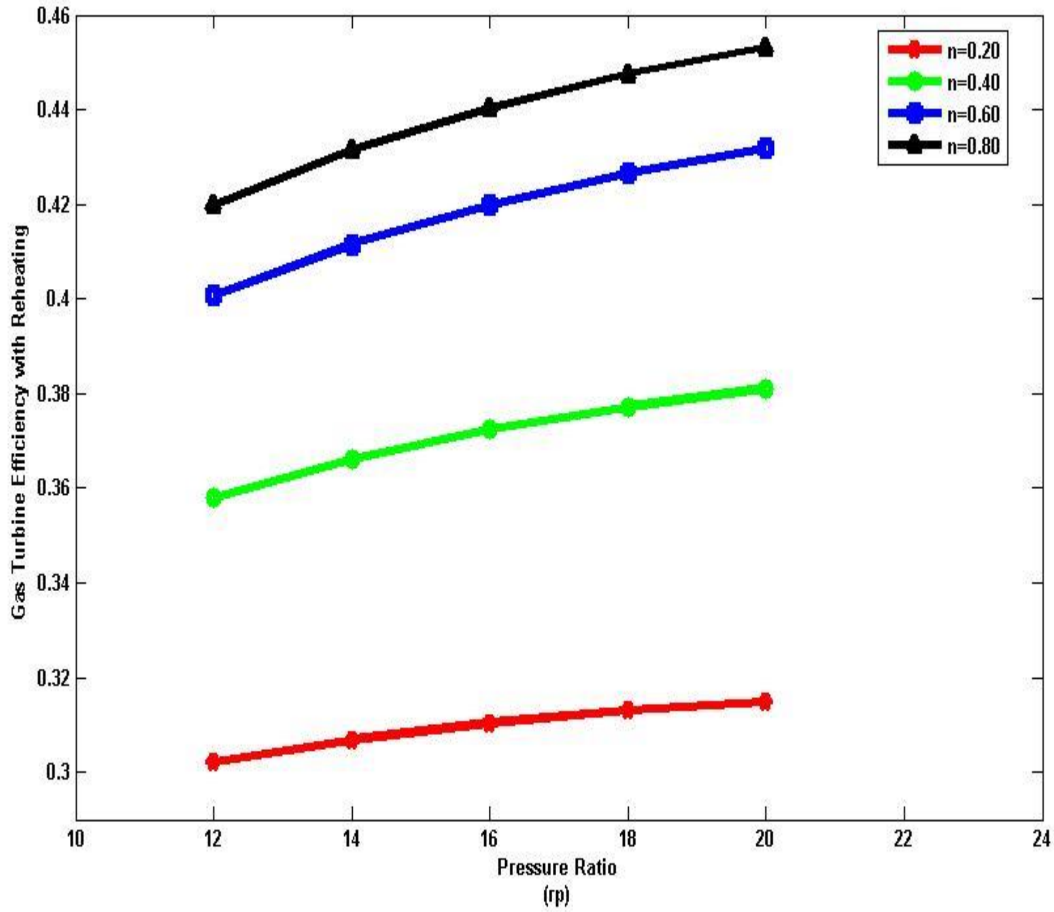

Fig. 6-11 Variation of Pressure Ratio with the Gas Turbine Efficiency with Reheating in the CCPP

6.1.2.8 PRESSURE RATIO VS OVERALL EFFICIENCY OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the n value the overall efficiency increases with the increase in the pressure ratio. Hence for n=0.80, the overall efficiency of the combined cycle becomes maximum.

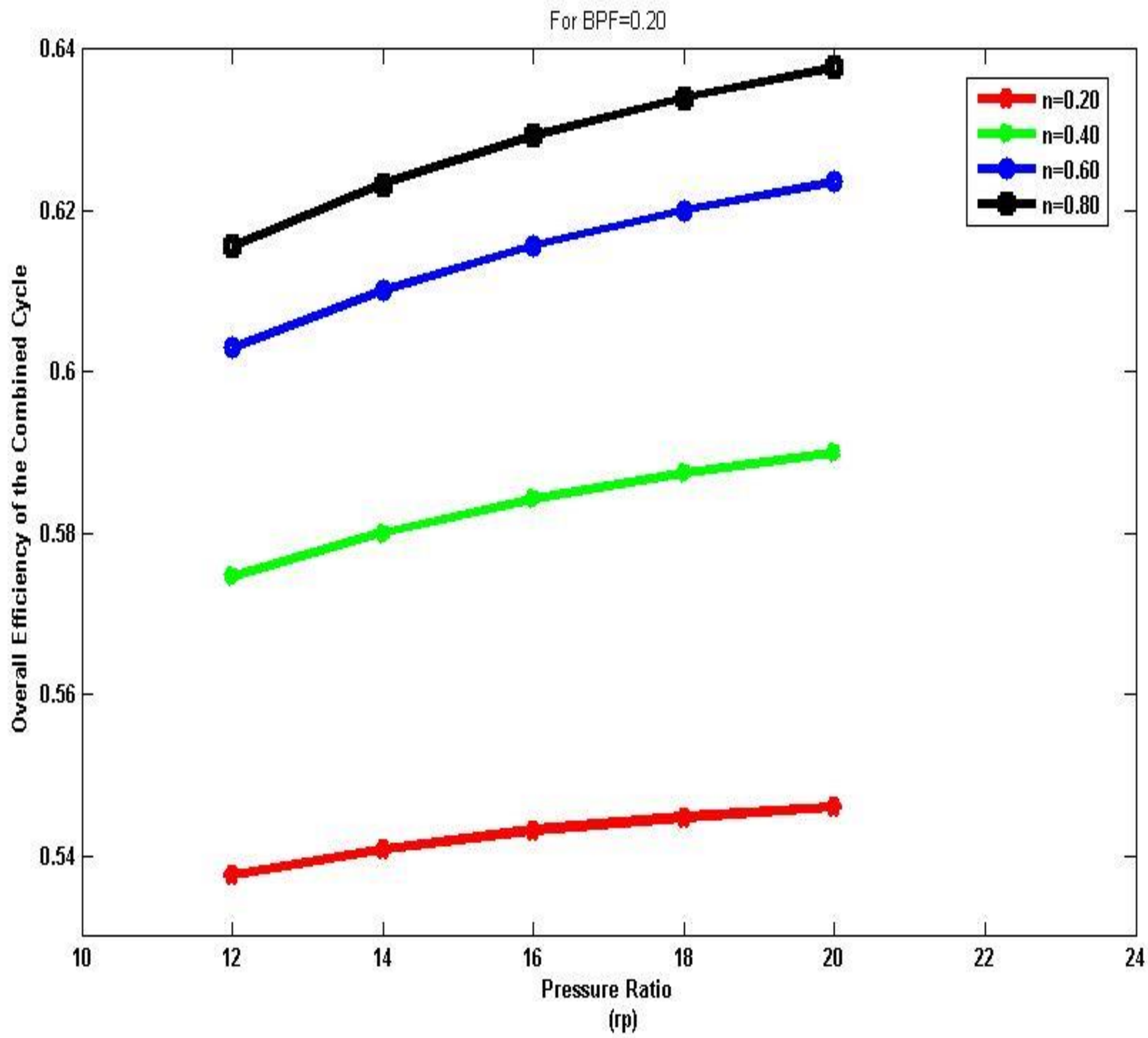

Fig. 6-12 Variation of Pressure Ratio with the Overall Efficiency of the CCPP

For BPF=0.30

6.1.2.9 PRESSURE RATIO VS GAS TURBINE WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the value of n from 0.2 to 0.8, it is concluded that for n=0.20 and n=0.80,the gas turbine work obtained during the reheating is same and similar is the case for n=0.40 and n=0.60.It is due to the methodology of calculating the intermediate pressure.(shown in methodology chapter).

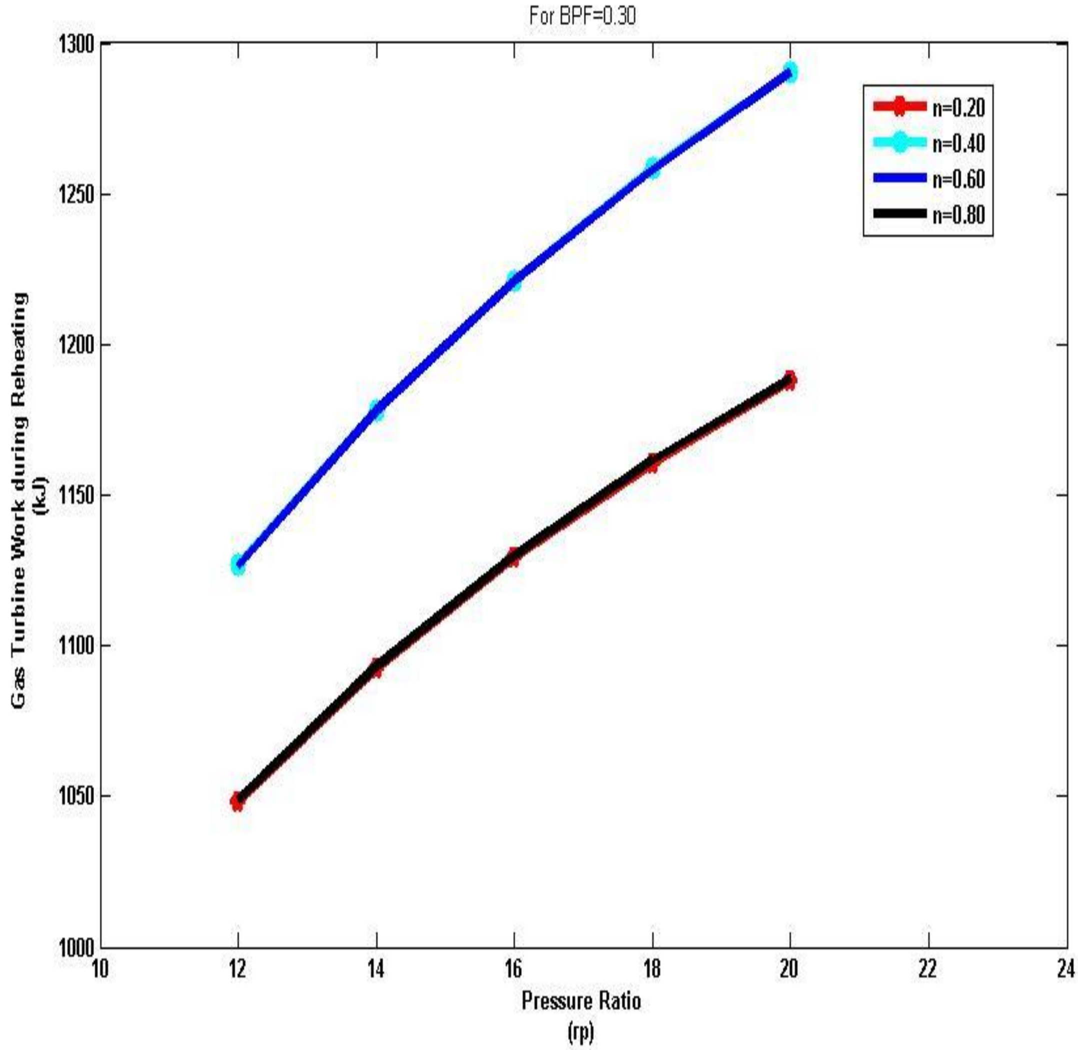

Fig. 6-13 Variation of Pressure Ratio with the Gas Turbine Work during Reheating in the CCPP

6.1.2.10.PRESSURE RATIO VS NET WORK WITH REHEATING OF THE COMBINED CYCLE POWER PLANT:

For the bypass factor of 0.30, it is found that at the pressure ratio of 12, the network of the combined cycle at n=0.20 and n=0.60 intersects and at n=0.80 the network becomes maximum.

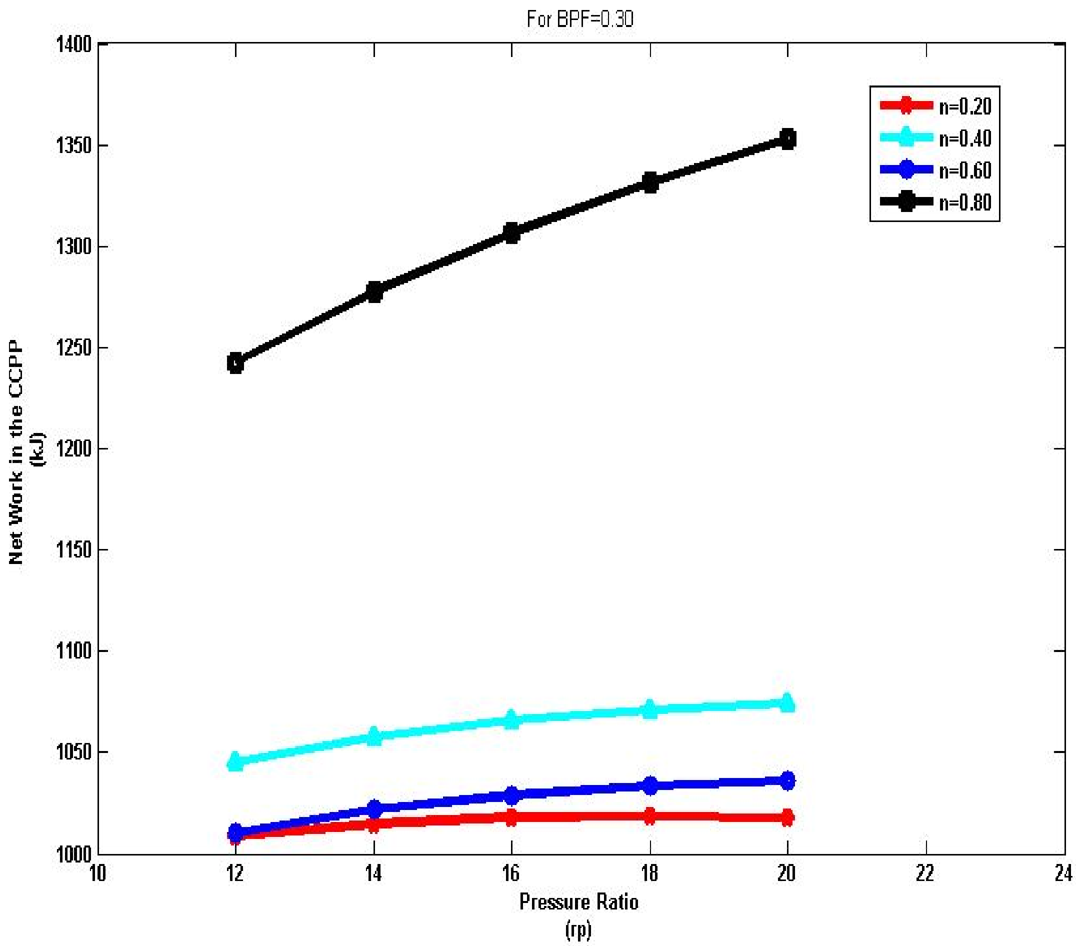

Fig. 6-14 Variation of Pressure Ratio with the Net Work during Reheating in the CCPP

6.1.2.11.PRESSURE RATIO VS GAS TURBINE REHEAT EFFICIENCY OF THE COMBINED CYCLE POWER PLANT:

It is found out that with the increase in the pressure ratio, the reheating efficiency of the gas turbine increases and also with the increase in the value of n the reheating efficiency increases. For n=0.80, reheating efficiency is maximum and for n=0.20, the reheating efficiency is minimum.

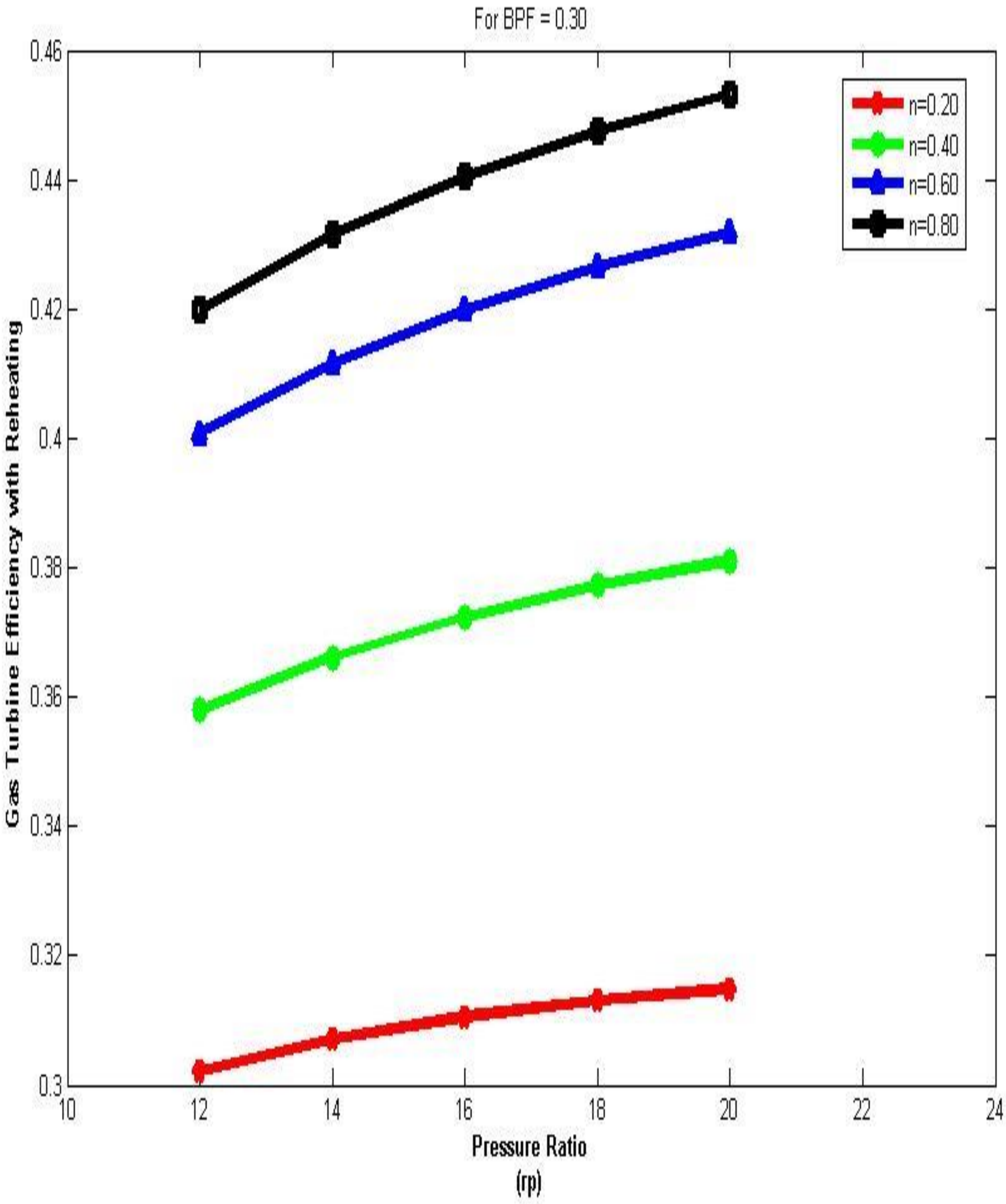

Fig. 6-15 Variation of Pressure Ratio with the Gas Turbine Efficiency with Reheating in the CCPP

6.1.2.12.PRESSURE RATIO VS OVERALL EFFICIENCY OF THE COMBINED CYCLE POWER PLANT:

It is found that with the increase in the n value the overall efficiency increases with the increase in the pressure ratio. Hence for n=0.80, the overall efficiency of the combined cycle becomes maximum.

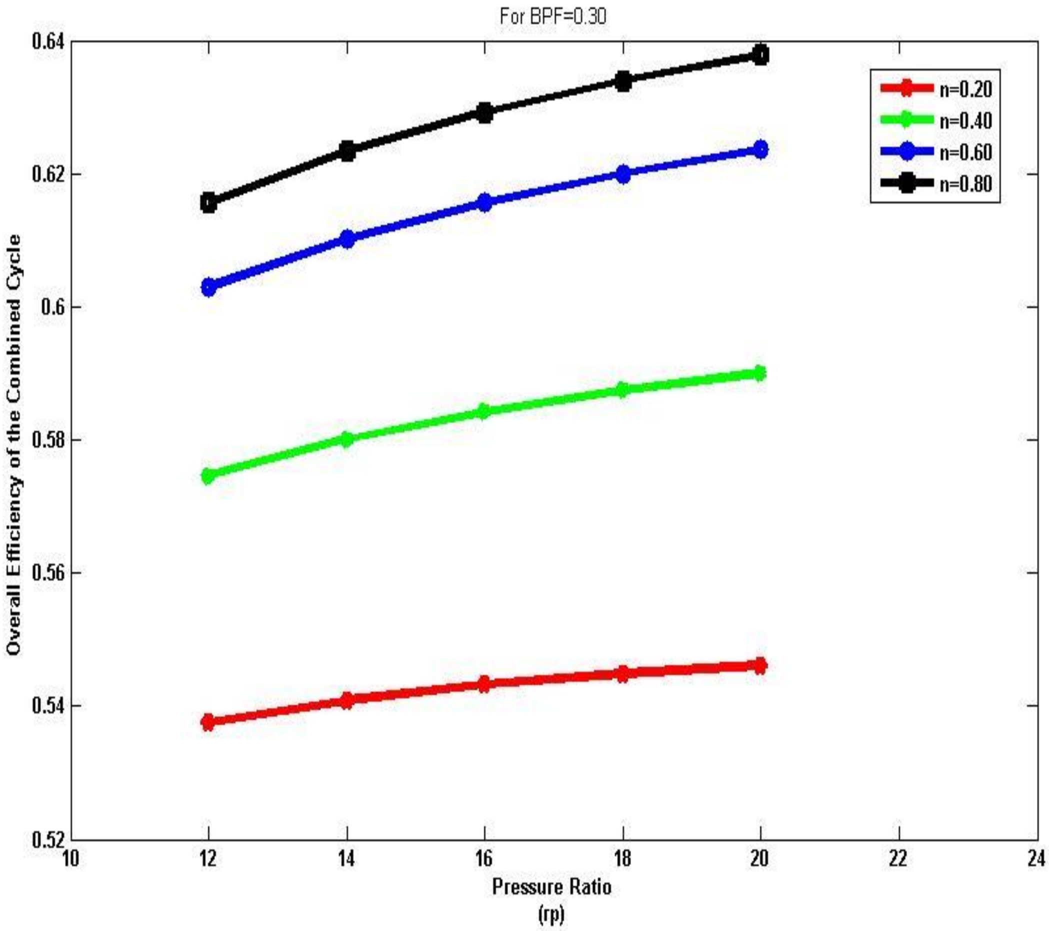

Fig. 6-16 Variation of Pressure Ratio with the Overall Efficiency of the CCPP

CHAPTER 7

CONCLUSIONS:

The first and second laws of thermodynamics are studied in this thesis for a combined cycle power plant. Thermodynamic Analysis of Energy is performed for different stages of the cycle using simultaneous equations.

According to this research, the CCPP system outperforms the Brayton and Rankine cycles individually

- In the first plot, we have taken the case for a simple gas turbine without the reheating condition and performed the bypassing at the exit of the combustion chamber and the remaining air enters into the steam turbine. Bypassing is done at 0%, 10%,15%,20%,25% and 30%. It was found that with upshot of bypass factor the magnitude of the overall efficiency decreases.
- Similar to the first case, we have found out the second graph that total work of the combined cycle power plant decreases with the increase in the bypass factor from 0%,10%,15%,20%,25% and 30%.
- In the third graph it was found that with the increase in the bypass factor from 0%,10%,15%,20%,25% and 30%, the mass of the steam increases with the increase in the steam pressure.
- Now in the reheating condition, it is found that the total reheating work is found to be same for n=0.20 and n=0.80 & n=0.40 and n=0.60 respectively due to the expression of the intermediate pressure shown in the methodology and literally it has no effect on the different values of the bypass factor.
- With the increase in the bypass factor, it is found that for the bypass factor of 0.20,it was observed that the net work done on the plant intersects at the pressure ratio in between 19 and 20 for n=0.20 and n=0.60.
- For BPF =0.20,the net work done increases for n=0.40.
- The total efficiency of the combined cycle power plant is found out to be 53% to 62%.

7.2. Recommendations

The following are some suggestions for future research:

- Natural gas is the working fluid in this research. When compared to the other organic working fluids, it has a comparatively high efficiency. While this feature is the primary reason for choosing this working fluid, it is recommended that you look into other organic fluids to see if you can get a more efficient operation. Binary organic fluids can also be investigated.

- HRSG is the main component of CCPP, and it can be further optimized by adjusting various HRSG parameters.
- Based on several literature references, a specific pressure range was chosen for this study. According to the findings, pressure changes have a considerable impact on performance. Changing the pressure ratio with the compressor's inlet temperature, on the other hand, may have a greater impact on performance.
- The emissions from the CCPP system can be decreased by switching to biomass fuel or utilizing a more efficient Stack after HRSG. However, if the carbon natural cycle is taken into account, CO_2 emissions from biomass fuel could be deemed to have a lower environmental impact.

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