Exergetic Evaluation of Regenerator and Reheater based Combined Cycle Gas power plant

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DECLARATION

I hereby declare that the work which is being presented in the major thesis entitled "**Exergetic Evaluation of Regenerator and Reheater based Combined cycle Gas power plant**" in the partial fulfillment for the award of the degree of Master of Technology in "**Thermal Engineering**" submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of **Prof. B. B. ARORA**, Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever. I confirm that I have read and understood the 'Plagiarism policy of DTU'. I have not committed plagiarism while completing the attached piece of work, similarity found after checking is 7% which is below the permitted limit of 20%.

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CERTIFICATE

This is to certify that SAGAR BHATIA, (Roll no. 2K17/THE/13), student of M.Tech., THERMAL ENGINEERING, Delhi Technological University, has submitted the dissertation titled "**Exergetic Evaluation of Regenerator and Reheater based Combined cycle Gas power plant**" under my guidance towards the partial fulfilment of the requirements for the award of the degree of Master of Technology under my guidance and supervision.

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ABSTRACT

Exergetic evaluation of combined vapour power cycle (CVPC) with regeneration and reheating is done in this study. Energy analysis gives quantitative aspect of the energy while exergy analysis is more qualitative. Due to lack of quality in the thermodynamic process there is wastage of energy which can be also denoted by another term i.e. exergy destruction and it affects the system's efficiency. A separate analysis has to be done for every major component we have and the whole information about losses occurring in the system along with their locations will be the outcome of this analysis. This outcome is very useful in analysing these mechanical components as it will define qualitative and quantitative aspect of the major losses occurring inside the system. This way of optimisation improves system performance by decreasing exergy destruction and exergy losses along with the increase in exergetic efficiency. Amount of exergy losses may vary in different components hence our primary objective is to perform energy analysis of these components and also pick those sites which are having major losses whether in the form of energy or exergy. This can also be done at different parameters, by varying load to get more accurate figures. Exergy flow rate at inlet and outlet of different components of a system is carried out like regenerators pumps and turbines. After performing all the calculations, we found that combustion chamber is the largest source of exergy destruction. Our calculations are based on the difference of exergy inflow and outflow.

We have N number of parameters which can affect system performance i.e. pressure ratio, AFR, temp. at inlet to compressor, ambient temperature, etc. but we will see the effect of AFR and inlet temp. on the exergy destruction rate of various components. With the use of EES a program code is made to perform basic calculations required in this area of exergetic evaluation.

Keywords: combined cycle, heat recovery steam generator, exergy, gas turbine, thermodynamic optimisation.

Scientific Field: Mechanical Engineering, Thermal engineering, Technical science

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NOMENCLATURE

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

Greek Symbols

γ	heat capacity ratio	-
Ψ	specific exergy	kJ/kg
η	efficiency	-
ρ	density	kg/m^3

Subscripts

CCGT	Combined cycle gas turbine
CV	Control Volume
PH	Physical
HRSG	Heat Recovery Steam generator
Dest.	Destruction
CC	Combustion Chamber
GT	Gas Turbine
ST	Steam Turbine
AFR	Air Fuel Ratio
PP	Pinch Point
CPPC	Combined power plant cycle
SPP	Steam power plant
CVPC	Combined vapor power cycle
GTPP	Gas turbine power plant
STPP	Steam turbine power plant
Eff.	Efficiency

CHAPTER 1

1. INTRODUCTION

Increase in population and the change in living standards of the people has led to drastic increase in the power demand. In the era of fast depleting conventional sources of energy and growing awareness towards environment, industry and the scientific community are doing efforts to improve the performance of thermal power plants. To achieve above we need to optimise our systems and research for technology which is environment friendly and highly efficient. There are many ways of utilising this heat like cogeneration, organic Rankine cycle, kalain cycle, etc. The performance can be increased by using the part of residual heat for preheating the air which is entering the CC. The energy eff. can also be increased by adding heat in such a way that a part of it is utilised in reheating the working fluid in successive expansion stages. Intercooling is another method of increasing eff. by dropping compression work and cooling of working fluid between successive compression stages. To improve eff. one innovative technology is used now a days that is "combined vapor power cycle (CVPC)".

A CVPC is comprised of two cycles, one is GT cycle (topping cycle) and other is steam turbine cycle (bottoming cycle) coupled with HRSG. An open cycle GTPP consist of 3 parts such as air compressor, CC and a GT. In air compressor where the air is entered into air compressor in which air is compressed to raise the temp. and pressure of air and then air continuously enter into the CC in which fuel is added to increase energy. After the combustion will take place the process in the combustion chamber is almost at CP. The flue gases enter into turbine to expand and produce work. Brayton cycle works at high temp. than Rankine cycle. The steam temp. at the turbine entry can be about 600°C for ST but it can be as high as 1450°C for GT. Now a days there is upgradation of turbine blade materials, cooling methods and high temp. resistant coating of the blades. In GT cycle energy is supplied at high temp. in order to obtain high thermal eff. but it also has some drawbacks like the turbine exhaust temp. is very high and is of the order of 520°C. Hence it makes engineering sense to utilise this energy as an input source for Rankine cycle. Rankine cycle consist of boiler, turbine, condenser and a pump. The energy from flue gases is utilised to make steam in HRSG which serve as a boiler for steam turbine cycle. Then expansion of steam takes place in turbine to produce useful work which is then condensation of steam takes place in condenser.

To analyse the performance of these plants 1st law and 2nd law of thermodynamics are being used. 1st law of thermodynamics only gives quantitative evaluation or it can be used in determining the overall eff. of power plant. It cannot give results for the losses in a system and also not able to define the location of these losses. This is the main drawback of 1st law of thermodynamics but it can be overcome by applying 2nd law of thermodynamics because it is the 2nd law that assign quality to the energy hence it is also known as qualitative law. Exergy analysis upholds 1st and 2nd law of thermodynamics. Exergy of a system is maximum useful work obtainable when system attains dead state. Exergy analysis gives us an insight of exergy dest. and degradation of energy in each part of the system. It has become a significant tool in determining amount of energy in the components. Exergy degrades in a process. Exergy analysis gives us a uniform base to relate different thermodynamic procedures. This analysis is useful in demonstrating the data with respect to the losses that incorporate their area subjectively and quantitatively. For the further change in the outline and operation of framework this data can be useful. The system eff. can be enhanced by finding the exergy dest. and by raising the exergetic eff. of the components and the system.

1.1 Objective of thesis

Motivation behind this work is to suggest some ways which can be incorporated in industry to minimise losses and enhance efficiency of a power plant. Exergy optimisation method is the best way to do this. Following are the steps by which we can do this:

1. Locate the regions where exergy destruction is taking place.
2. Methods to reduce major losses.
3. Specify the component where major exergy destruction is taking place.
4. Ways by which efficiency of the whole system can be increased like regeneration or reheat.
5. To find out the optimum value of the various parameters which directly or indirectly affects system efficiency.

1.2 Layout of the thesis

There are 7 chapters in this thesis. Exergetic evaluation is done and optimisation strategy is concisely considered.

Chapter 2 presents a literature survey, research and optimisation about exergy and energy analysis.

Chapter 3 Description of various components of combine cycle power plant.

Chapter 4 Energy and Exergy balance equations for various systems.

Chapter 5 Mathematical simulation CVPC.

Chapter 6 Outcomes of working variables by varying different parameters.

Chapter 7 Deduce the results of study.

CHAPTER 2

2. Literature Survey

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

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

Moreover, the optimization can be dissected from a thermodynamic perspective, as per the first and additionally 2nd law analysis, or utilizing a thermoeconomic or environmental-economic strategy (Kaviri et al [1], Ahmadi and Dinner [2], Boyano et al [3] and Petrakopoulou et al [4]).From the perspective of optimization technique, there are numerous kinds of analysis. In this work, the survey will feature the most basic technique: the exergy dest. method, and the exergoeconomic method.

2.1 Review of Analysis and Optimization of Topping Cycle

2.1.1 Thermodynamic Analysis and Optimization

The gas turbine operating parameters which impact the combined cycle gas turbine performance are; ambient conditions, Compressor pressure ratio, and turbine inlet temp.

2.1.1.1 The Effect of Ambient Conditions

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

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

Hosseini et al [8] shown that the gas turbine compressor is intended for consistent air volume flow, which makes the electric power output reliant on the ambient temp. through the particular mass flow rate. They included that the increase in the ambient temp. additionally diminishes the compressor's output pressure, which lessens the gas turbine cycle efficiency, while the increment noticeable all-around thickness decreases the gas turbine's heat rate and builds its particular fuel consumption. They expressed that for each 1°C increment in the ambient air temp., the electric power output of the gas turbine diminishes by 0.5% to 0.9% , and by 0.27% for a combined cycle.

2.1.1.2 Effect of Compressor Pressure Ratio

The properties of air entering the combustion Chamber depending on the Compressor pressure ratio as reported by Ibrahim et al [5], Ibrahim and Rahman [9], and Khaliq and Kaushik [10].

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

2.1.1.3 Effect of Turbine Inlet Temperature

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

Sanjay [11] expressed that the parameter that influences cycle performance most is the turbine inlet temp. TIT. The TIT ought to be kept on the higher side, as at lower values, the exergy dest. is higher.

2.2 Review of Analysis and Optimization of Bottoming Cycle

2.2.1 Thermodynamic Analysis and Optimization

The efficiency of Steam power plants can be improved by expanding the live Steam and reheat-Steam parameters, and by introducing high-efficiency, low-loss turbine blade geometries. The main objective, to increase the Steam parameters, is fundamentally accomplished by picking proper materials for the components working under live-Steam and reheat-Steam conditions while retaining the proved designs. Collaborative European projects have prompted the advancement and capability of steels with much-improved creep properties at temp. of up to 600 °C, suitable for the production of key components. In the meantime, optimization of the blade profiles and geometries permitted further significant upgrades in operating efficiency. The reachable enhancements in efficiency are about 0.5% per 10 °C live Steam and reheat (RH) temp. increase and 0.2 % per 10 bar pressure increase. The second significant piece of the bottoming cycle is the heat recovery Steam turbine (HRSG), its design and optimization influences to an enormous degree impact the efficiency and the expense of the entire plant.

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

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

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

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

Mansouri et al [15] investigated the effect of pressure levels of Steam generation at heat recovery Steam generator HRSG on the energetic and exergetic efficiency of HRSG, bottoming cycle and combined cycle power plants, as well as the effect of HRSG (heat recovery Steam generator) pressure levels on exergy destruction at HRSG and other main components of the bottoming cycle. Their result show that an increase in pressure levels of Steam generation at HRSG leads to an increase in the exergy efficiency of HRSG and CCPP increase respectively. In addition, an increase in pressure levels at HRSG decreases the exergy destruction due to heat transfer in HRSG: the exergetic efficiency of HRSG increases with an increase in pressure levels of Steam generation and adding reheat to the cycle.

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

2.3 Exergy Analysis

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

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

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

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

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

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

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

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

2.4 Exergy Destruction Method

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

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

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

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

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

Bassily [28] presented the effects of varying the inlet temperature of the gas turbine and PP on the performance of a dual pressure reheat combined cycle. He also modelled some feasible techniques to reduce the irreversibility of the HRSG of both cycles, and showed that optimizing or reducing the irreversibility of these cycles could increase their efficiencies by 2–3%. Applying gas reheat increases the generated power and average temperature at which heat is supplied, whereas applying gas recuperation takes advantage of the increased gas temperature at the outlet of the GT to enhance cycle efficiency. For gas-reheat gas-recuperated combined cycles, recuperated heat exchangers fabricated from stainless steel have to be used to withstand these conditions. He compared the optimized results with the regularly designed triple pressure reheat combined cycle Bassily. Boonnasa et al studied the performance improvement of an existing combined cycle power plant located in Bangkok that consisted of two gas turbines (110.76MW each), and one 115.14MW Steam turbine in ISO conditions. The plant used an absorption chiller to cool one of the two gas turbine's intake-air to 15°C, in addition to having a thermal energy storage tank that stored the sensible heat of the chilled water to meet the varying daily cooling load. Low-pressure Steam from a heat recovery Steam generator was used to drive the absorption chiller needed to meet a maximum load of 7049.58kW with the help of the thermal heat storage. As a result, the power output of the cooled gas turbine increased by

10%, improving the CCPP total power output by 6.24%. Economically, the study found that due to the low initial investment cost of retrofitting the absorption chiller the internal rate of return was 40%, and the payback period was just 3.81 years. However, the authors also reported a reduction by 2.85% in the Steam turbine power output, which was due to powering the absorption chiller directly from the HRSG unit Steam that was powering the Steam turbine. This reduction in the Steam turbine power output could have been avoided if they had used a boiler that utilized the waste heat energy from the stack after the HRSG unit.

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

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

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

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

CHAPTER 3

3. Description of Combined Cycle Power Plant

3.1 Introduction

The CVPC is an assembly of heat engine cycles that work as a pair from an identical heat source, changing it into the mechanical energy, it typically drive electrical generators. By combining two or more thermodynamic cycles there is an enhancement in overall thermal eff. and the fuel cost also gets reduced. A gas turbine plant is used for peak load conditions for certain hours of a day, when there is high demand of power. Therefore, high temp. exhaust from a gas turbine plant is utilised by operating gas turbine in concurrence with steam turbine to offer the gas turbine advantages of quick starting and stopping, less installation cost and fast response to load changes.[31]

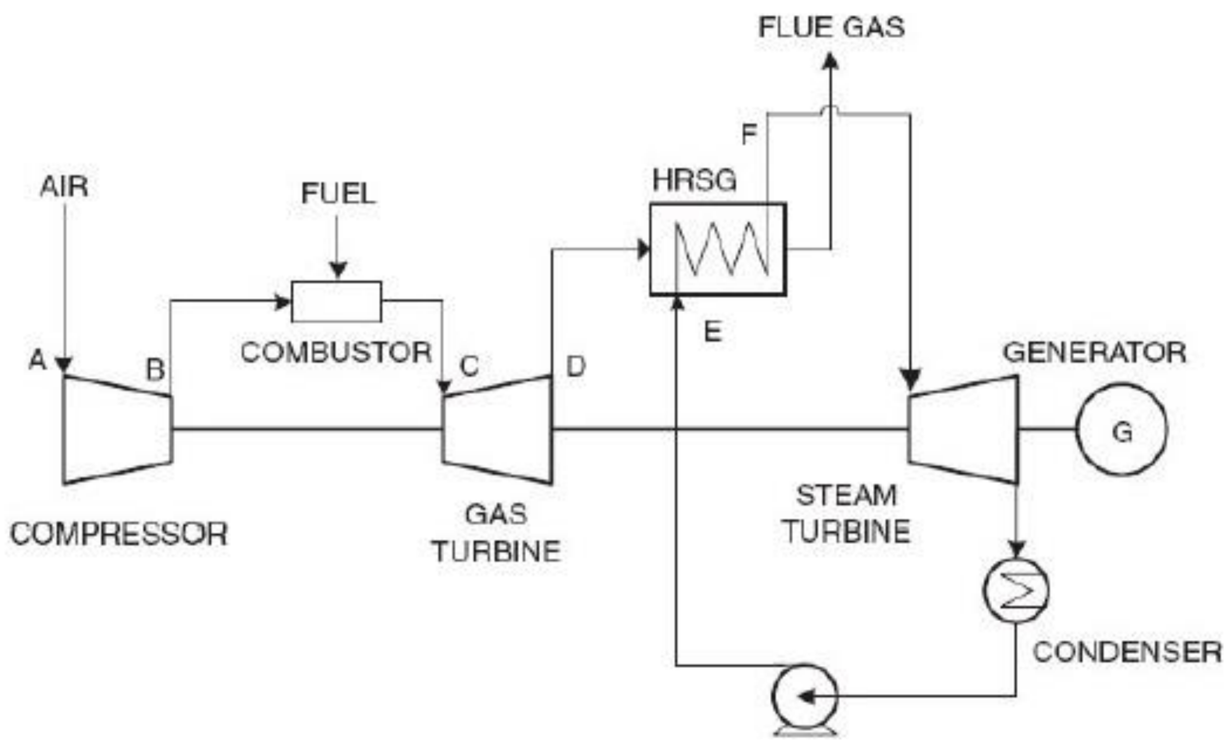


Fig. 3.1 Schematic diagram of combined vapor power cycle. [32]

3.1.1 Compressor

In many GT units which has a capacity more than 5 MW axial flow air compressors are being used. In axial flow compressors, working fluid enters into compressor in a direction parallel to axis of rotation of rotor. Compression take place by kinetic energy transfer from rotor. For larger units with high pressure ratios like industries and marine gas turbines. Axial flow air compressors are more efficient and is usually preferred.

It consists of a row of moving blade mounted on the circumference of rotor and a row of fixed blade mounted on the casing. In axial flow compressors, air flows along the axial direction and it consist of number of stages. Each stage consists of a set of fixed blades and moving blades and cross-section between the blade passage is of diverging type. There is a rise in static pressure due to diffuser effect in moving blades along with the increase in velocity which is further converted into fixed blade.

Based on aerodynamic theory the blades of compressor are of air foil shape. For efficient operation the blades of compressor are so designed that the losses due to shock and turbulence must be minimized and the blades are free from stalling effects. The fixed blades convert a part of Kinetic Energy (K.E.) of the fluid into useful pressure energy. This conversion process is achieved by diffusion process carried out in the diverge blade passages. These blades also guide the flow of fluid in such a way that there is no shock at the entry of next stage.

Fluid is quickened by pivoting air foils (front lines) called the rotor and a while later diffused successively of the stationary cutting edges (stator). Compressor includes a couple of stages: (1) A rotor and stator makes up one stage in a compressor. (2) An extra column of stationary sharp edges are routinely used in blower bay and these are the IGV to ensure that air enters the primary stage rotors at ideal stream point, these vanes are furthermore contributed variable this way may be changed as per moving stream requirements of motor.

In a pivotal stream blower, air goes beginning with one phase then onto the following, every stage raising the pressure to some degree. Higher efficiencies can be obtained by lower pressure augments on the solicitation of 1.1:1 to 1.4:1. In some aeronautical areas the utilization of various stages licenses generally speaking pressure increments of up to 40:1 and a pressure ratio, 30:1 for industries.

A lot of work has been done in the area of Gas turbines in the past two decades. The improvement is driven by extension in blower weight extent, propelled ignition systems, the advancement of materials development, new coatings, and new cooling plans.

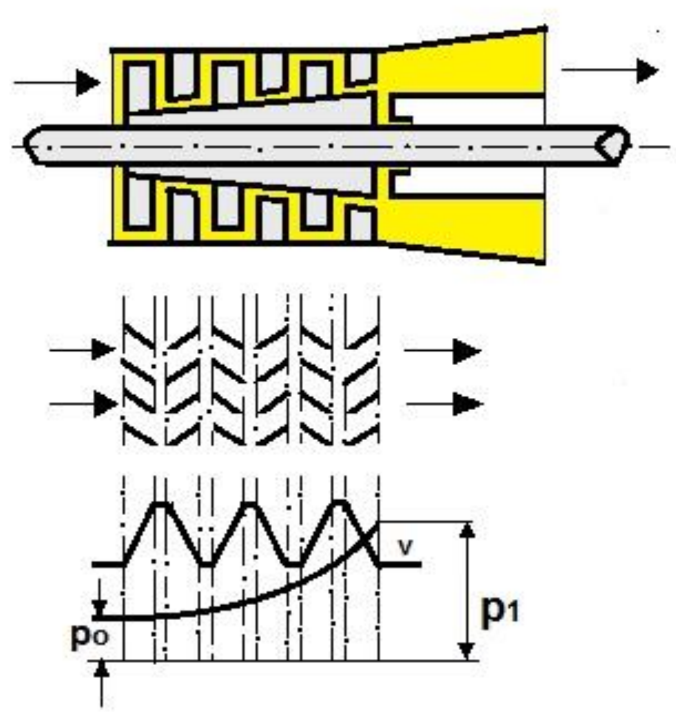


Fig. 3.2 Axial Flow compressor. [33]

3.1.2 Combustor

The combustor is the component of gas turbine where combustion of air fuel mixture occurs. In gas turbine motor, the combustor is bolstered high weight air by the pressure framework. The combustor then warms this air at steady weight. Subsequent to warming, air goes from the combustor through the spout control vanes to the turbine. On account of a ramjet or scramjet motors, the air is legitimately sustained to the spout.

The proportion of fuel gave to the air relies upon requested temp. raising. In any case, the most outrageous temp. is confined in 850-1700^{o}C between time, contingent upon characteristics of materials of which turbine vanes and edges are established and by cooling course of action of same sharp edges and vanes. The air enters in consuming chamber with temp. associated with the region of 200 and 550°C in perspective on the warming expedited by weight, so the required temp. raising in the start chamber is 650-1150°C.

The air starting from the compressor enters the CC with speed that can accomplish 150 m/s. This speed is exorbitantly high for start, so first that must happen in a combustor is a spread, that is a deceleration of air with a subsequent augmentation of the static weight. The fuel devouring speed to the conventional mixing rate air-fuel is of little meters/second, any flame gained in these conditions would be overpowered. Appropriately, a low-speed hub district must be made in the chamber with the goal that the flame stays ignited and secured in all conditions of engine activity. In customary conditions, the total air-fuel rate in a start chamber can vary in the region of 60:1 and 130:1. The fuel just expends in a viable way to deal with a rate near 15:1, along these lines the fuel must be singed just with a bit of entering the air in that which fundamental zone is called. This is gotten by a flame tube that has a couple gets to that direct the air scattering along combustor. Generally, 20% of as far as possible enters in start chamber through the nose. Quickly downstream to the nose, there are whirl vanes and a punctured plate through which the air goes in the fundamental zone of the combustor. The air that begins from the spins starts a development of circulation in the fundamental zone.

3.1.3 Gas Turbine

It is a work producing device which conveys power or thrust. The dynamic vaporous fluid is used to deliver impetus and to control. The turbine changes over the energy of the gas into mechanical work. Some portion of this work is utilized to drive the compressor and the rest is utilized to drive the shaft of electric generator . In the power era field, the GT can be utilized as a remain solitary unit or with CVPCs. Power creating GTs are generally open cycle worked. The GT execution relies upon the execution of its segments i.e. compressor, CC, and turbine.[34]

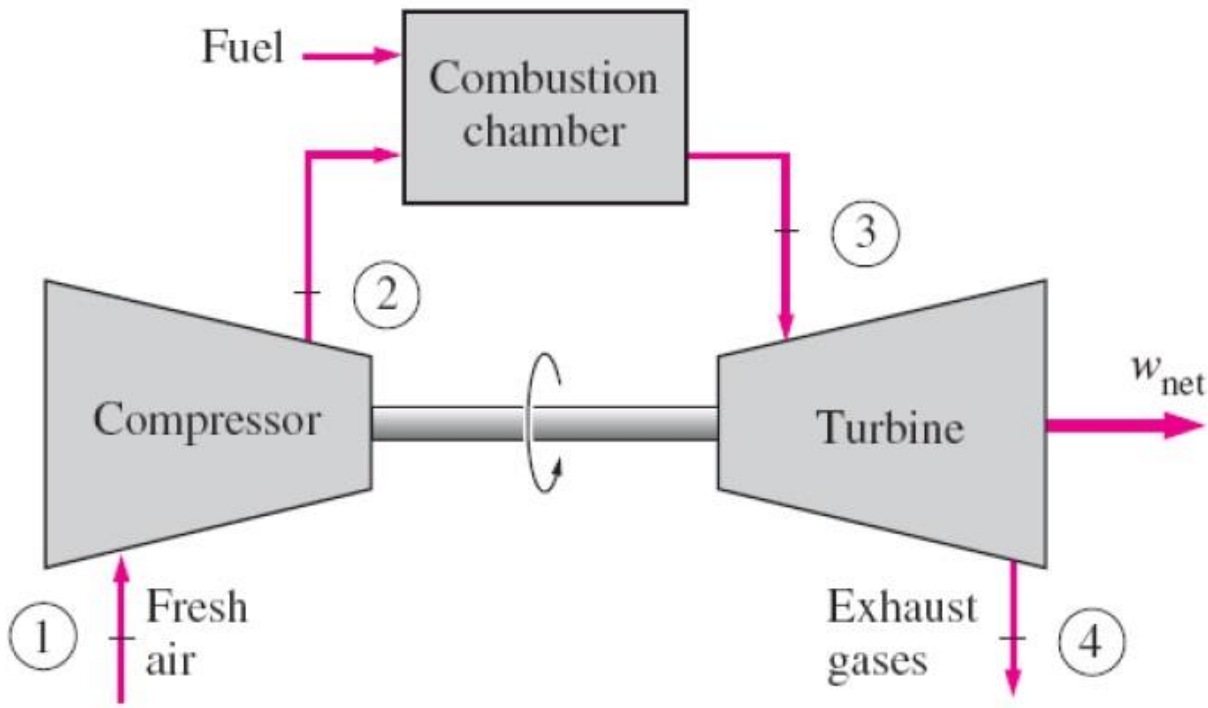


Fig 3.3 Diagram of open cycle gas turbine. [35]

GT generally works on open cycle. Air drawn into compressor at encompassing conditions at P_1, and the pressure rise take place to P_2, expanding both pressure and temp. at the costs of utilizing compressor work (W_{comp}) which is provided by the turbine itself. Then fuel is scorched into the ingoing air from the ignition chamber to add heat. This heat added (Q_H) raises the temp. from T_2 the turbine inlet temp. (highest cycle temp.) T_3. The subsequent gases at high temp. enter the turbine to produce useful work. Heat Q_L is rejected to the environment. The cycle 1-2-3-4 as shown in the figure below is known as Open cycle gas turbine as the exhaust from the turbine is thrown into the atmosphere. Parameters such as compression ratio and temperature at the inlet of turbine adversely affects efficiency of the system hence these parameters must be taken carefully.[36]

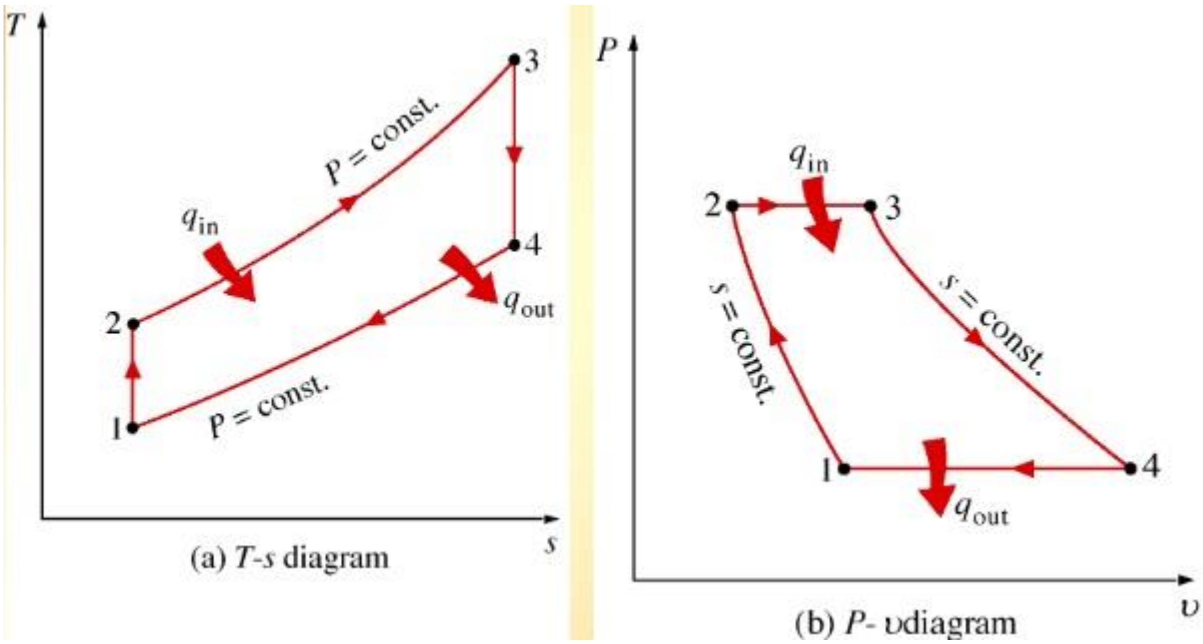


Fig 3.4 Diagram of Brayton Cycle [37]

3.1.4 Steam Turbine

It is a prime mover which converts heat energy of the steam into mechanical work and further utilized in power plant to produce electricity. The steam turbines are usually axial flow machine i.e. steam flow over the blade in a direction parallel to the axis of rotation. Steam turbines are classified on the basis of:

1. Principle on which it Operates

(a) Impulse Turbine

(b) Reaction Turbine

2. On the basis of shaft orientation

(a) Tandem Compounded: In this type of assembly all the turbines are connected to the single rotor

(b) Cross Compounded: In this type two separate shafts or rotor are used to run the two separates generator at different speed. It is generally used when volume handling capacity is very high.

Steam turbines expand the high pressure and high temperature steam to produce mechanical work. Expansion of Steam is allowed to occur gradually in more than one stage, for achieving significantly higher efficiencies. The steam develops through dynamic rings of moving edges on a shaft and fixed sharp edges in the packaging, making absolutely turning development. When turbine is joined with a shaft of generator, then it becomes a champion among basic techniques for making mass electric power on the planet. The propelled ST may have three phases. The high-weight portion has little sharp edges. They are little in light of the fact that the moving toward steam has high amount of energy at high temp. After the steam experiences the high-pressure region, it is sent back to the radiator to be heated. Then steam flows through the accompanying region of the turbine, called the intermediate pressure segment. Sharp edges in his region are greater than edges in high-pressure segment. In the wake of experiencing this fragment, the steam is sent to the low pressure region of the turbine. Since most of the energy was ousted from the steam, the edges here are greatest in turbine. Steam leaves turbine through its base, where it is merged into water. Starting from there it will again goes into the boiler, to convert into steam. ST is routinely used as a part of CVPC where turbine is driving a machine meanwhile: steam isolated from the machine is used for local heating as well as for process heat systems.

The most elementary cycle used for power production is Rankine cycle. In spite of the way that a depiction of the Rankine cycle can be found in any structure thermodynamics course book, it is immediately verified here. The working fluid in boiler feed pump is supplied to the boiler at a very high pressure. As the liquid is risen in the evaporator it passes through its 3 parts i.e. economizer evaporator and superheater. Isobaric process takes place in these parts to convert saturated liquid into HP vapor. After which the HP vapor made is reached out through ST, as such taking out heat from it and simultaneously producing work. Now, the high-density vapors are again required to be converted into saturated liquid so that it can be lifted to evaporator pressure. Hence a condenser is used here to this job by rejecting the large quantity of heat in turbine exhaust to the flowing cooling water.

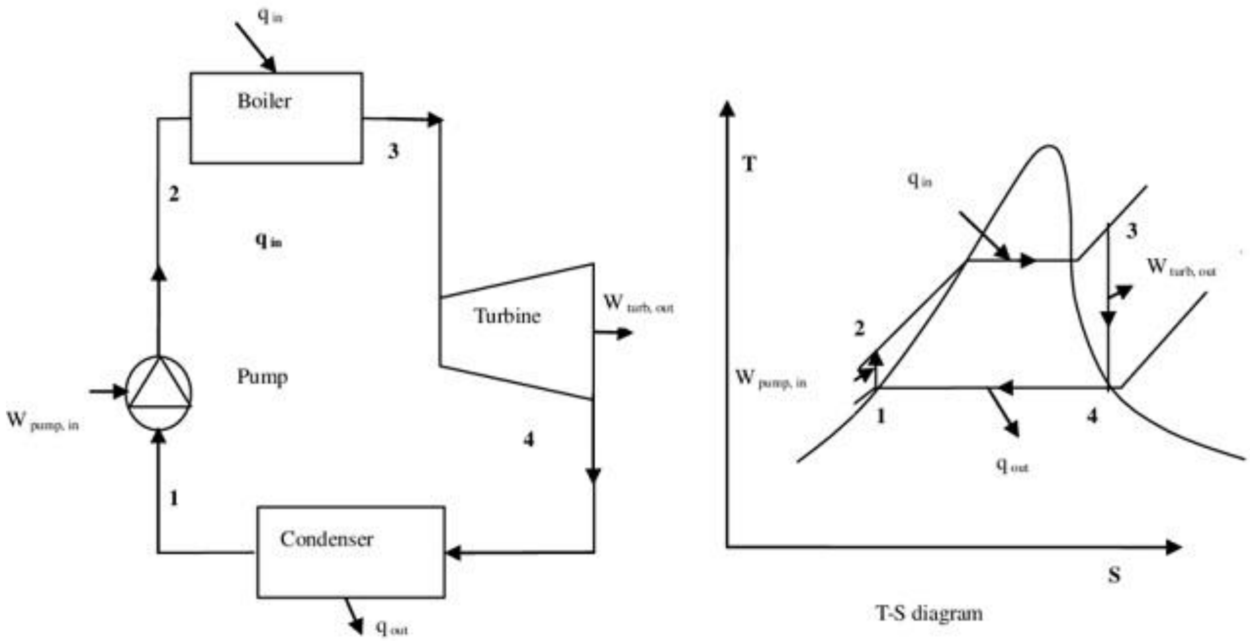


Fig 3.5 Diagram of Rankine cycle. [40]

A couple of modifications such as recuperation, regeneration and superheating are there to accomplish better results in the form of more efficient system having less losses and less operating cost. Steam can be gathered absolutely in the condenser and a robust segment of troubles identified in the heat engine cycle can be cleared with the help of superheating process in the evaporator. Losses in the form of internal irreversibility are minimized in ideal Rankine cycle to a large extent. Cycle consists of 4 processes: ,heat is added in the evaporator and the process is isobaric, then the steam is expanded in the expander and the process is reversible adiabatic, heat is required to be rejected to the cooling water in the condenser, fluid is pumped back to the evaporator by reversible adiabatic process in the pump. All of the 4 segments of Rankine cycle are resolute state suffering stream gadgets.

A solitary unit of ST can make power stretching out as small as 1 MW to as large as 1000 MW. Now a days we have steam power plants more than the capacity of 120 MW have thermal efficiencies as high as 38% and it can also go up to 40% [38] with some modifications. Water can work over a wide extent of temp. and pressure also it has enormous heat limit, unfaltering, safe, and biologically agreeable. Conventional sources are primarily used in power plants to deliver steam these can be in the form of coal of different category like bituminous and lignite, light diesel oil, gas and nuclear sources of energy can also be sed as a clean fuel.

Production capacity: Turbines coupled with generators can produce power as small as 500KW or maximum it can go up to 7500KW while considerable turbo alternators have cutoff contrasting from 10MW to as large as 90MW. There are constraints when the size of unit can be as large as 500MW. There can be N number of factors on which capacity depends like steam pressure and temp, exhaust pressure and number of stages, etc. Gigantic units of steam turbines have a usage of steam ranging from 3.5 - 5 kg for each kwh produced.[39].

3.1.5 Condenser

As it is known to be a direct transfer type heat exchanger therefore its role is to condense hot steam at the exit of turbine by rejecting its heat to the cooling water flowing through it. It serves following purposes:

1.If we want to increase available heat energy for converting into mechanical work, low pressure side must be maintained by the condenser at the outlet of turbine. It helps in maintaining low back pressure at the exit of turbine. By doing so expansion of steam takes place to a greater extent and work output will surely increase.

2. Evaporator requires pure feed water which is also supplied from this device as it gets collected in hot well and it can serve the purpose of feed water along with the make-up water.

To expel the heat of this exhaust steam from ST, condenser requires cooling water so that it can absorb the heat rejected by the steam. This job can be done by making flow wet steam around countless minimal chilly tubes in which water is made to flow. Cold water is ordinarily given from a close-by sea, lake, conduit, or from a cooling tower. To begin this cycle again in which water is forming into steam and steam again change its phase into water this dense steam is gathered at the base of condensor and then it will return to the evaporator by changing its phase with the help of reinforce water pumps. By reducing the temp. of condensed steam 6we can bring down turbine backpressure which will also let the system to work at a higher eff. and we will witness growing turbine power yield and productivity.

Accompanying auxiliary capacities of condensor:

1. An assembly of hot well is there from which condensate will be sucked and syphons will perform the required course of action.

2. At this very moment storing of condensate.

3. A very LP gathering point for condensate channels from various systems in plant.

4. Provide for de-air circulation of the gathered condensate.

Huge power plant condensers are normally 'shell and cylinder' heat exchangers.

These kinds of condensers are likewise ordered:

1. As single pressure or multi-pressure, dependent upon whether the cooling water stream route makes in any event one turbine backpressures;

2. By the quantity of shells (which is dependent on amount of low-pressure turbine lodgings); and

3. As either single pass or two-pass, dependent upon the amount of parallel water stream ways through each shell.

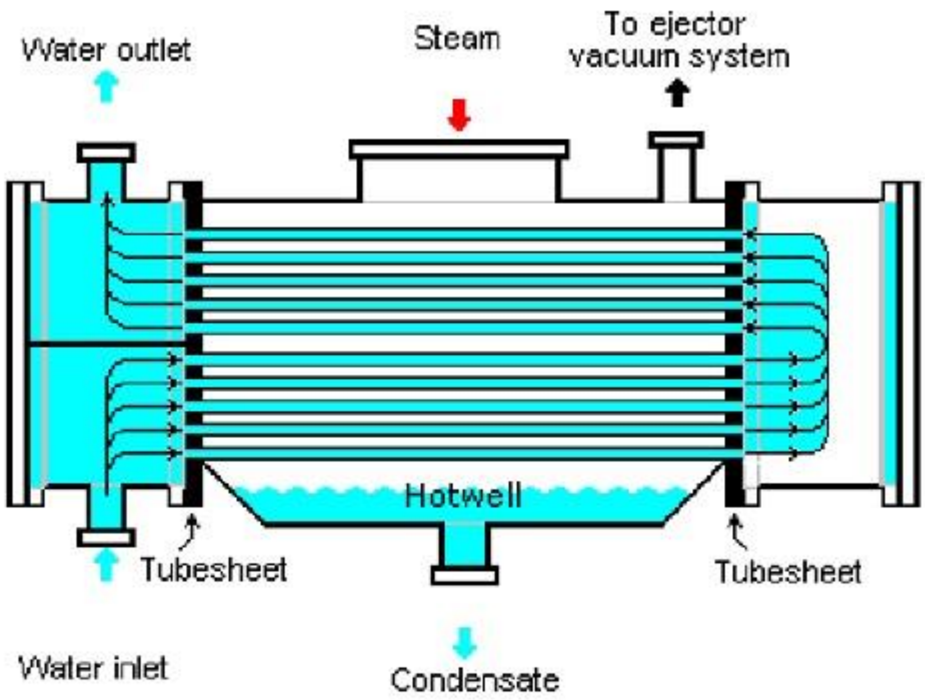


Fig. 3.6 Shell and Tube steam condenser [41].

3.1.6 Heat Recovery Steam Generator (HRSG)

In present advancement, there are most effective energy change frameworks to convey electrical and process heat energy are joined CCPP. Most of the time fuming heat from the GT exhaust can be recovered in HRSG and is used to form steam which is further utilized in CVPC as a working fluid to run other cycle which works in tandem with GT cycle. HRSG execution bigger influences the general execution of a CCPP. Steam produced in HRSG with various weight levels rely upon the plan. HRSG comprise of three heat exchanger bundles (economizer, evaporator, and superheater).

First the exhaust gas enter into the economizer then comes evaporator and lastly it enters into the superheater. 3 stages are there in which the addition of heat takes place from gas side to water side.

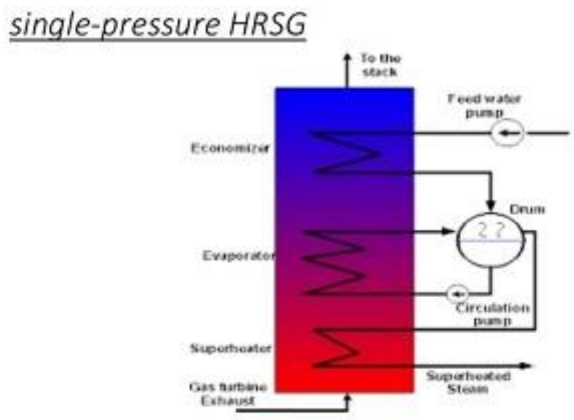


Fig. 3.7 HRSG of a Combined cycle [42].

Natural Circulation HRSG consist of vertical tubes and horizontal flow arrangement and the flow occurs due to density difference. In forced circulation type there are horizontal tubes and the flow occurs in vertical direction. Pumps are used in this arrangement to facilitate the flow.

3.1.6.1 Important HRSG design parameters

(a) Pinch point temp. : If we take difference of temp. of gas exiting evaporator and the temp. of saturated water in the evaporator we get PP temp.

(b) Approach point : Here we have two variables saturated steam and the water enterng into the evaporator, difference between these two gets the approach point. Care must be given in selecting these two variables as these can increase the size of heat exchangers we have which will eventually increase overall cost of equipment.

(c) For an economical design temp. and pressure at the outlet of HRSG or we can say at the inlet of turbine must be calculated.

(d) Superheater approach temp. : Superheated steam temperature and GT outlet temp. are the 2 variables to calculate this value. Difference between these two is taken to calculate this.

(e) Temp. of stack : Temp. of Exhaust that is thrown out in the environment.

(f) Allowable backpressure: Cross-sectional zone of HRSG fundamentally impacts turbine backpressure. Driving through the pipe becomes easy when there is higher gas turbine backpressure and smaller and compact HRSGs require this[43].

3.1.6.2 T-Q Diagram

T-Q outline exhibits profiles of heat transfer for fumes gas or steam, making use of temp. as y-coordinate and heat exchanged as x-coordinate.

For better understanding and designing combined cycles, the utilization of T-Q chart is essential. Fig. 3.8 demonstrates T-Q outline for a single pressure combined cycle. Least temp. difference in HRSG is known to be pinch-point (PP), and it is situated at the cold side of evaporator. The fig. depicts temp. profile of flue gases and water or steam as well. The temp. of flue gases can be represented by a line which is having constant slope and it is the upper line. Temp. profile of water or steam is shown by lower line on this diagram. 3 distinct segments are there in a single pressure HRSG for a combined cycle.

First segment: This is also regarded as economiser because the heat addition takes place at lowest temp. Water from the boiler feed pump is heated at CP in this segment and is in the liquid form i.e. saturated liquid. To avoid the phenomenon of water hammering in this segment we should always keep the temp. of water below saturated state. At this state evaporation could occur and the vapor bubbles formed can cause blockage of flow hence this must be avoided for proper functioning of the system.

Second segment: Here in this area the process is at constant temp. and water change its phase to saturated vapor to form steam.

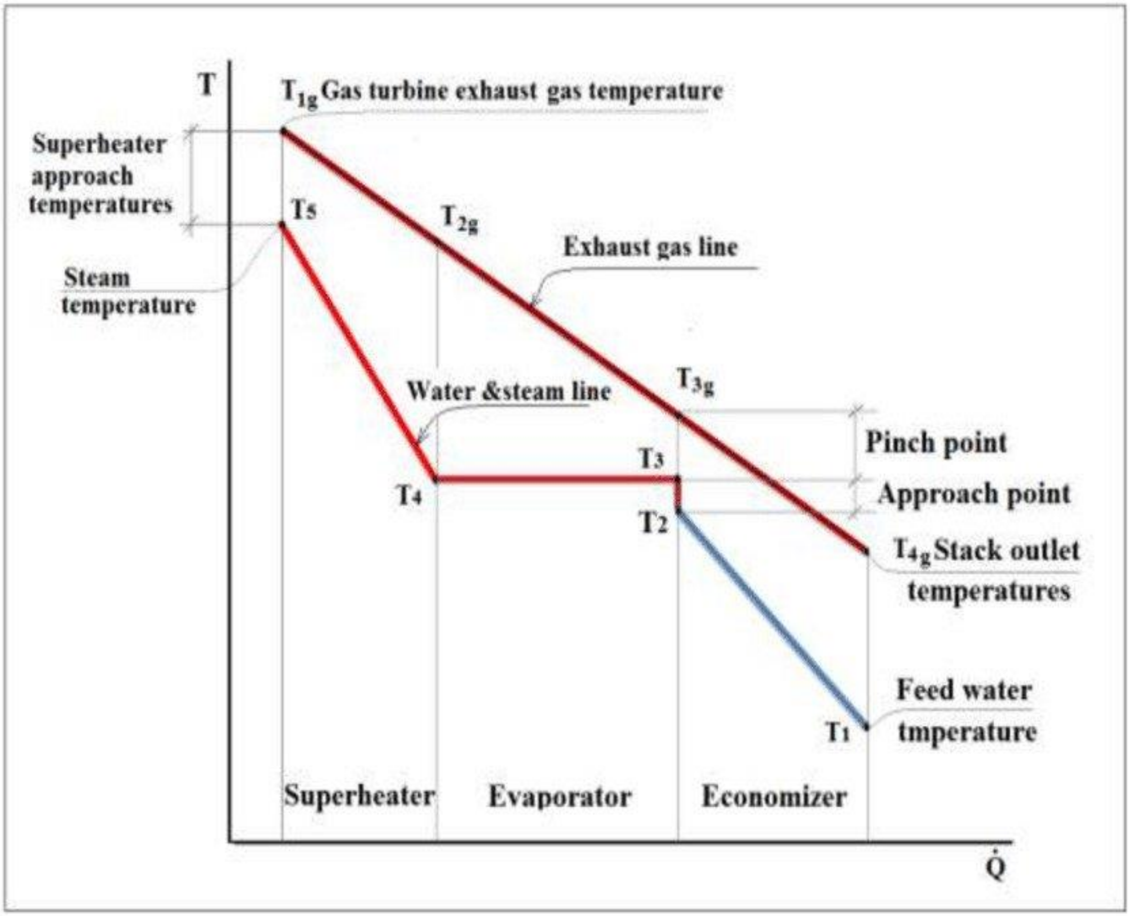
Third segment: The component which forms the third segment is the superheater. Here the temp. of steam is raised at CP.

Relation between temp. and heat can be represented by the formulae given below:

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

The above relation holds true when there is not a phase change of working medium. Evaporator helps in changing the phase of water to steam [54], process (3-2) can be written in the form of equation:

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



Fig, 3.8 T-Q Diagram of HRSG [44]

3.1.7 Combined Cycle Power Plant (CCPP)

Conventional CCPP comprises of the mix of Brayton and Rankine cycle resulting in most effective cycles utilized for power generation these days. The combination of two mostly used cycles i.e. Brayton and Rankine at the top and bottom make up the Combined cycle. Thermal eff. can be fairly high for these cycles in todays world and it can also surpass 60%. GT cycles consistently work at a much higher temperatures as compared to steam based cycles. Largest fluid temp. is nearly 620°C at the inlet of turbine in today’s scenario for steam based power plants, for GTPP the limit is very high and it can go up to 1425℃. Late upgrades in turbine blade cooling and applying different layers of covering with the help of high temp. safe materials on the blades has made it possible that we are using such higher temperatures.There is an immense potential in these units to work continuously and efficiently, this happens only because of higher temp. which is encountered at the inlet of

GT. In any case, there is one intrinsic shortcoming of these systems as exhaust gas always leave from GT at high temp. which is of the order of 500°C and it will erase all the possibilities or if there is a potential to enhance eff. of the system. Here in bottom cycle we are using energy which can be considered as waste energy from the top but it always looks good to capitalize amazingly alluring characteristics of GT cycle and whether the change we are getting is narrow but we will get increment and conditions can be strengthened with this recovery. Energy can be retrieved from flue gases and this can be utilised to form steam by transferring it to the saturated liquid in a heat exchanger which will be accumulated in boiler drum. With everything taken into account, (at any rate one) sufficient heat is required to form steam which must be be supplied by the GT. Furthermore, we can employ some modifications in the form of regeneration and reheating in ST cycle which can raise thermal efficiency. With the effective utilisation of exhaust gases and fuel sources, energy for the heating mechanism can be provided.

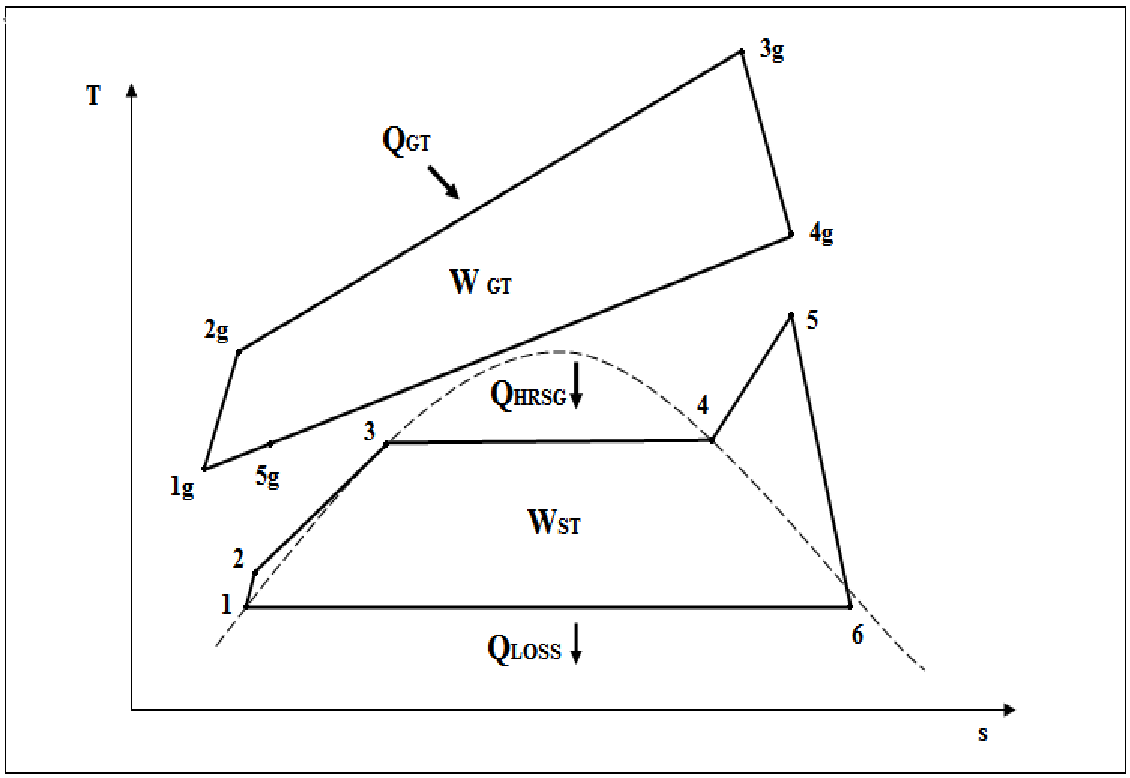


Fig. 3.9 Diagram of CCGT cycle [45]

CHAPTER 4

4.Exergy Analysis- Background

An essential background towards improvement of exergoeconomic advancement is explained in this chapter, with dictating conditions important for accomplishing the optimization objective. The below mentioned steps explain the exergoeconomic enhancement strategy.

4.2 Exergy Analysis

Exergy is explained by: the greatest theoretical work acquired through one system when it’s taken from one state to equilibrium along the surroundings while having an interaction with surroundings only. Temp., Pressure and Composition characterize the state of that particular system. Apart from this, exergy denotes the base theoretical work expected to take away the system from equilibrium with nature to the mentioned state. This suggests that exergy depicts the take-off of the state of one system from the state of the environment. Hence, exergy is a property of the system as well as condition. The significance of exergy won't get completed, regardless, till some reference condition is specified.

Exergy examination can be considered as a methodology which makes use of Preservation of mass along with protecting the energy standards altogether the 2nd law of thermodynamics so that the changing of assessment, plan and energy system can be done. Exergy strategy can be considered as one of the significant instruments which encourages the idea of productive utilization of energy resource, hence engaging the areas along with authentic extents losses which should be settled.

4.2.1 Exergy of a System

The exergy which is taken out with the help of heat and work processes can be explained as total exergy. Therefore,

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

Exergy associated with work transfer: Work is considered to be proportional to the energy as a measure of its exergy, obviously work and exergy are equated with each other.

Therefore, one can show exergy transfer with regard to both size as well as bearing with the help of work transfer with which it is compared

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

Exergy associated with heat transfer: When it is assumed a steady temp. dispersal in a heat energy reservoir, the rate of exergy transfer, $\dot{\Delta}E_{work}$ related to the rate of heat transfer Q, is expected from consequent technique work:

$$\dot{\Delta}E_{heat} = \left(1 - \frac{T_0}{T}\right)\dot{Q} \qquad 4.3$$

Putting the value of (4.2) and (4.3) in equation (4.1),

$$\dot{\Delta}E_{system} = (Q - T_0 \int dS) - (W - \int P_0 dV) \qquad 4.4$$

and rearranging

$$\dot{\Delta}E_{system} = Q - W - T_0 \int dS + \int P_0 dV \qquad 4.5$$

First law is given as:

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

Using equation (4.6) in (4.5)

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

Expanding

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

Finally, after integration,

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

Here, V = Relative velocity of system with frame of reference as surroundings

z = system's height relative to frame of reference of surroundings (where $z_0 = 0$).

4.2.2 Exergy Component

Overall exergy of a system $\dot{E}$ is separated in 4 parts: $\dot{E_{PH}}$, $\dot{E_{KN}}$, $\dot{E_{PT}}$ and $\dot{E_{CH}}$:

Where, $\dot{E_{PH}}$ = Physical Exergy

$\dot{E_{KN}}$ = Chemical Exergy

$\dot{E_{PT}}$ = Potential Exergy

$\dot{E_{CH}}$ = Chemical Exergy

$$\dot{E} = \dot{E_{PH}} + \dot{E_{KN}} + \dot{E_{PT}} + \dot{E_{CH}} \qquad 4.10$$

4.2.2.1 Kinetic Exergy

$\dot{E_{KN}}$ can be taken as the kinetic energy obtained by taking relative velocity of movement w.r.t surroundings:

$$\dot{E_{KN}} = m\frac{1}{2}V^2 \qquad 4.11$$

4.2.2.2 Potential Exergy

$\dot{E_{PT}}$ can be calculated from P.E. w.r.t. ground level accompanied with environment. As P.E. must always take into consideration every one of strengths influencing analysed matter along with the condition, this implies, other than surrounding constraint, the drive made by the pressure of nature parts should be similarly considered. Assuming that 'g' does not get altered as we move upwards, P.E. can be figured using condition:

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

Where g, h and γ_0 are acceleration due to gravity, height of the centre of matter w.r.t ground level and density of surroundings respectively.

4.2.2.3 Physical Exergy

$\dot{E_{PH}}$ is classified as one piece of exergy which is obtained because of the variation in the observed matter's temp. and pressure and the temp. and pressure of surroundings (T_0, P_0). It is automatically divided into two components.

Thermal component $E_{PH}^{\Delta T}$, is that component which is due to the temp. change between the stream and the environment:

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

Pressure component $E_{PH}^{\Delta P}$, is the component which is due to the pressure change between the stream and the environment:

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

4.2.2.4 Chemical Exergy

When a mixture is taken, its general form of equation of chemical exergy is given below,

$$\dot{E_{CH}} = \sum x_n(\dot{e_0})_n + \overline{R}T_0 \sum x_n \ln x_n \quad 4.15$$

$$\dot{e_0} = T_0 \ln \frac{p_0}{p_{00}} \quad 4.16$$

Condition of fuel specific exergy is mentioned below:

$$\dot{e}_{CH,fuel} = \beta(LCV) \quad 4.17$$

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

where c, h, o, n denote fractions of mass of the corresponding elements.

4.2.3 Exergy Balance

When the 1st and 2nd law of thermodynamics are combined, the relation of exergy balance can be given as[46]:

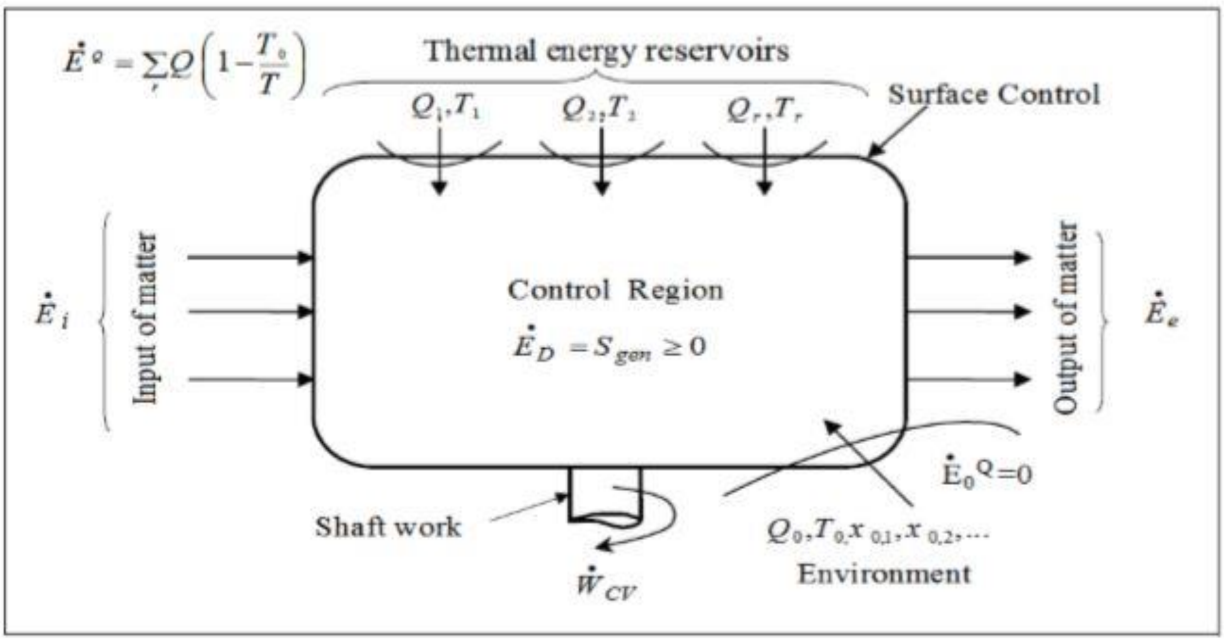


Fig. 4.1: Steady state process in an open control region

For an open system, the expression for exergy analysis is given as:[43]

$$\left(\frac{d\dot{E}}{dt}\right) = \sum_j \left(1 - \frac{T_0}{T_j}\right)\dot{Q}_j - W_{CV} + \sum m_i \dot{e}_i - \sum m_e \dot{e}_e - \dot{E}_D \quad 4.19$$

4.2.4 Exergy Wastes (Exergy Destructions and Exergy Losses)

The exergy loss finds relation with generation of entropy condition.

$$\dot{E}_{loss} = \dot{E}_{out} - \dot{E}_P \quad 4.20$$

$$\dot{E}_D = \dot{E}_{in} - \dot{E}_{out} \quad 4.21$$

$$\dot{E}_{waste} = \dot{E}_D + \dot{E}_{loss} \quad 4.22$$

During the examination of exergy, it's important that fuel along with the product is characterized for the system as well as for each element.

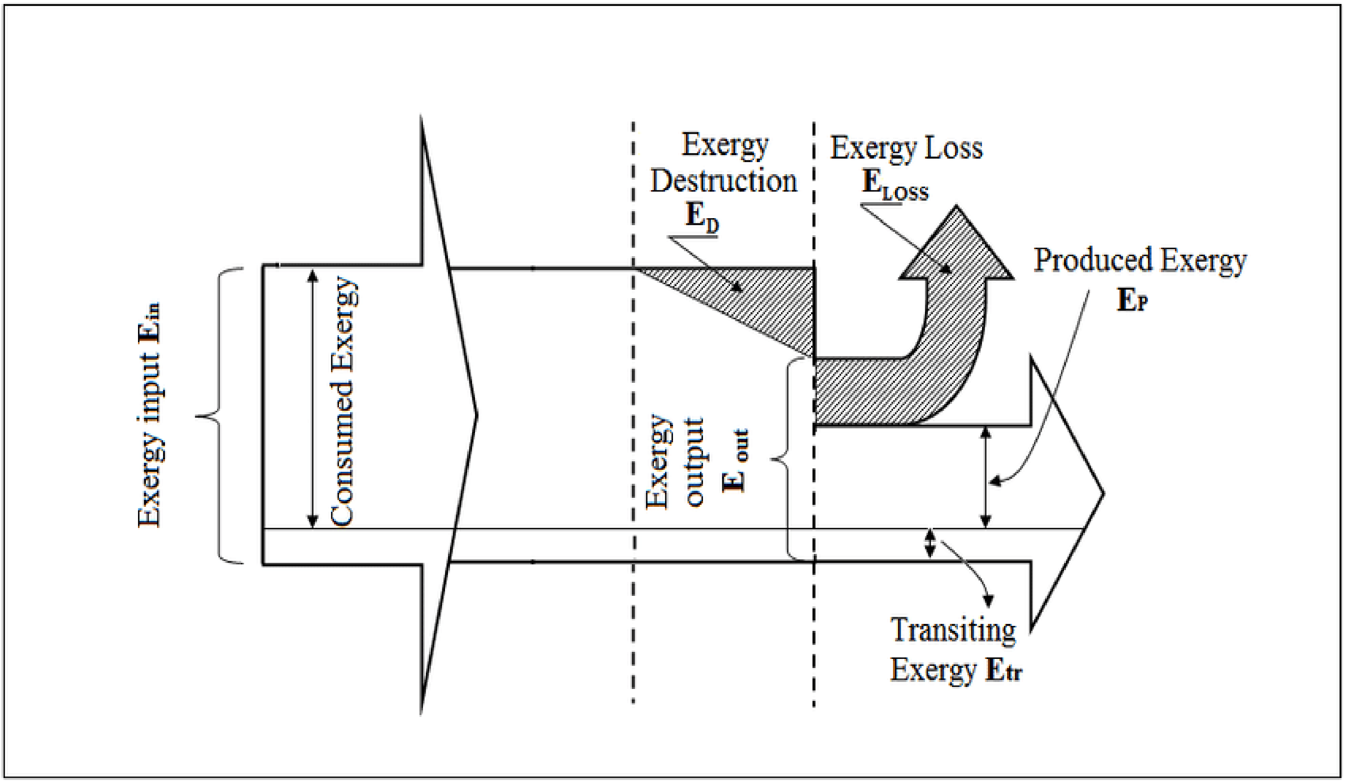


Fig. 4.2 Overall exergy balance shown using pictorial representation [47]

Condition (4.19) is used for determining the destruction of exergy.

Condition 4.23 gives the exergy losses condition of a control volume at steady state having almost zero kinetic and P.E.

$$\dot{E}_D = \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_J - W_{CV} + \sum \dot{E}_\iota - \sum \dot{E}_0 \quad 4.23$$

Exergy balance exists for reversible systems. In this way, for genuine techniques, for example irreversible methods, exergy is never in balance, in light of the fact that the total exergy input reliably outperforms the total exergy output. In this way, it is misleading to talk about an exergy modification for authentic techniques.

By establishing the exergy waste, for example pulverization and loss, we can envision possible procedure upgrades. All things considered, upgradation of the element must be taken into consideration when the loss of exergy is high. Regardless, the methodology isn't commonly fitting. The reason is that all parts of the framework depend upon one another and an adjustment in one area may increase wastes in various elements which can result in combined wastes for whole balanced process fundamentally greater as compared to that of primary procedure arrangement. Thusly, the issue needs an even more accurate attention, that is discussed underneath [48].

Exergy wastes ratio is given as:

$$y_D = \frac{\dot{E_D}}{\dot{E}_{F,TOT}} \quad 4.24$$

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

$$y_D^* = \frac{\dot{E_D}}{\dot{E}_{D,TOT}} \quad 4.25$$

Also,

$$y_{loss} = \frac{\dot{E_{loss}}}{\dot{E}_{F,TOT}} \quad 4.26$$

4.2.5 Exergy Efficiency

Exergy efficiencies can be utilized for different purposes. A conspicuous application is to utilize them for surveying, scrutinizing, and optimizing procedures and frameworks. Exergy efficiencies are especially important in breaking down and upgrading frameworks. An exergy examination more often than includes a thorough estimation of the exergy standards of process flows and the exergy waste in the framework. Such a count demonstrates the spots in the framework where waste happen. In the examination, the question that must be addressed is the way the exergy losses can be prevented or minimized. In light of the estimation of the exergy loss, it is typically hard to assess whether an exergy loss in an apparatus is pointlessly enormous. An exergy eff. in which the exergy loss is related with the additional or transferred exergy gives a better image of the value of the procedures in the mechanical assembly, and in this way gives a superior impression of whether exergy waste can be decreased.

CHAPTER 5

5.Formulation of Equations

5.1 Exergy Analysis of CVPC with Regeneration and Reheat

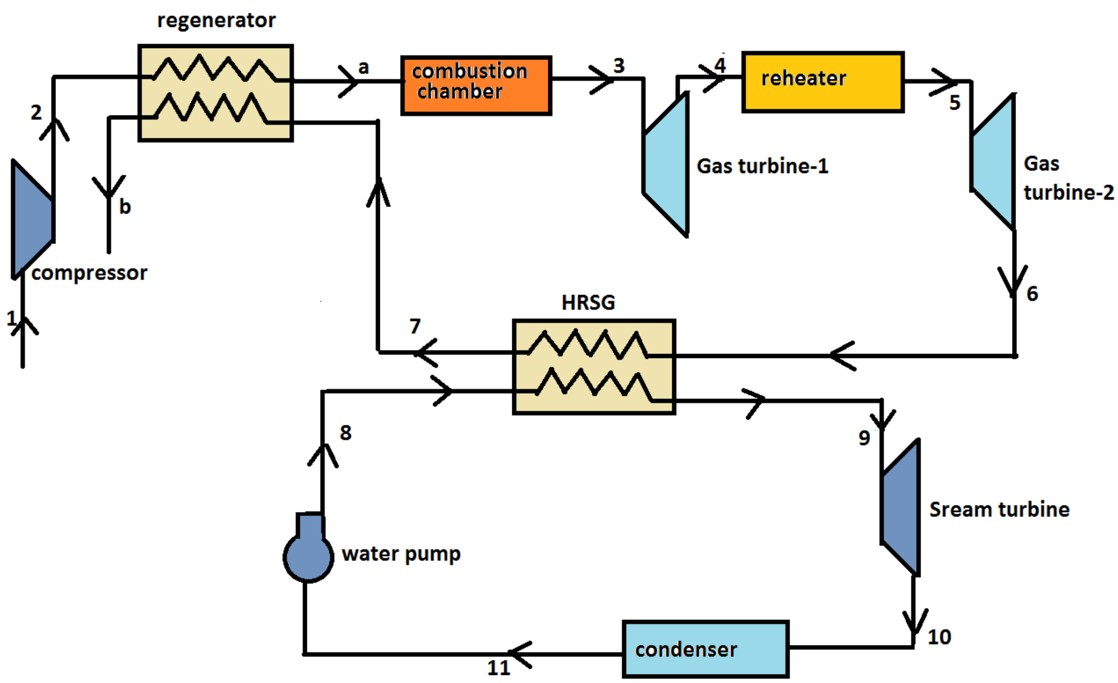


Fig 5.2 Schematic diagram of CVPC with regenerator and reheat

Table 5.1: Table indicating various points of schematic diagram

Sr. No.	Points on diagram
1	Compressor inlet
2	Outlet of compressor
a	Air entering the CC
3	Combustion gases entering GT_1
4	Gases exiting GT_1
5	Reheated gases entering GT_2

6	GT_2 exit
7	Exit of HRSG
b	Gases at stack temp.
8	Entry Water in HRSG
9	Superheated steam at ST inlet
10	Steam at ST exit
11	Saturated liquid entering feed water pump

Introduction of this segment belongs to the fundamental conditions utilized in performing exergy analysis on the selected CVPC. Exergy analysis along with energy analysis for all the components present in the system is done whether it is exergy balance or exergy dest. calculation to define the irreversibilities for every major c0mponent have been done. Finally, we have found overall exergy eff. and exergy dest. for the entire framework.

Equations for exergy eff. and decimation of exergy for every other component in the bottoming cycle are determined with the application of exergy balance and also the condition given in equation (4.23) is used.

We are assuming ideal insulation hence the loss which would be there due to exergy transfer due to loss of heat energy from HRSG will be taken as nil.

5.1.1 Compressor

$$X_1 + W_C = X_2 + \Delta X_{dest} \quad 5.1$$

$$\Delta X_{dest} = X_1 - X_2 + W_C \quad 5.2$$

Where

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

$$X_2 = \dot{m}_a[h_2 - h_0 - T_0(s_2 - s_0)] \quad 5.4$$

5.1.2 Combustion Chamber

$$X_a + X_f = X_3 + X_{cc} \quad 5.5$$

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

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

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

Where

$$e_{ch} = \dot{x}_i e_{chi} + RT_0 \sum x_i \ln x_i + G_e \quad 5.9$$

Where G_e is Gibbs free energy which is a negligible quantity in a gas blend worked at low pressure. So, for the determination of fuel exergy, the given expression does not hold good. Subsequently, the fuel exergy can be determined as the ratio of fuel exergy to lower heating value of fuel-

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

e_f is the specific exergy of the fuel.

For gaseous fuel with composition C_xH_y, the value of Ω can be calculated as

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

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

Then

$\Omega = 1.06$

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

5.1.3 Gas Turbine

$$\Delta X_{dest} = X_3 - X_4 - W_T \quad 5.13$$

Where

$$X_4 = \left(\dot{m}_a + \dot{m}_f\right)[h_4 - h_0 + T_0(s_4 - s_0)] \quad 5.14$$

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

5.1.4 HRSG

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

Where,

$$X_6 = \left(\dot{m}_a + \dot{m}_f\right)\psi_6 \quad 5.17$$

$$X_8 = \dot{m}_w\psi_8 \quad 5.18$$

$$X_{in} = X_6 + X_8 \quad 5.19$$

$$X_7 = \left(\dot{m}_a + \dot{m}_f\right)\psi_7 \quad 5.20$$

$$X_9 = \dot{m}_w\psi_9 \quad 5.21$$

$$X_{out} = X_7 + X_9 \quad 5.22$$

$$\eta_{II} = \frac{X_8 - X_9}{X_6 - X_7} \quad 5.23$$

$$(X_6 - X_7) = \left(\dot{m}_a + \dot{m}_f\right)[(h_6 - h_7) - T_0(s_6 - s_5)] \quad 5.24$$

$$(X_8 - X_9) = (\dot{m}_w)[(h_8 - h_9) - T_0(s_8 - s_9)] \quad 5.25$$

5.1.5 Steam Turbine

$$X_9 = \dot{m}_w\psi_9 \quad 5.26$$

$$X_{10} = \dot{m}_w\psi_{10} \quad 5.27$$

$$(X_9 - X_{10}) = (\dot{m}_w)[(h_9 - h_{10}) - T_0(s_9 - s_{10})] \quad 5.28$$

5.1.6 Condenser

$$X_{in} = \dot{m}_w\psi_{10} + \dot{m}_{cw}\psi_{cw,in} \quad 5.29$$

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

5.1.7 Pump

$$X_{11} = (\dot{m}_w)[(h_{11} - h_0) - T_0(s_{11} - s_0)] \quad 5.31$$

$$X_8 = (\dot{m}_w)[(h_8 - h_0) - T_0(s_8 - s_0)] \quad 5.32$$

CHAPTER 6

6. Results and Discussion

After the investigation of various cycles on which the thermal power plant work as for exergy. The exergy devastation demonstrates a loss that can be recuperated by utilizing the reasonable structure of the different parts of the framework and furthermore it affirms the most ideal procedure of the influence plant as indicated by 2nd law of Thermodynamics. As the exergy and cost of energy are complimentary to one another exergy dest. demonstrates a misfortune, which can be evaluate by investigation the framework in numerically. In the present work the investigation is done in the CVPC.

6.1 Results of Exergy Destruction Rate

6.1.1 Effect of AFR at various Inlet temperature

Fig. 6.1 depicts Exergy Dest. of Air Compressor vs AFR at various Inlet temperatures. As the AFR increases the amount of air is increasing but the amount of fuel remains constant i.e. 1.5 kg. Therefore, to compress more air work input to the compressor increases and hence exergy dest. rate also increases.

As the work input to the compressor is directly proportional to the inlet temp. to the compressor therefore, as the inlet temp. increases work input to the compressor increases hence exergy dest. rate also increases.

Fig. 6.2 reveals the effect of AFR on exergy dest. rate of regenerator at various inlet temperatures. At a particular temp. as the AFR increases the exergy dest. also increase because the heat supplied will increase. This will happen at a certain extent of inlet temp. to the compressor. As the inlet temp. to the compressor further increases. The temp. at the outlet of compressor will also increase and it will be more than the temp. of hot gases entering inside regenerator, hence the exergy dest. rate will be negative.

Fig. 6.3 depicts Exergy Dest. Rate of CC vs AFR at various Inlet temperatures. As the AFR increases heat supplied will also increase which lead to the increase in Exergy Dest. rate.

Fig. 6.6 depicts the Exergy dest. rate of HRSG vs AFR at various inlet temperatures. As the AFR increases exergy dest. rate will decrease because at high AFR inlet temp. in HRSG will also decrease.

Fig. 6.7 illustrates the Exergy Dest. rate of steam turbine with increase in AFR. As the AFR increases exergy dest. rate will decrease because the amount of fuel decreases and temp. at the inlet to the steam turbine decreases.

Fig. 6.8 depicts the exergy dest. rate of pumps vs AFR. Exergy Dest. rate will decrease as the AFR increase because the inlet temp. to pumps will decrease.

6.2 Results of Thermal Efficiency

6.2.1 Effect of AFR at various Inlet temperature

Fig. 6.9 shows the effect on GT thermal eff. as the AFR rises. As the AFR rises heat addition will decrease for the large volume of air added. Hence thermal eff. will decrease.

Fig. 6.10 depicts the effect of AFR on Overall thermal eff. of CVPC at various inlet temperatures. As the AFR increases The heat addition will be less for the large volume of air added because the amount of fuel remains same hence thermal eff. will decrease.

6.3 Results of Exergetic Efficiency

6.3.1 Effect of AFR at various Inlet temperature

Fig. 6.11 represents Exergetic eff. of a compressor vs AFR. AFR has no effect on Exergetic eff. hence it will remain constant as the AFR increases but, as the inlet temp. to the compressor increases the work input to the compressor will decrease for the same pressure rise. Hence the exergetic eff. will increase.

Fig 6.12 represents Exergetic eff. of CC vs AFR at various inlet temperatures. At a particular inlet temp. to the compressor and with the increase in AFR Exergetic eff. will increase.

Fig. 6.13 depicts the variation of Exergetic eff. of HRSG vs AFR. Exergetic eff. will increase with increase in AFR because the temp. of hot gases entering HRSG will be low at high AFR.

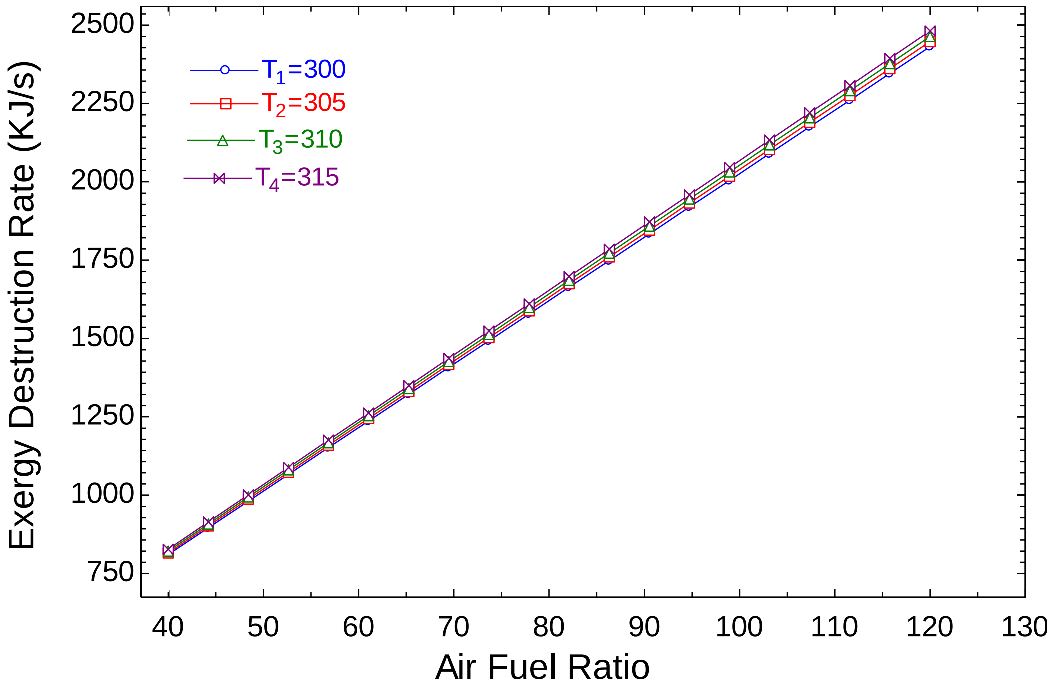


Fig. 6.1 Exergy Destruction Rate of air compressor vs AFR at various inlet temp.

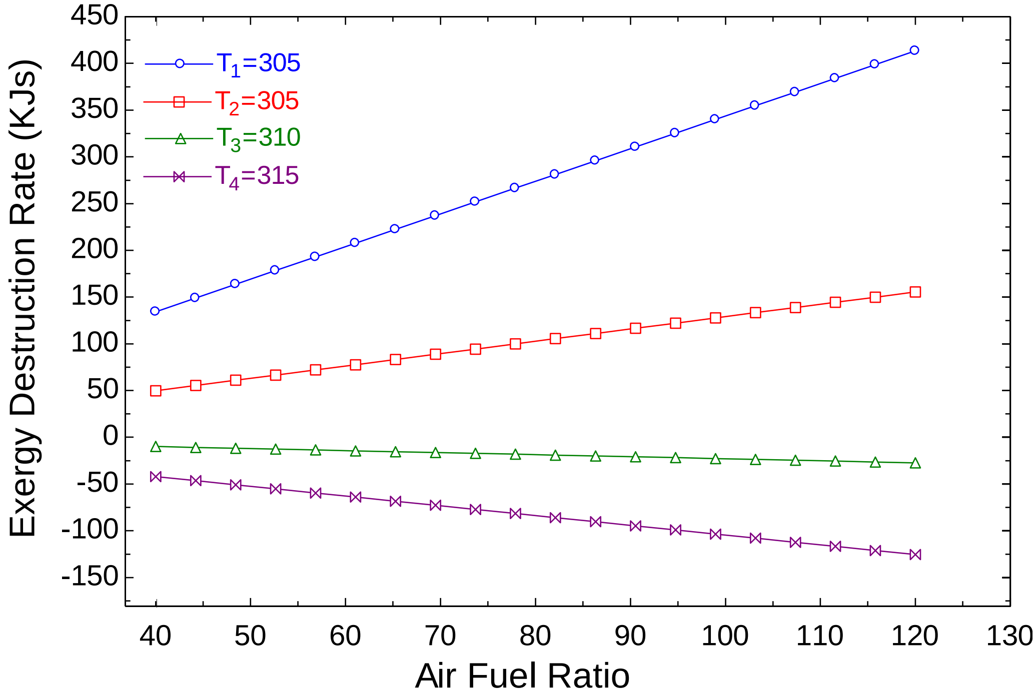


Fig. 6.2 Exergy Destruction Rate of regenerator vs AFR at various inlet temp.

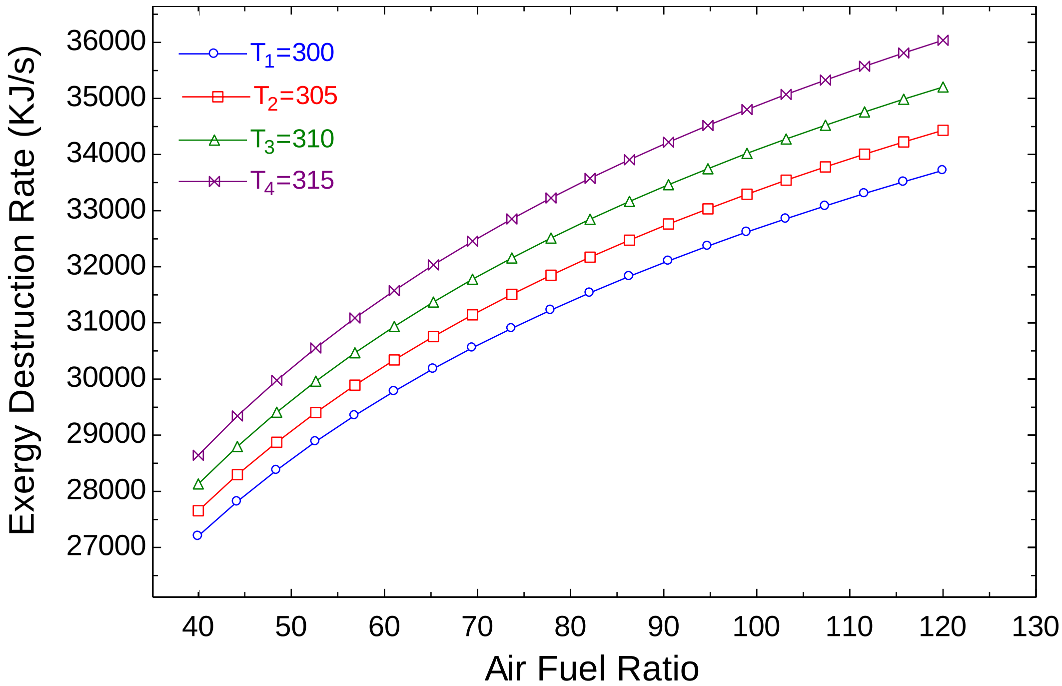


Fig. 6.3 Exergy Destruction Rate of combustion chamber vs AFR at various inlet temp.

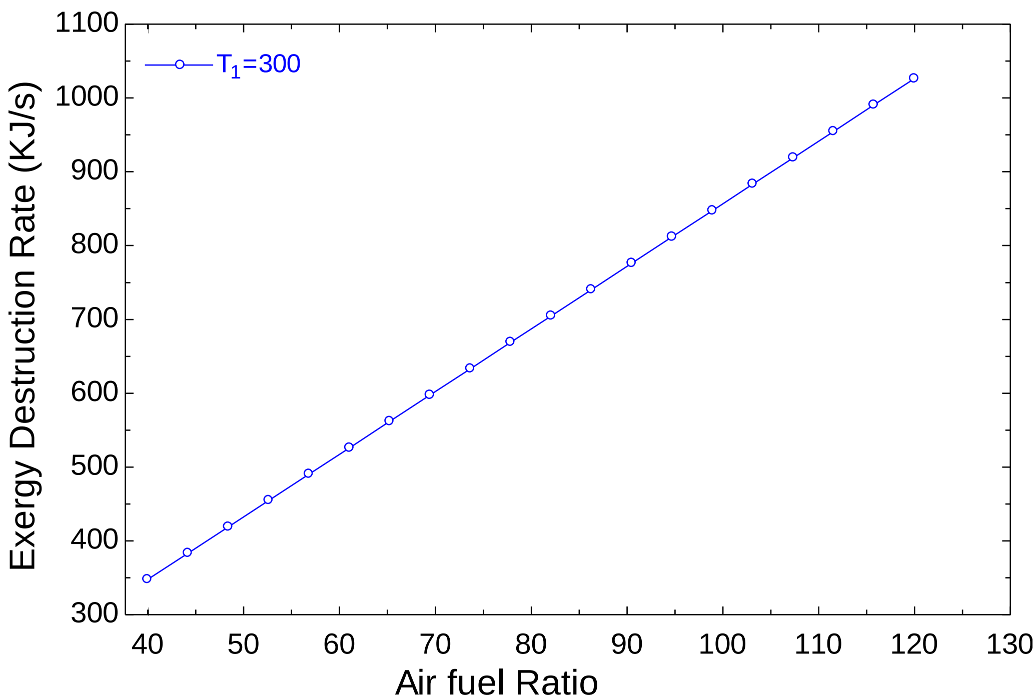


Fig. 6.4 Exergy Destruction Rate of gas turbine-1 vs AFR at inlet temp.

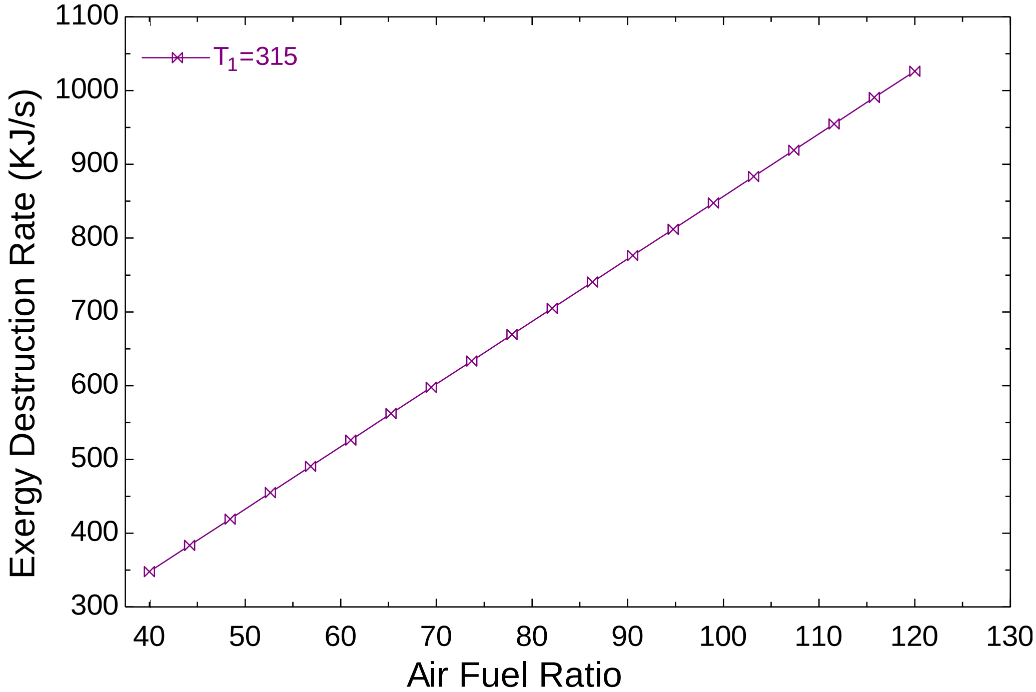


Fig. 6.5 Exergy Destruction Rate of gas turbine-2 vs AFR at inlet temp.

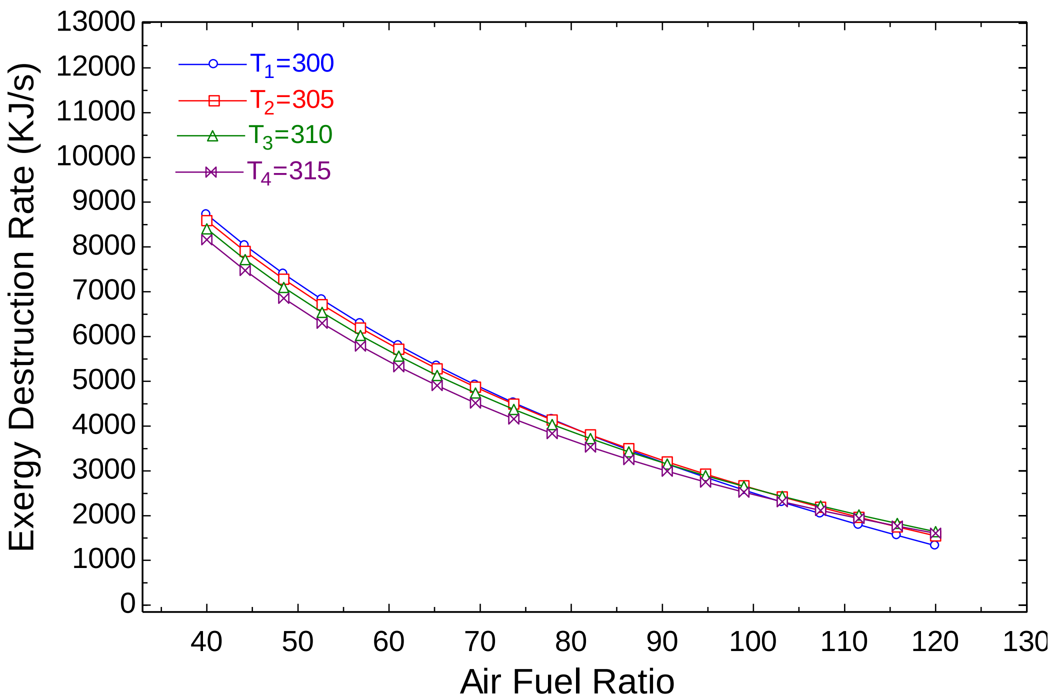


Fig. 6.6 Exergy Destruction Rate of HRSG vs AFR at various inlet temp.

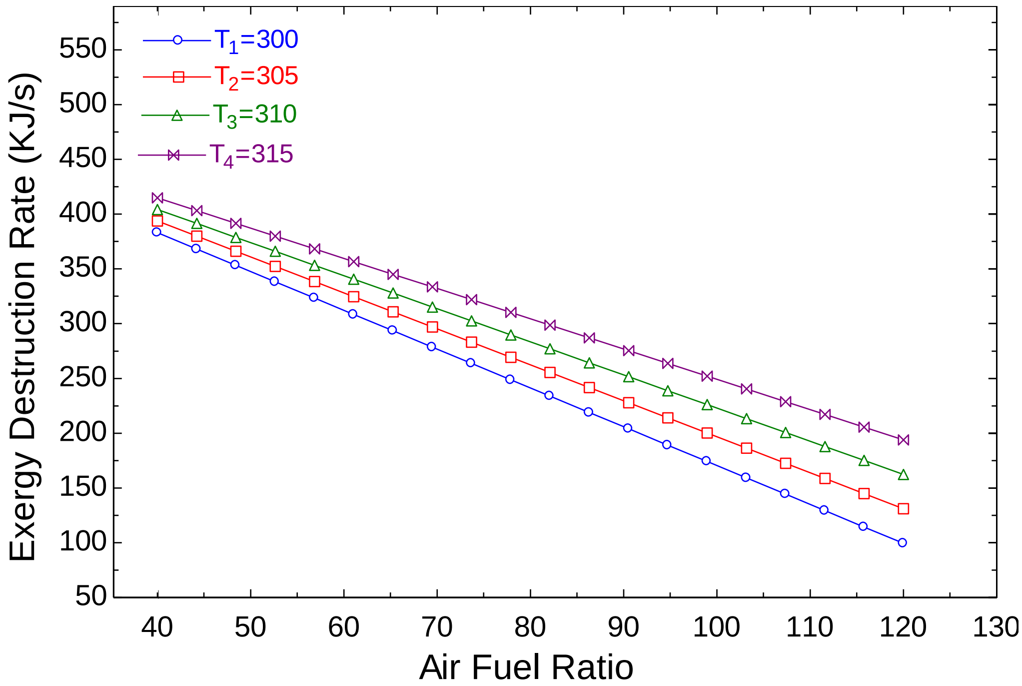


Fig. 6.7 Exergy Destruction Rate of Steam Turbine vs AFR at various inlet temp.

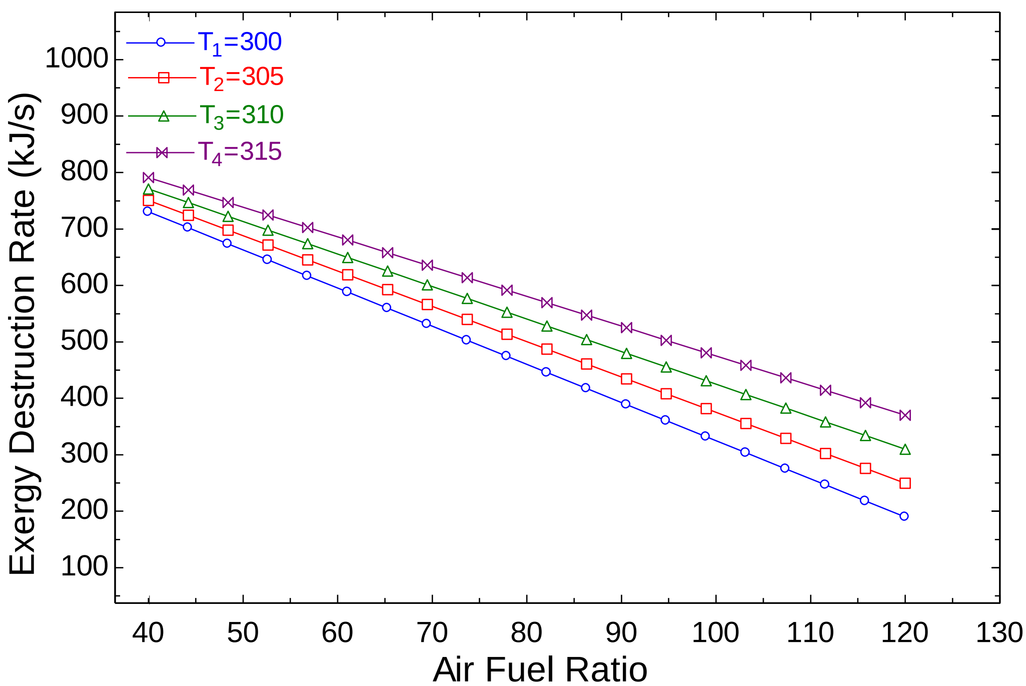


Fig. 6.8 Exergy Destruction rate of pump vs AFR at various inlet temp.

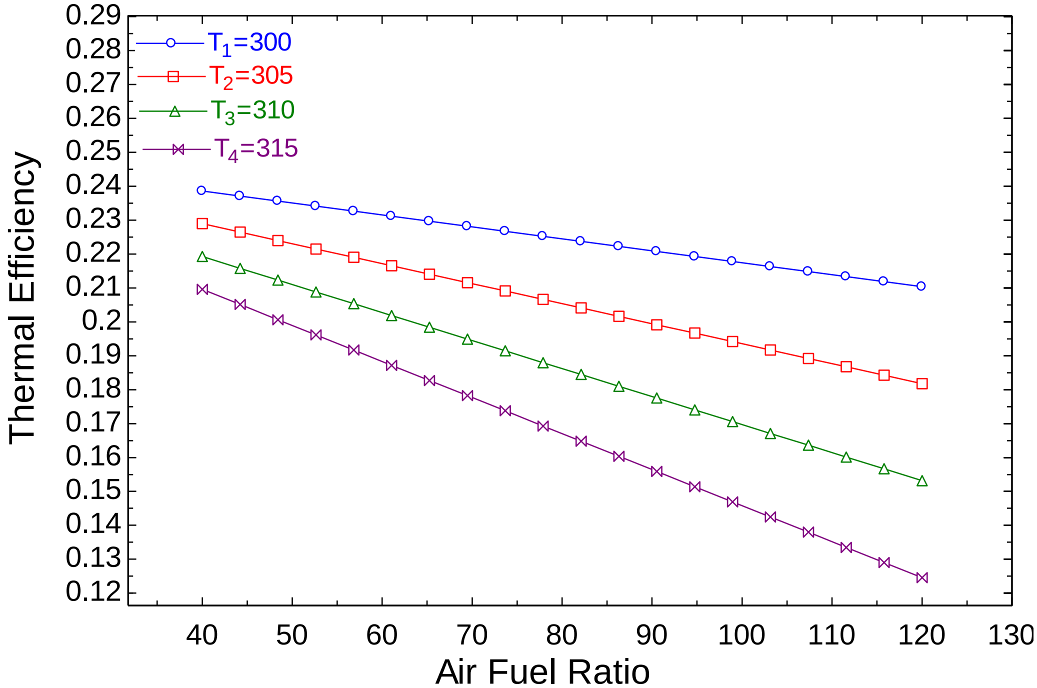


Fig. 6.9 Thermal efficiency of GT cycle vs AFR at various inlet temp.

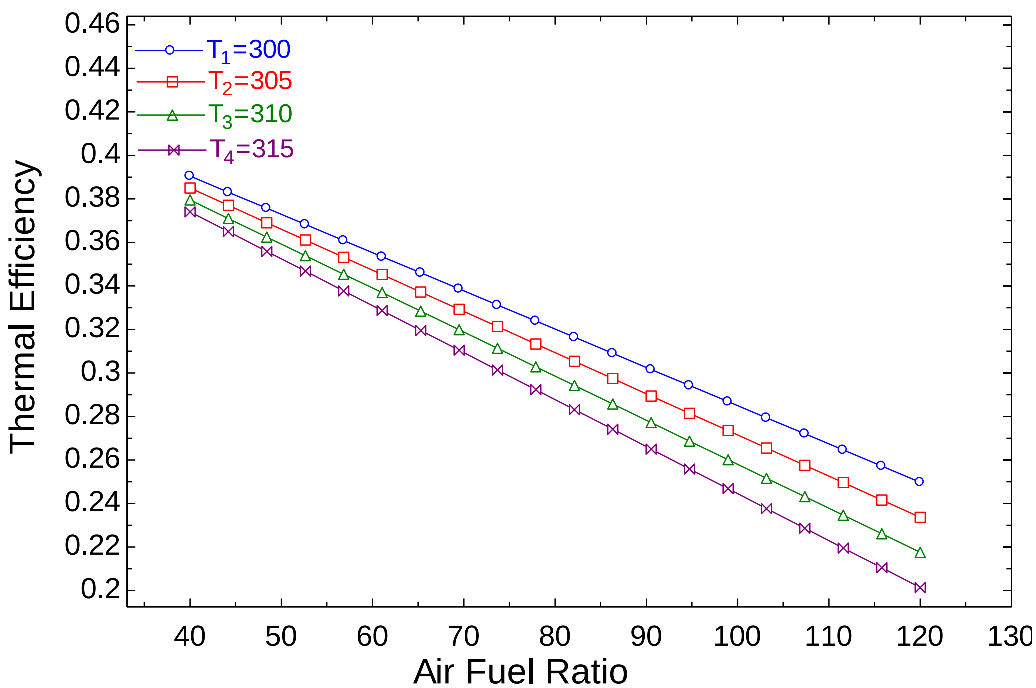


Fig. 6.10 Thermal efficiency of CVPC vs AFR at various inlet temp.

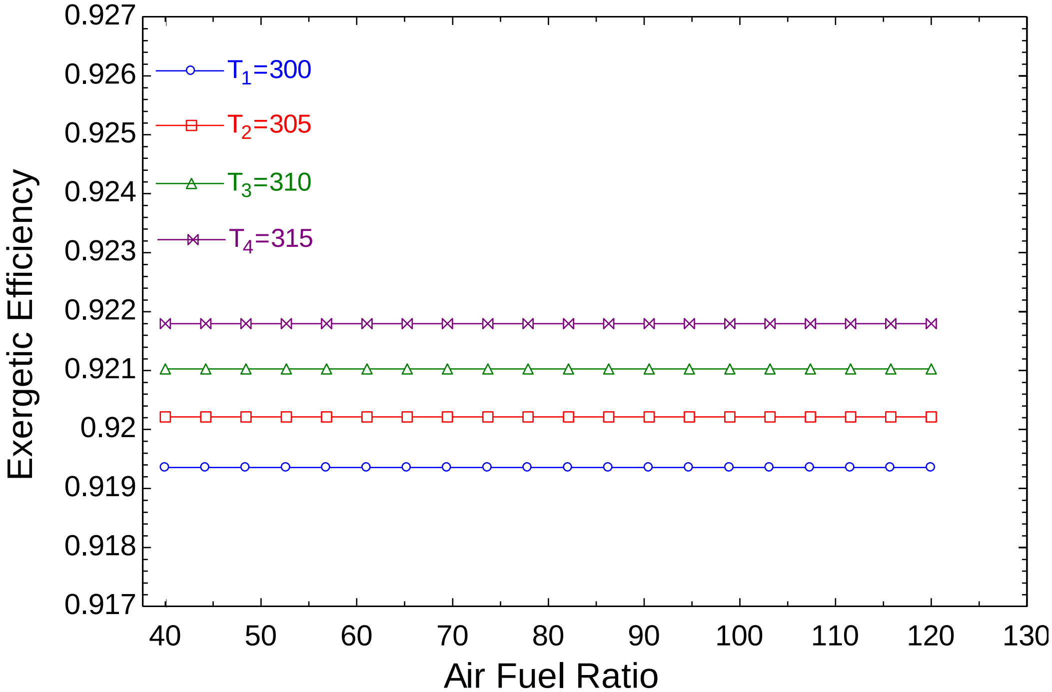


Fig. 6.11 Exergetic efficiency of air compressor vs AFR at various Inlet temp.

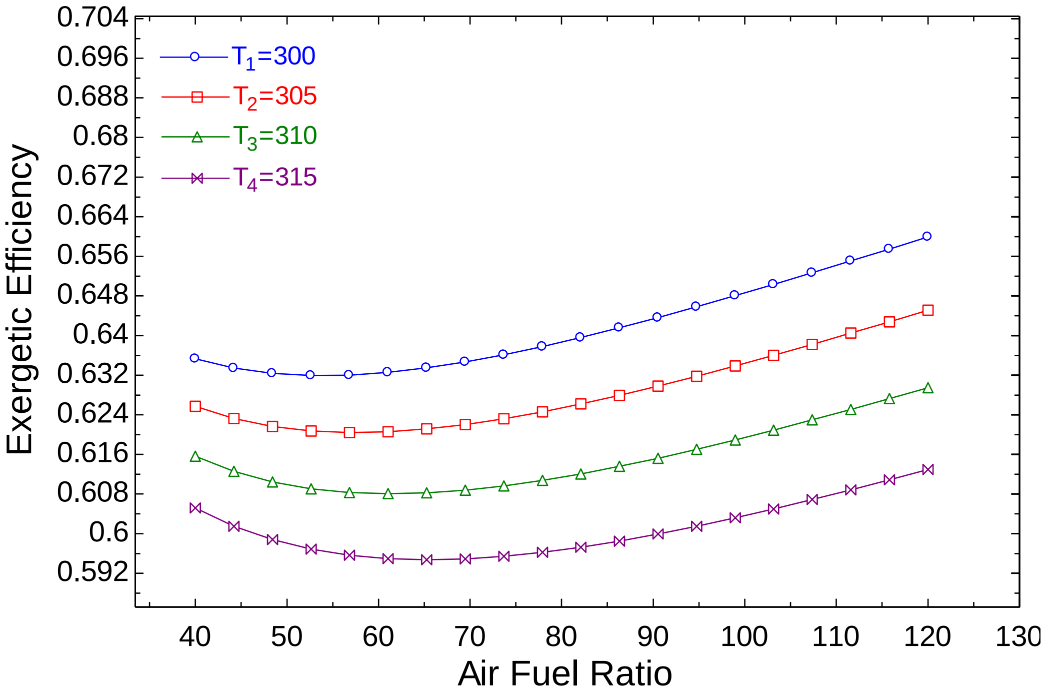


Fig. 6.12 Exergetic efficiency of combustion chamber vs AFR at various inlet temp.

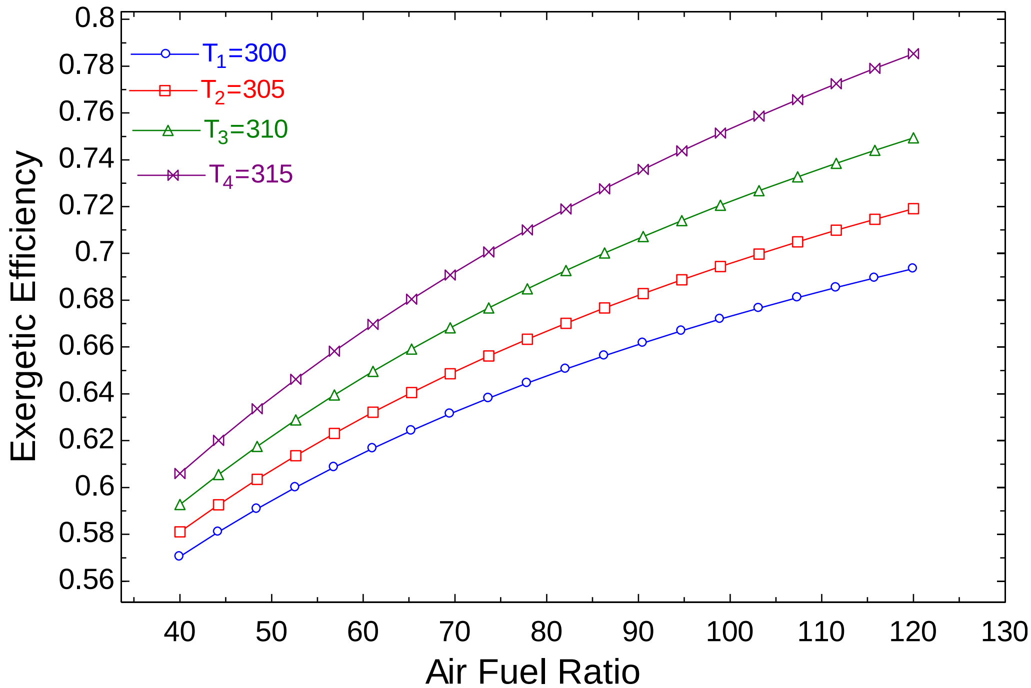


Fig. 6.13 Exergetic efficiency of HRSG vs AFR at various inlet temp.

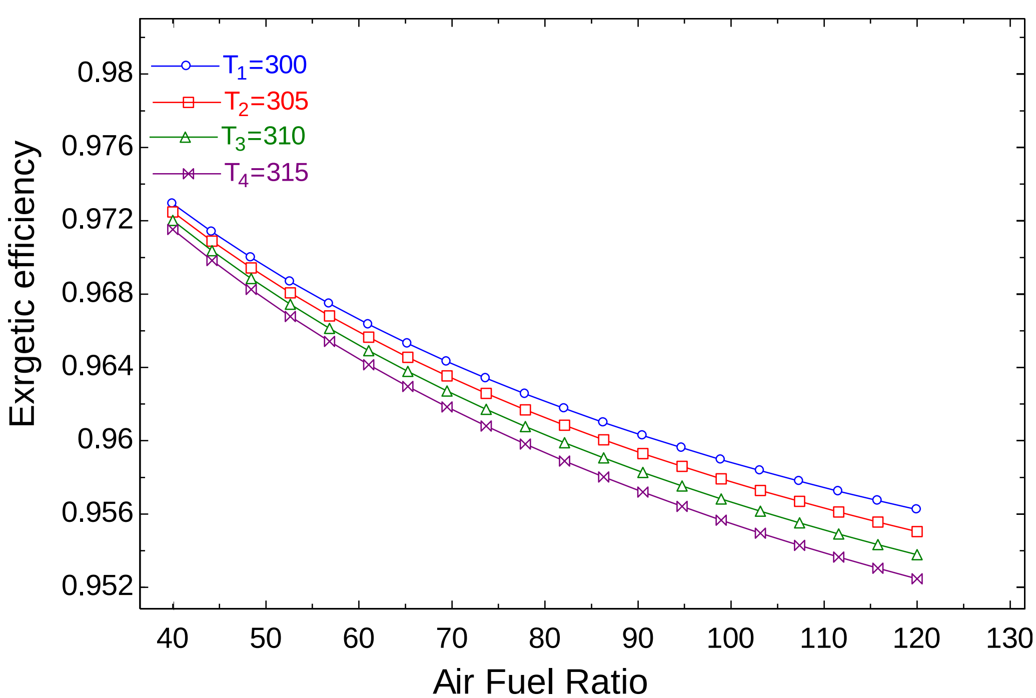


Fig. 6.14 Exergetic efficiency of gas turbine 1 vs AFR at various inlet temp.

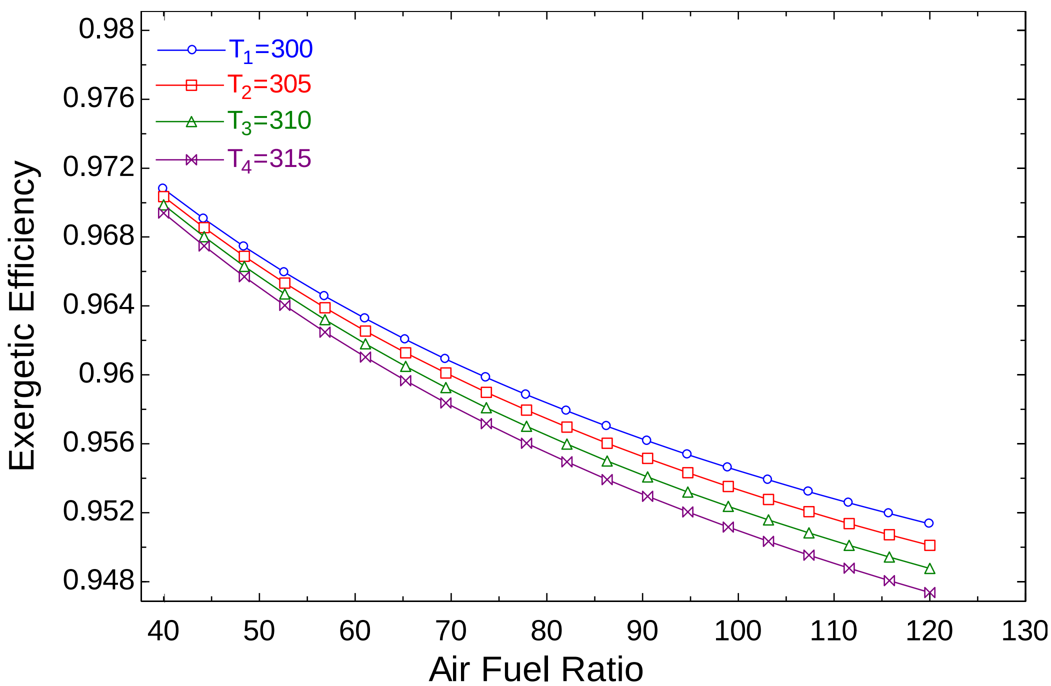


Fig. 6.15 Exergetic efficiency of gas turbine 2 vs AFR at various inlet temp.

CHAPTER 7

CONCLUSION

The purpose of the exergy assessment is to recognize the degrees and the zones of genuine energy losses, remembering the true objective to improve the present frameworks, methods or portions. A 2nd law investigation of a thermal PP has been performed in this venture report, close by a parametric survey that considers the effects of various parameters like inlet temp. and air fuel ratio on the framework performance. CVPC deal with the creating energy demand, and thus, we must pay heed to describe a strategy for the progression of these frameworks. Exergy investigation showed for a CVPC has given information on the irreversibilities of each procedure. While the major losses in the form of useful work occur at stack, examination of exergy loss in this system which is in the form of a perplex plant has proved that the foremost losses of exergy take place in CC, GT and in HRSG. Several approaches are anticipated to extend profitability of these plants i.e. higher temp. in GT, using noteworthy super mixes to fabricate hot parts, compressor inlet air cooling and an improvement methodology followed in HRSG working parameters.

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APPENDICES

Appendix 1: Gas Turbine Parameters

PARAMETERS	VALUE
Ambient air pressure [bar]	1.013
Ambient air temp. [°C]	290-315
Turbine isentropic efficiency [%]	90
Compressor isentropic efficiency [%]	88
Compression ratio [-]	4
Air Fuel Ratio [-]	50-130
Lower heat value of the fuel [kJkg-1]	43500
Heat Capacity Ratio for air [γa] [49]	1.4
Heat Capacity Ratio for gas [γg] [49]	1.33

Appendix 2: HRSG Parameters

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

Appendix 3: Steam Turbine Parameters

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

Appendix 4: Thermodynamic Properties of Steam

t, °C	1.0 MPa (t_S = 179.878 ˚C)			
	v	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	10000	000.98	-0.00009
5	0.999 59	1000.1	22.01	0.076 24
10	0.999 87	1000.3	42.99	0.151 00
15	1.000 48	999.52	63.94	0.224 31
20	1.001 38	998.62	84.85	0.296 28
25	1.002 55	997.45	105.75	0.366 97
30	1.003 97	996.05	126.64	0.436 45
35	1.005 60	994.43	147.53	0.504 78
40	1.007 44	992.61	168.41	0.572 02
45	1.009 48	990.61	189.30	0.638 19
50	1.011 71	988.43	210.19	0.703 35
55	1.014 11	986.09	231.09	0.767 53
60	1.016 69	983.59	252.00	0.830 77
65	1.019 43	980.95	272.92	0.893 10
70	1.022 33	978.16	293.86	0.954 55
75	1.025 39	975.24	314.81	1.0152
80	1.028 60	972.19	335.77	1.0750
85	1.031 97	969.02	356.75	1.1340
90	1.035 50	965.72	377.76	1.1922
95	1.039 17	962.30	398.79	1.2497
100	1.043 00	958.77	419.84	1.3065
105	1.046 99	955.12	440.92	1.3626
110	1.051 12	951.36	462.04	1.4181
115	1.055 42	947.49	483.19	1.4729
120	1.059 87	943.51	504.38	1.5272
125	1.064 49	939.42	525.60	1.5808
130	1.069 27	935.21	546.88	1.6339
135	1.074 23	930.90	568.20	1.6865
140	1.079 35	926.48	589.58	1.7386
145	1.084 66	921.95	611.01	1.7901
150	1.090 15	917.31	632.50	1.8412
155	1.095 83	912.55	654.06	1.8919
160	1.101 71	907.68	675.70	1.9421
165	1.107 80	902.69	697.41	1.9919
170	1.114 10	897.58	719.20	2.0414
175	1.120 63	892.35	741.08	2.0905
180	194.44	5.1431	2777.4	6.5857

185	197.42	5.0653	2790.7	6.6148
190	200.34	4.9916	2803.5	6.6427
195	203.20	4.9212	2816.0	6.6695
200	206.02	4.8539	2828.3	6.6955
210	211.5	4.7268	2852.2	6.7456
220	216.98	4.6087	2875.5	6.7934
230	222.31	4.4983	2898.4	6.8393
240	227.56	4.3944	2920.9	6.8836
250	232.75	4.2965	2943.1	6.9265
260	237.88	4.2038	2965.1	6.9681
270	242.96	4.1159	2986.9	7.0087
280	248.01	4.0322	3008.6	7.0482
290	253.01	3.9524	3030.2	7.0868

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	0.9527	7376.8	10.489

Appendix 4: Water Properties

P (MPa)	T (^{o}C)	Density (kg/m^3)		Enthalpy (kJ/kg)		Entropy (kJ/kg-k)		Volume (cm^3/g)	
		ρ_L	ρ_v	h_L	h_v	s_L	s_v	v_L	v_v
0.0070	39.0	992.5	0.04872	163.3	25717	0.55903	8.274	1.0075	20524.