EFFICIENCY IMPROVEMENT OF COMBINED CYCLE POWER PLANT USING TRIPPLE PRESSURE HRSG

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ASHISH THOMAS

Roll. No.: 2K13/THE/28

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Under the Guidance of

Prof. R. S. Mishra & Dr. Akhilesh Arora

DEPARTMENT OF MECHANICAL ENGINEERING

DELHI TECHNOLOGICAL UNIVERSITY

(FORMERLY DELHI COLLGE OF ENGINEERING)

CANDIDATE'S DECLARATION

I hereby declare that the work which being presented in the major thesis entitled "**EFFICIENCY IMPROVEMENT OF COMBINED CYCLE POWER PLANT USING TRIPPLE PRESSURE HRSG**" in the partial fulfilment for the award of the degree of Master of Technology in "THERMAL ENGINEERING" submitted to Delhi Technological University (Formerly Delhi College of Engineering), is an authentic record of my own work carried out under the supervision of Dr. Akhilesh Arora & Prof. R.S. Mishra, Department of Mechanical Engineering, Delhi Technological University (Formerly Delhi College of Engineering). I have not submitted the matter of this dissertation for the award of any other Degree or Diploma or any other purpose what so ever. I confirm that I have read and understood 'Plagiarism policy of DTU'. I have not committed plagiarism when completing the attached piece of work, similarity found after checking is 8% which is below the permitted limit of 20%.

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****

CERTIFICATE

This is to certify that the above statement made by ASHISH THOMAS is true to the best of my knowledge and belief.

DR. AKHILESH ARORA
Assistant Professor
Department of Mechanical Engineering
Delhi Technological University
(Formerly Delhi College of Engineering)
Delhi -110042

Prof. R. S. MISHRA
Professor
Department of Mechanical Engineering
Delhi Technological University
(Formerly Delhi College of Engineering)
Delhi -110042

ABSTRACT

By combining a Brayton cycle used as topping cycle and a Rankine cycle Bottoming it we can achieve a solution that allow efficiency and power of installation. Such an arrangement when used to produce power by electric generator is called combined cycle power plant (CCPP). It utilizes the gas turbine for Brayton cycle and steam turbine for Rankine cycle with air and water as working fluids to achieve efficient, reliable, and economic power generation. Currently overall thermal efficiencies up to 60% are confirmed by the foremost manufacturers from the sector as state of the art and special modifications have been proposed to improve the overall thermal efficiencies more than 60%.

The exhaust gas temperature entering the HRSG is in the range of 600 °C and gradually reduces to 150 °C or less as it flows over the various heat transfer sections. For Rankine cycles to be more efficient the steam pressure has to be high. Currently many of the steam turbines in Combined Cycle plants operate at 140 bar - the High Pressure (HP) circuit. At pressures of 140 bar the saturation temperature will be above 335 °C. The gas exiting the evaporative section will be in the order of 335 °C. This means absorption of the balance heat of the flue gas has to be in the economizer which is not feasible.

The high temperature exhaust is let to atmosphere and energy is wasted hence to absorb more heat two additional evaporators are used. One operating at lower pressure 40 bar - Intermediate Pressure (IP) circuit with gas exiting at 255 °C. The second one at 7 bar - Low Pressure (LP) circuit with gas exiting at 170 °C is used. This triple pressure HRSG together with a combination of Re-heated Rankine cycle help improve cycle Rankine efficiency together with absorbing more heat.

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I am using this opportunity to express my gratitude to everyone who supported me throughout the course of this project. I am thankful for their aspiring guidance, invaluably constructive criticism and friendly advice during the project work. I am sincerely grateful to them for sharing their truthful and illuminating views on a number of issues related to the project.

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I would also like to take this opportunity to present my sincere regards to **Dr. R. S. MISHRA**, Head of Department, Mechanical Department, Delhi Technological University (Formerly Delhi College of Engineering), for his kind support and encouragement to help me understand heat transfer from the basic which developed to have clarity of the subject and to solve the complex thermal calculation required for the combined cycle power plants.

I am thankful to my friends and classmates for their unconditional support and motivation for this dissertation.

Thank you,

Ashish Thomas

2K13/THE/28

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NOMENCLATURE

CCPP	Combined Cycle Power Plant
Cp	Specific Heat (kJ/kg)
CRH	Cold Re-Heat (Exhaust Steam from HP turbine to Re-heater)
EVAP	Evaporator
ECON	Economiser
GEN	Generator
GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
HP	High Pressure
HRH	Hot Re-heat (Re-heated Steam from Re-heater to LP Turbine)
h	Specific Enthalpy (kJ/kg)
IP	Intermediate Pressure
LHV	Lower Heating Value of Fuel (kJ/kg)
LP	Low Pressure
m	Mass flow rate (kg/s)
P	Pressure (bara)
Q	Heat Exchange per unit mass (kW)
SPH	Superheater
ST	Steam Turbine
s	Specific Entropy (kJ/kgK)
T	Temperature (^{o}C)
TG1	Exhaust Gas temperature inlet to HRSG (^{o}C)
TG2	Exhaust Gas temperature inlet to Evaporator (^{o}C)

TG3	Exhaust Gas temperature inlet to Economizer (^oC)
TG4	Exhaust Gas temperature outlet from HRSG (^oC)
Ts1	Superheated steam temperature from superheater (^oC)
Ts	Saturated temperature of steam in evaporator (^oC)
Tw1	Feed water inlet temperature to HRSG (^oC)
Tw2	Feed water inlet temperature to Evaporator (^oC)
W	Specific Work (kJ/kg)
η	Isentropic Efficiency (non dimensional)

CHAPTER 1

1. INTRODUCTION

Electrical energy has become one of the fundamental requirements to exist in a modern society and it is the base on which we keep our homes, business, transportation etc. running smoothly and because of this pervasive need of electricity in our society there is a high interest in the technology of electric power generation. The electric power generation has evolved during the last century and these developments has been driven by economics, technological advancements and government regulations.

As per shift project data the total world electric power production is classified into Coal, Gas, hydro, Nuclear, Oil and other Renewable Energy Sources. Coal being the largest contributor serves to around 38% of power generation while Gas being second with 22%, Hydro 17%, Nuclear 11%, Oil 5% and other 7%.

Since electricity must be produced at lowest cost to keep accessibility together with the availability year around the most economical fuel turns outs to be coal or gas. One can argue for wind, solar, geothermal, ocean or other renewable since these are free fuels but these technology is not available in bulk also the due to continuous power generation from these sources are not possible. Wind and Solar both requires huge area or land per kw power generated. Nuclear is the third most economical fuel since it can generate huge energy relative to the cost of fuel but operating and maintaining a nuclear power plant is much expensive also due to risk and cost of disposing used radioactive fuel makes it less lucrative.

Coal power plants also has its impacts like environmental degradation, Global warming potential, mercury-polluted water, acid rain, coal mining deaths, and so forth but today coal plant owners mostly only pay for equipment to reduce acid rain. They don’t have to pay for past pollution, or emitted airborne particulate matter.

The coal based power plant works on the Rankine Cycle principle and its efficiency is around 42% while the Gas based power plant while operating in simple Brayton cycle has an efficiency of 35% the details of each cycle is explained in forthcoming pages.

1.1. RANKINE CYCLE

The Rankine cycle is the process on which steam-operated heat engines usually found in coal based thermal power generation plants operate. The fuels used in these power plants are usually fossil fuels such as coal, natural gas, and oil. If used in combined cycle the Boiler becomes Heat Recovery Steam Generation and recovers heat from exhaust flue gases.

The efficiency of the Rankine cycle is limited due to the high heat of vaporization of the working fluid which is water unless the pressure and temperature reach supercritical values. The Supercritical pressure of working fluid is at 221 bar which is quite high pressure. The actual overall thermal efficiency of a modern coal-fired power station is up to 42%. The low temperature of Rankine cycle helps it to be run as a bottoming cycle with a Brayton cycle which operates at much higher temperature and has an exhaust temperature in range of 600 ^{o}C.

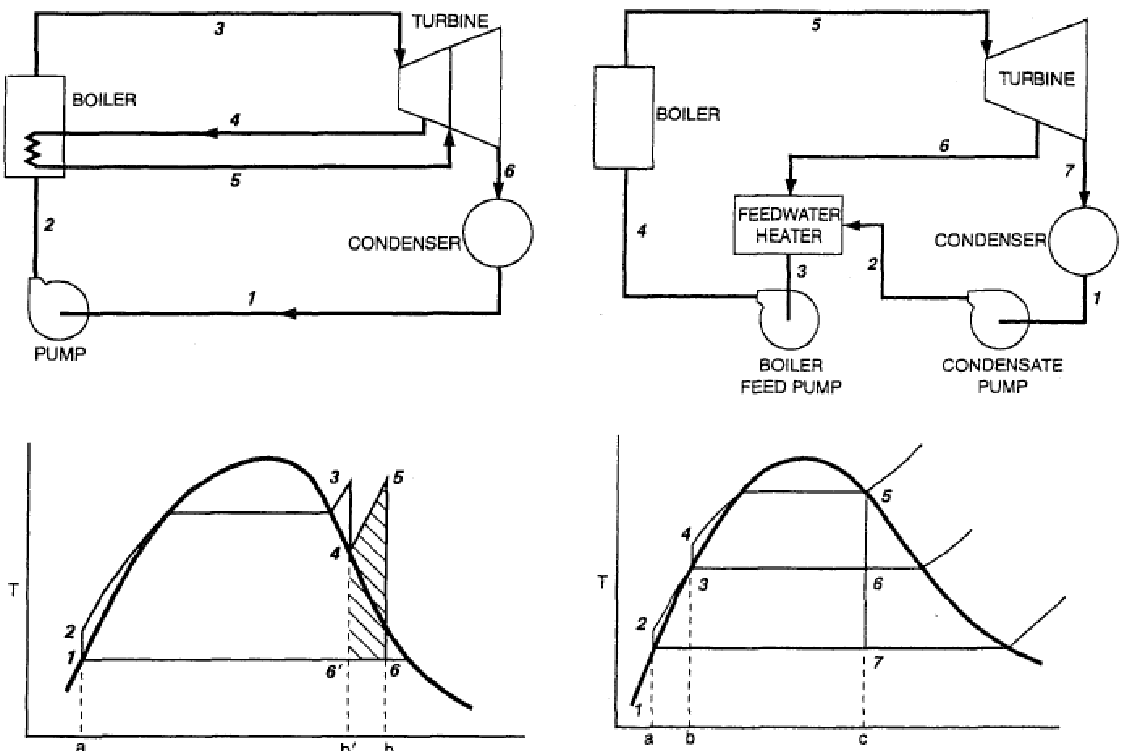


Fig. 1. T-s Diagram of Rankine Cycle

1.2. BRAYTON CYCLE

The Brayton cycle simulates the operation of a gas turbine engine. This cycle consists of 4 processes, as shown in PV & TS diagram alongside a sketch of an engine:

An Ideal Brayton cycle Consist of:

Process 1 to 2: Atmospheric air is drawn into the compressors stages, where it is compressed (Isentropic).

Process 2 to 3: The compressed air then flows through a combustion chamber, where fuel is fired which results in heating that air this is a constant-pressure process (Isobaric), as the combustion chamber is open to flow in and out.

Process 2 to 3: The heated, compressed air then imparts its energy, expanding through a turbine (Isentropic). where the work extracted by the turbine and a part of which is also used to drive the compressor.

Process 3 to 4: Heat rejection (to Atmosphere or HRSG) is also at constant pressure (Isobaric).

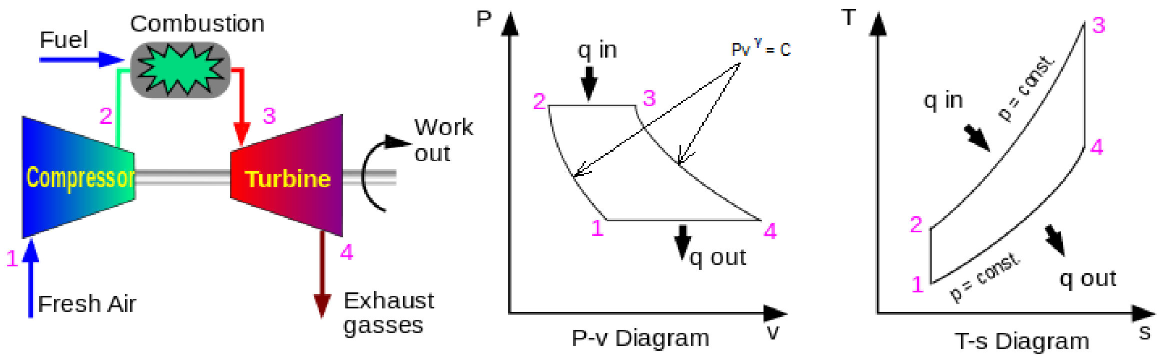


Fig. 2. Schematic, P-v & T-s Diagrams of Brayton Cycle

1.3. COMBINED CYCLE POWER PLANT

As mentioned earlier the efficiency of individual Rankine and Brayton Cycle is limited. With the existing technology, the most efficient energy conversion system is the Combined Cycle Power Plant (CCPP). It is the combination of a gas turbine cycle and steam cycle bottoming it, allows efficiency and power of installation. In a typical CCPP, the exhaust heat from the gas turbine is recovered in a heat recovery steam generator (HRSG). Normally when two cycles are combined, the cycle which is operating at the higher temperature level is called the topping cycle the waste heat produced by topping cycle is then used in second process which operates at a lower temperature level and is therefore called a bottoming cycle.

The Steam Turbine (ST) produces about 50% of power compared with the Gas Turbine (GT). The power product by the ST is without any additional fuel requirement, as the combustion happens only in the GT. As a consequence, the combined cycle efficiency is about 1.5 times the efficiency of a GT and at least 1.2 times the efficiency of an equivalent conventional steam plant.

Combined-cycle systems utilizing the Brayton Cycle gas turbine and the Rankine Cycle steam system with air and water as working fluids achieve efficient, reliable, and economic power generation. The Flexibility provided by these systems fulfills both utility-power generation and industrial-cogeneration applications where heat is required.

The cycle combines existing gas and steam technologies into a single unit, yielding significant improvements in thermal efficiency over conventional Rankine cycle power plants. In a CCPP plant the thermal efficiency can be extended above 60 per cent, by utilizing the exhaust gas from the gas turbine into a heat recovery steam generator rather than exhausting it to atmosphere.

However, the heat recovered in this process is sufficient to drive a steam turbine with an electrical power output of approximately 50 per cent of the gas turbine generator.

1.3.1 Working

The working of a combined-cycle plant to produce electricity and recover waste heat from the exhaust to increase cycle efficiency and electrical output.

1. Burning of Fuel.

The compressor of gas turbine compresses the air and mixes it with fuel which is fired to a very high temperature. The heated air-fuel mixture now moves through the gas turbine blades imparting the energy to gas turbine blades which makes it to spin

The fast-spinning turbine then in turn drives an electric generator which converts the mechanical energy into electricity.

2. Exhaust Heat Recovery with HRSG

A Heat Recovery Steam Generator (HRSG) recovers the exhaust heat from the gas turbine that would have otherwise escape through the exhaust stack to atmosphere.

The HRSG generates steam from these hot flue gas and delivers it to drive a steam turbine.

3. Additional Power Generation from Steam turbine.

The steam from HRSG is used to drive the steam turbine which drives a generator drive shaft, where additional electricity is produced from the exhaust gases.

4. The Exhaust of steam turbine is the condensed in a condenser and which provides condensate to be reused in the HRSG to produce steam. The Condenser acts as the heat sink here.

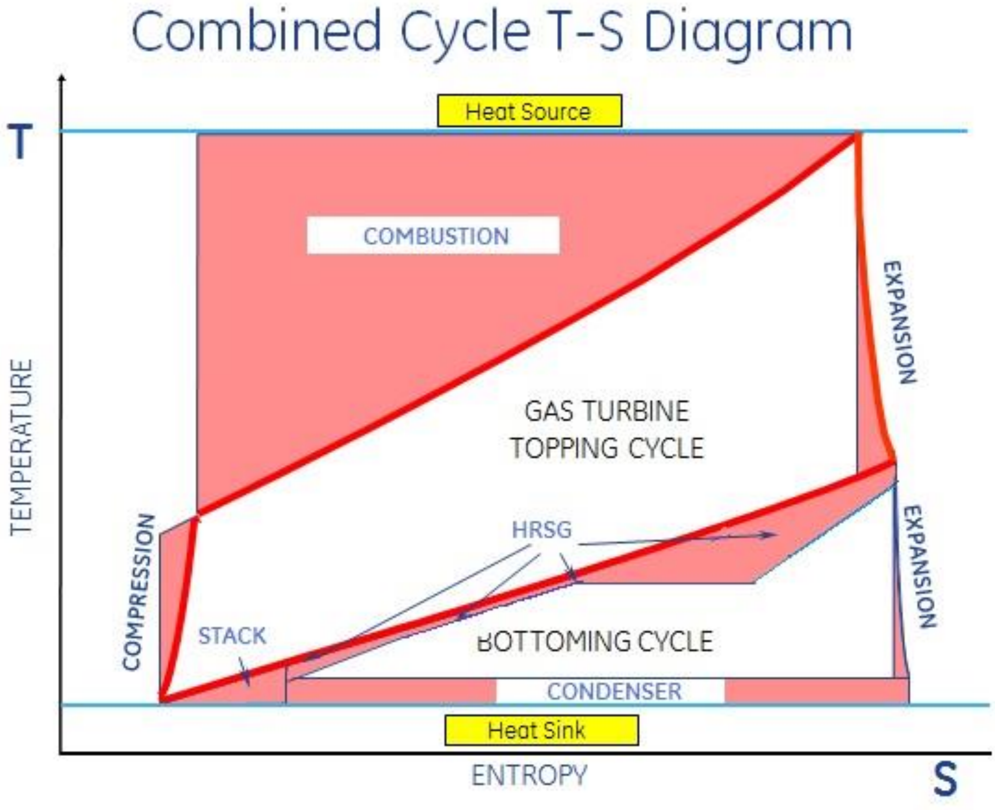
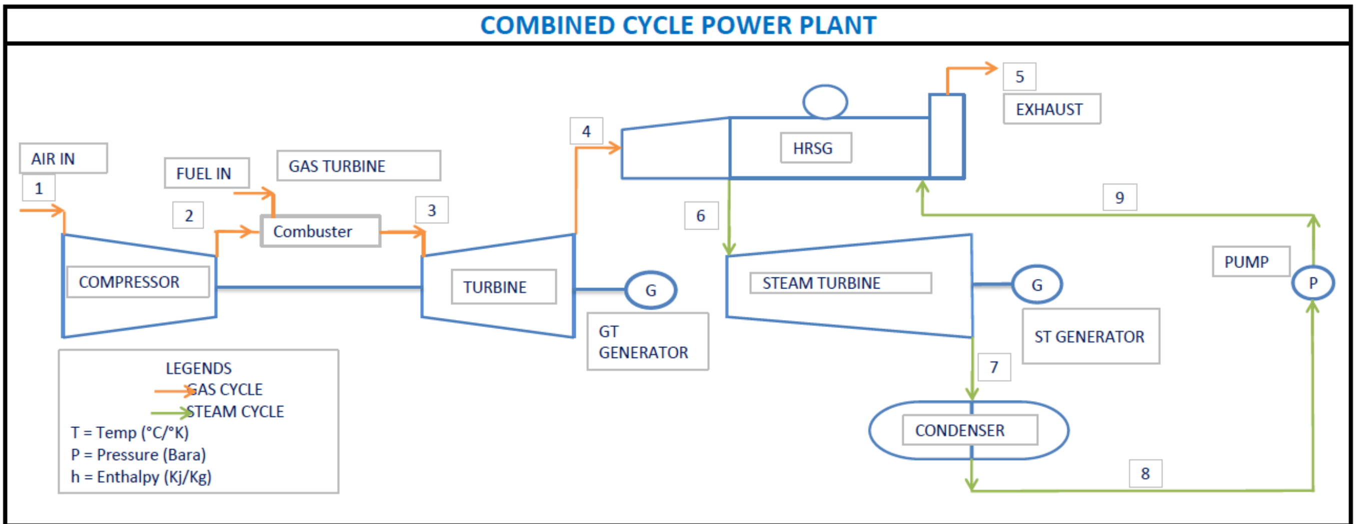


Fig. 3. Temperature Entropy diagram of combined cycle power plant



	BRAYTON CYCLE
1	AMBIENT AIR INLET TO GAS TURBINE
2	COMPRESSOR AIR OUTLET TO COMBUSTER
3	HIGH ENERGY HOT GAS FROM COMBUSTION CHAMBER
4	HIGH TEMPERATURE EXHAUST FROM GAS TURBINE
5	LOW TEMPERATURE EXHAUST TO ATMOSPHERE
	RANKINE CYCLE
6	HIGH PRESSURE & TEMPERATURE STEAM FROM HRSG TO STEAM TURBINE
7	LOW PRESSURE & TEMPERATURE STEAM FROM STEAM TURBINE TO CONDENSER
8	LOW PRESSURE & TEMPERATURE CONDENSATE FROM CONDENSER TO PUMP
9	HIGH PRESSURE CONDENSATE TO BOILER

ST	=	STEAM TURBINE
GT	=	GAS TURBINE
P	=	PUMP
G	=	GENERATOR

Fig. 4. Flow diagram of combined cycle power plant

1.3.2 Main Components

- ✓ Gas turbine
- ✓ Heat Recover Steam Generator (HRSG)
- ✓ Steam Turbine
- ✓ Condenser
- ✓ Condensate Pump
- ✓ Feed water Pump

1.4. SCOPE OF PRESENT WORK

The scope of present work is to maximize the efficiency of a Combined Cycle Power Plant where focus is kept on maximizing the heat extraction that can be possible from the gas turbine exhaust gases and thereby reducing the discharge gas temperature which is finally exhausted to the atmosphere. This is achieved by improving the heat recovery from HRSG using a triple pressure system and also together by improving the Rankine Cycle by optimizing it with a Re-heat Rankine cycle to converge on the maximum heat extraction that can be possible from the exhaust gases which is finally exhausted to the atmosphere.

CHAPTER 2

2. REVIEW OF LITERATURE

In this chapter, it is intended to give a brief literature review of work being carried out on combined cycle.

2.1. LITERATURE

Meeta [1]

has carried out analysis of Exergy analysis of dual pressure HRSG for different dead states and varying steam generation states in gas/steam combined cycle power plant. The conclusion shows In the LP and HP super heater sections of the HRSG, the magnitude of exergy losses are substantial but it reduces at higher dead state temperatures however the exergy efficiency shows minor variations. In LP and HP evaporator sections, the exergy losses increase for higher dead states and therefore exergy efficiency decreases. The results obtained are helpful in reducing the exergy destruction for enhancing HRSG performance which eventually improves the efficiency of combined cycle power plants. This study is also beneficial for finding quality of useful energy at different components of dual pressure HRSG for power plant designer

Dillip and Vijay [2]

has done analysis of a single pressure combined cycle and has identified the basic formulas which may be used for analyses. Its been concluded that there have significant effects on the performance of the CCGT with maximum cycle pressure, pressure ratio, maximum temperature and turbine inlet temperature.

Wenguo and Yingying [3]

has studied the influence of HRSG inlet gas temperature on the steam cycle efficiency. The result shows that increasing the HRSG inlet temperature has less improvement to steam cycle efficiency when it is over 590°C. Also Partial gas to gas recuperation in the topping cycle is studied. By Joining HRSG optimization with the use of gas to gas heat recuperation, the combined plant efficiency is claimed to rise up to 59.05% at base load.

Vinod [4]

has carried out presentation of how components of a combined cycle and co-generation plants are arranged with elaborated working principles and highlighting merits and de-merits of each components used also highlighting the developments in cogeneration-the process of producing useful heat and electricity together from the same fuel source has increased the efficiency of fuel burning from 30% to 90%, thereby reducing damage to the environment while increasing economic output through more efficient use of resources.

Nico et al [5]

has carried out the exergy analysis and by the evaluation of exergy losses in combined cycle plants shows that these losses are mainly dominated by the exergy losses of thermal combustion. Possibilities to reduce these losses are limited. The exergy flow diagrams (Figs. 11 and 12) show that the highest losses are caused by (thermal) combustion of the fuel. The further enhancement of overall power plant efficiencies to 70% or even higher will require the development of high temperature fuel cell systems like SOFC–GT hybrid systems

Abdulrahman et al [6]

has carried out the analysis of the inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia. An absorption chiller system can be used to boost power output continuously. However, the systems have high capital costs, and are complex to install and maintain. The high capital costs are only justified if the demand for CT power increase is continuous, and is not limited to peak times. The overall conclusion is that the high-water requirements of evaporative cooling and high-pressure fogging mean that they are not suitable for use in the desert climate of Saudi Arabia. Absorption cooling is an expensive system with limited cooling ability

during peak hours. Mechanical refrigerative air inlet cooling is simple and reliable, yet without thermal storage it has high parasitic power requirements. The preferred option, therefore, is refrigerative cooling with chilled water or ice thermal storage. Of the two, ice storage can produce lower inlet air temperatures and requires a smaller storage volume.

Xiaojun et al [7]

has proposed an integrated advanced thermal power system to improve the performance of the conventional combined cycle power plant. Both inlet air cooling and inter-cooling are utilized within the proposed system to limit the decrease of the air mass flow contained in the given volume flow as well as reduce the compression power required. The latent heat of spent steam from a steam turbine and the heat extracted from the air during the compression process are used to heat liquefied natural gas (LNG) and generate electrical energy.

Ameri and Hejazi [8]

This paper presents an overview of an intake air-cooling system that uses a steam absorption chiller and an air cooler to address the above problem in the Chabahar power plant. By using this technique, the output power will increase by 11.3%. The economic studies have shown the internal rate of return will be 23.4%.

Kaushika et al [9]

Has performed Exergy analysis in this article to be able to help understand the performance of coal fired, gas fired combined cycle thermal power plants and identify design possible efficiency improvements. It gives logical solution improving the power production opportunities in thermal power plants. By the exergy analysis we can conclude that main energy loss in boiler in coal based thermal power plant and combustion chamber in gas fired combined cycle thermal power plant.

Khaliq and Kaushik [10]

This paper has presented a methodology based on first and second law for the evaluation of thermodynamic performance, of gas turbine cogeneration systems with reheat. This methodology is quite simple and systematic. The most important aspect of the study is that performance evaluation of combustion gas turbine cogeneration based on first law alone is inadequate and more meaningful evaluation must include second-law analysis. The analysis of a certain case of combustion gas turbine cogeneration has proven the usefulness of the presented method for analyzing the energetic and exergic performance of cogeneration plant.

Thamir et al [11]

Has simulated modeling results which show that the compression ratio, air-to-fuel ratio, ambient temperature, and turbine inlet temperature have significant effects on the performance of the Combined cycle power plant and has demonstrated a maximum thermal efficiency of a combined cycle power plant at 57%. Also concludes that the overall thermal efficiency decreases and the power output increases linearly with any increases in the ambient temperature or air-to-fuel ratio.

Naimul et al [12]

Has simulated modeling results in MATLAB/Simulink which show that the exhaust temperature of the gas turbine is quite an important parameter which has to be maintained high for the bottoming rankine cycle to operate efficiently. The study shows rise in temperature is controlled by flow in the gas turbine and with a rise in ambient temperature the power output of the gas turbine falls. The power output of gas turbine can increased by cooling the inlet air to the compressors of gas turbine. Study results also show that when fuel firing to the gas turbine is increased to increase the output then the mass flow of air also needs to be adjusted accordingly otherwise the temperature may uncontrollable rise and may cross the threshold values.

Polyzakis et al [13]

This work contains an optimization analysis of four potential GT cycles, namely single cycle (SC), intercooled cycle (IC), reheated cycle (RH) and intercooled and reheated cycle (IC/RH). The design point and off design point performances of the four cycles alone or in combination with the steam cycle have been investigated. The three options, SC, IC and RH perform very similarly OD. The thermal efficiencies and specific power outputs are not the optimum values they would be for GT cycle operation. Furthermore, the IC/RH engine was run differently from the IC one. When the ambient temperature changes, it is prudent to assume that the intercooler exit temperature changes simultaneously. Based on this analysis the optimum gas turbine cycle to operate in a combined cycle power plant came out to be the reheated cycle.

Firdaus et al [14]

In this paper, the effect of ambient temperature on the performance of the MGT -CGS was investigated by considering its application under various ambient temperature conditions in cold region and the following results were obtained. It was found that when ambient temperature increased, electrical efficiency decreased but exhaust heat recovery increased. It was also found that when ambient temperature increased, and that performance of heating cycle was more efficient than cooling cycle for the AHE. The exhaust heat to power ratioQexe/Pehad a similar behaviorwith exhaust heat recovery to power ratio Qehr/Pe. Qexe/Pe and Qehr/Pe increased when ambient temperature increased. From the clarification of the MGT-CGS performance under various temperature conditions, the controlling of heat and electrical output of the CGS could be applied. If heat and electrical output of the MGT-CGS and heat and electrical demand of a facility is known, one could decide whether inlet precooling is required or not.

Xinguo et al [15]

The paper shows comparison of organic Rankine cycle (ORC), an organic Rankine cycle with ejector (EORC) and a and a double organic Rankine cycle (DORC). The power output is higher in the EORC and DORC compared to the ORC. The power output and thermal efficiency of the DORC is superior to the EORC, but another expander-generator and its auxiliary equipment is required for the DORC led to the increase of the investment and operation management compared to the EORC. At the maximum net power output, the most exergy losses took place in the evaporator, condenser and the exhaust water, while it is the smallest in the ejector in the EORC cycle.

2.2. CONCLUSION AND GAPS

With the existing technology, the most efficient energy conversion system is the Combined Cycle Power Plant CCPP.

Based on literature review, it can be concluded that researchers have gone through detailed investigation on the combined cycle performance by first law analysis of energy efficiency and exergetic efficiency by second law analysis of single, dual pressure and triple pressure HRSG systems.

From the literature review its clear that efficiency of the individual cycles can be in improved as mentioned below

BRAYTON CYCLE

The efficiency of a Brayton engine can be improved in the following manners:

- ✓ Increasing pressure ratio
- ✓ Reduce the inlet air temperature to compressor
- ✓ Increase the firing temperature
- ✓ Reheating

RANKINE CYCLE

The efficiency of a Rankine cycle can be improved in the following manners:

- ✓ Increasing Inlet Pressure to turbine
- ✓ Increasing Inlet Temperature to turbine
- ✓ Reducing Condensing Pressure
- ✓ Reheating
- ✓ Regeneration

2.3. PROBLEM FORMULATION

In the present study, performance evaluation of combined cycle power with a GE 9HA Gas turbine is carried out using multiple HRSG combinations to find the most efficient combination and to reduce the effect of Stack Loss on Combined Cycle Efficiency

It is proposed to examine the effect of following parameters on the performance of combined cycle power plant.

1. Effect of exhaust stack temperature
2. Effect of steam generation pressure (single dual and triple)
3. Effect of Condensing pressure

CHAPTER 3

3. THERMODYNAMIC ANALYSIS

In this chapter, it is intended to give a thermodynamic analysis of present work, which has been carried out on the heat recovery from HRSG.

2.1. SYSTEM DESCRIPTION

- ✓ **SINGLE PRESSURE HRSG**

The exhaust gas from Gas Turbine post driving the turbine section is passed through a duct to reach the HRSG and the HRSG uses this exhaust heat to generate steam. In a single pressure HRSG there are 3 different sections namely the economizer, evaporator and the superheater.

The first section that the exhaust gas comes in contact is the superheater, this is arranged based on countercurrent heat transfer arrangement to utilize maximum amount of heat from the exhaust gases. The last section through which the exhaust gases pass is the economizer, here the feed water is heated to near about saturation temperature so that it can be passed on to the evaporator section where steam can be generated. The evaporator section consists of a steam drum which separates the steam from the boiling water from where saturated steam is passed on to the super heater so is can superheated and used in the steam turbine to generate electricity.

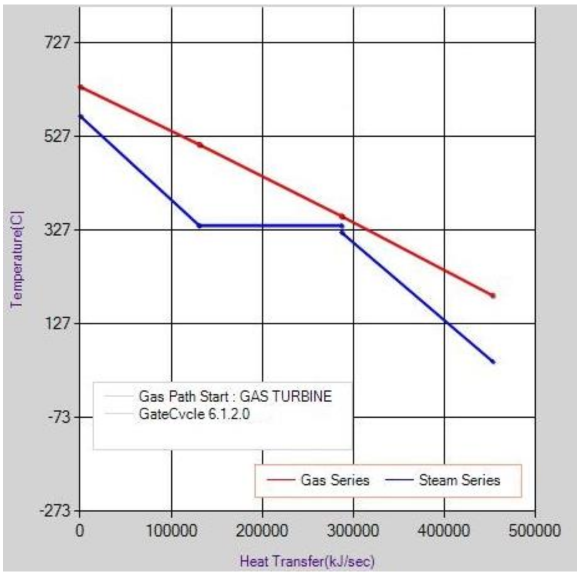


Fig. 5. Heat Release Plot of a Single Pressure HRSG

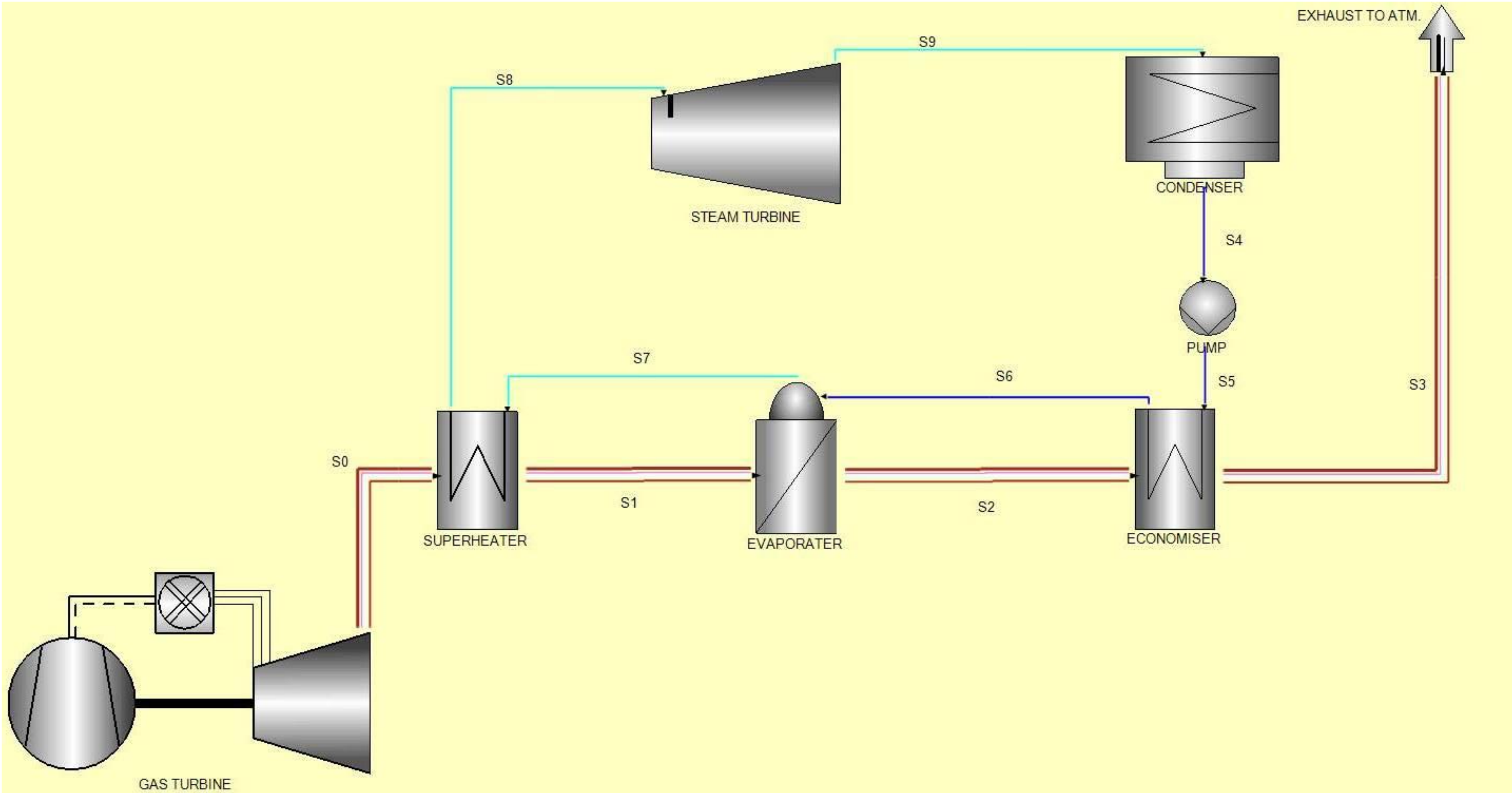


Fig 6. Single Pressure HRSG Flow Diagram

- ✓ **DUAL PRESSURE HRSG**

For Dual Pressure HRSG also exhaust gas from Gas Turbine post driving the turbine section is passed through a duct to reach the HRSG and the HRSG uses this exhaust heat to generate steam. The addition change happens is that steam is generated at 2 different pressures. In a dual pressure HRSG there are 6 minimum different sections namely they are

1. HP Superheater
2. HP Evaporator
3. HP Economizer
4. LP Superheater
5. LP Evaporator
6. LP Economizer

The Flue Gases passes through these sections in ascending order the first section being HP superheater where high temperature steam is generated and is heated by highest temperature exhaust gases. The last section through which the exhaust gases pass is the LP economizer, here the feed water is heated to near about saturation temperature so that it can be passed on to the LP evaporator section where LP steam is generated. The evaporator section consists of a steam drum which separates the LP steam from the boiling water from where saturated LP steam is passed on to the LP super heater so is can superheated, this super-heated steam injected to the steam turbine as an LP Steam Injection.

The HP train just remains the same as in a single pressure HRSG. The HP feed water is passed to the HP economizer, where this feed water is heated to near about saturation temperature and passed on to the HP evaporator section where HP steam is generated. The evaporator section consists of a similar steam drum only operating at a higher pressure which separates the HP steam from the boiling water and this saturated HP steam is the passed on to the HP super heater finally super-heated HP steam is fed to the inlet of steam turbine. Post working in steam turbine the exhaust steam is condensed in the condenser where its condensed to feed water. This feed water is circulated by LP pumps to the LP economizer and a portion of feed water is again pressured to High pressure to be sent to HP economizer.

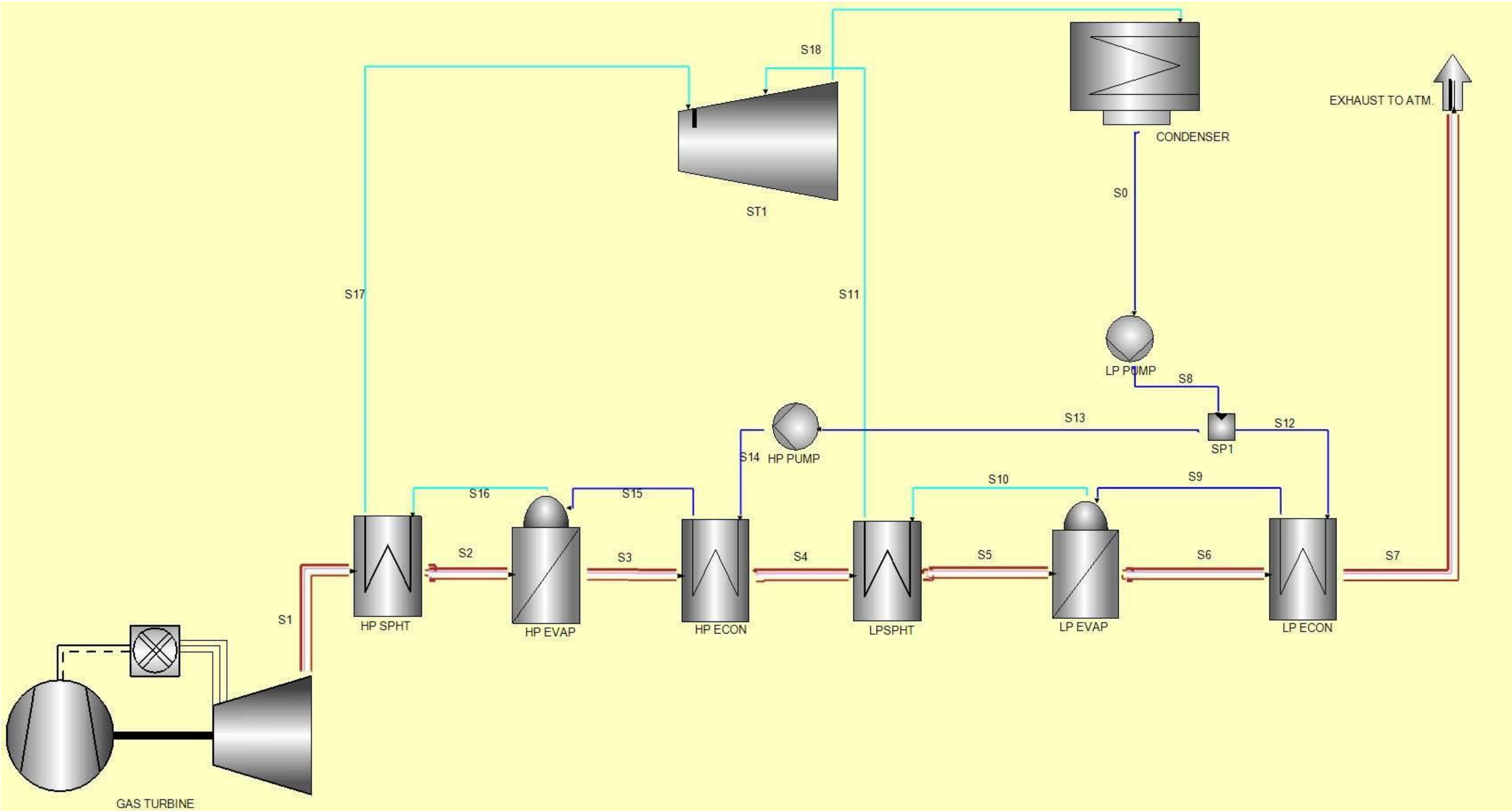


Fig. 7. Dual Pressure HRSG Flow Diagram

- ✓ **TRIPPLE PRESSURE HRSG WITH REHEATING**

For TRIPPLE Pressure HRSG also exhaust gas from Gas Turbine post driving the turbine section is passed through a duct to reach the HRSG and the HRSG uses this exhaust heat to generate steam. The change happens here is that steam is generated at 3 different pressures. In a dual pressure HRSG there are 6 minimum different sections namely they are

1. HP Superheater
2. Re-Heater
3. HP Evaporator
4. HP Economizer 3
5. IP Superheater
6. LP Superheater
7. HP Economizer 2
8. IP Evaporator
9. IP Economizer
10. HP Economizer 1
11. LP Evaporator
12. LP Economizer
13. Condensate Preheater

The Flue Gases passes through these sections in ascending order the first section being HP superheater & Re-heater where high temperature steam is generated and is heated by highest temperature exhaust gases. The last section through which the exhaust gases pass is the Condensate Preheater which preheats the condensate to bring it to LP economizer temperature, here the feed water is heated to near about saturation temperature so that it can be passed on to the LP evaporator section where LP steam is generated. LP steam is then passed on to the LP super heater so is can superheated, this super-heated steam injected to the steam turbine as an LP Steam Injection in same manner as it was done in the dual pressure HRSG.

The additional IP train is added with a IP economizer, Evaporator and superheater. The IP steam generated here is mixed with cold re-heat coming from the HP steam turbine and then passed on to reheater to heat the steam which is then sent to inlet of LP steam turbine.

The HP train just remains the same as in a dual pressure HRSG only change is that the economizers are divided in 3 sections to ensure proper heat distributed. The HP feed water is passed to the HP economizers, where this feed water is heated to near about saturation temperature and passed on to the HP evaporator section where HP steam is generated. The evaporator section consists of a similar steam drum only operating at a higher pressure which separates the HP steam from the boiling water and this saturated HP steam is the passed on to the HP super heater finally super-heated HP steam is fed to the inlet of HP steam turbine.

Post working in steam turbine the exhaust steam from HP turbine is combined with IP steam from HRSG and is re-heated before sending it to the LP Turbine where it is expanded to the condensing pressure. The LP turbine has an injection for LP steam from HRSG also. The Condenser condenses the near saturated steam from turbine and is collected in the hotwell of condenser

The LP condensate pumps circulates this condensate from condenser which is then fed to the condensate preheater from there a portion is sent LP economizer and continues to produce LP steam which will be injected in the LP steam turbine and a portion of feed water is again pressured by an IP pump to be sent to IP economizer which will finally generate the IP steam to be mixed with cold reheat from HP turbine and another portion is pressurized with a High pressure pump to be sent to HP economizer which will produce the HP steam which is to be fed to the HP turbine.

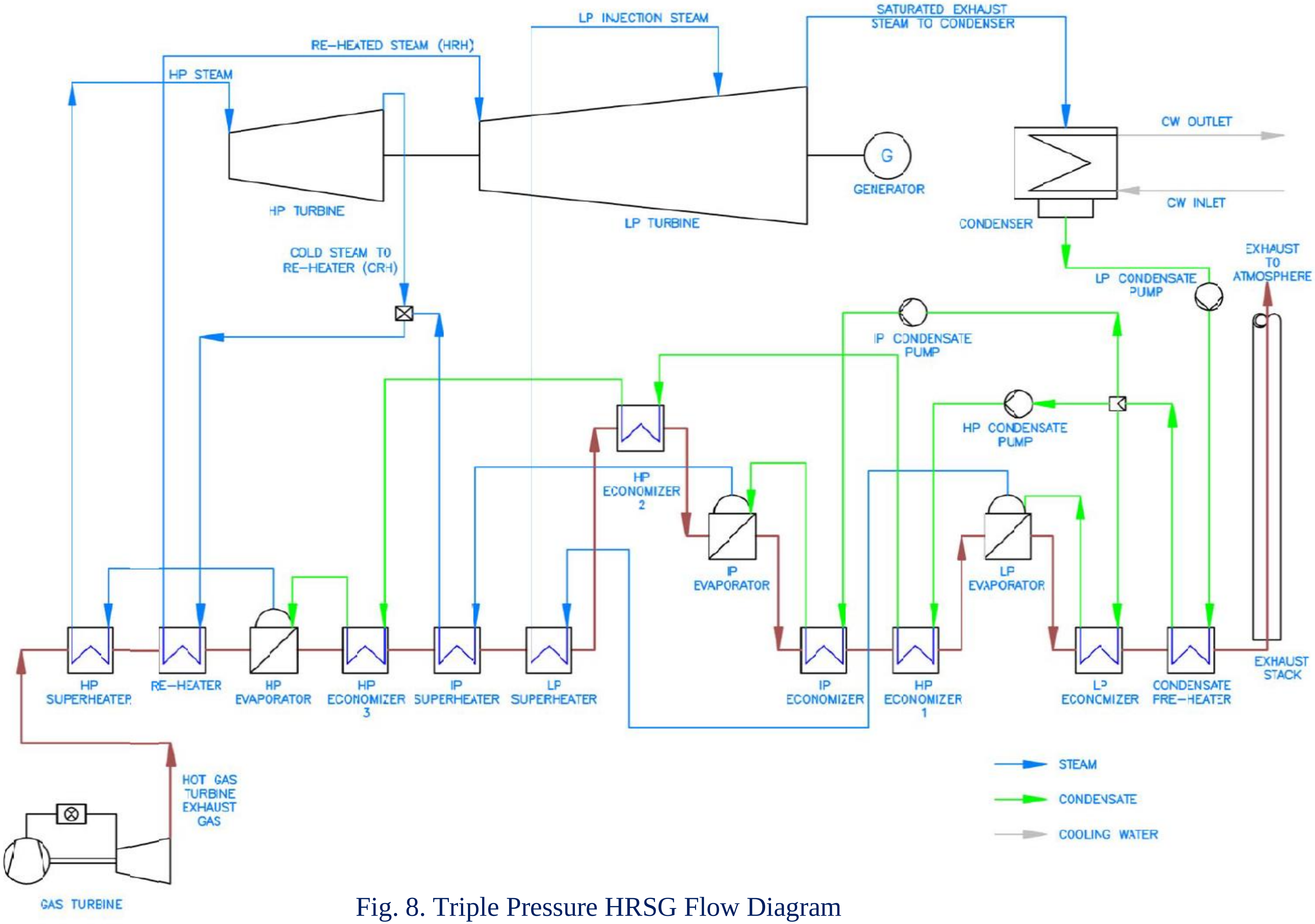


Fig. 8. Triple Pressure HRSG Flow Diagram

2.2.ENERGY ANALYSIS

In the present work, a study with various evaporating and condensing temperatures and pressures has been conducted to determine the performance of Combined Cycles with multiple Configurations.

The following assumptions are made to simplify this analysis, which includes the energy analysis.

1. All components are assumed to be a steady flow and steady-state process.

2. The changes in the kinetic energy and the potential energy of the components are negligible.

3. The pressure drops and heat loss in the components and the piping's are negligible.

From the first law of thermodynamics, energy is conserved for a process. It is applied to the combined cycle plant components to determine their energy efficiencies. Energy efficiency of a component is defined as the useful energy output divided by the energy supplied to the component in the form of fuel energy. The useful energy output is the electric work produced in case of the combustion gas turbine and steam turbine, whereas it is the energy gained by steam for the boilers.

An important parameter to measure the performance of the Combined cycle power plant is the overall plant efficiency. Using energy efficiency equations derived from first law of thermodynamics, overall plant first law efficiency "η" may be defined as:

$\eta = P_{Total} / Q_{Input}$

Where

P Total = Total Electrical Power from Plant (Gas Turbine + Steam Turbine)

Q Input = Fuel Energy Input to the System

Net Energy Input to a combined cycle power plant is the Fuel Energy inputted to the system and measured by LHV or Lower Heating Value of the fuel. LHV of a fuel is the amount of energy released by it during full combustion of the specified amount. Hence its unit is energy/weight (kJ/kg) therefore

Q Input = Fuel LHV x Fuel Mass Flow Rate.

The Individual Components efficiencies of combined cycle analysis is as calculated in the table below by firsts law analysis

GAS TURBINE	Efficiency = Net Electrical Power from Gas turbine / (Fuel Flow x LHV) For Analysis its considered as per mentioned by manufacturer 42.0%
HRSG	Efficiency = (Ms x (h9 – h6)) / (Mg x Cp x T4)
STEAM TURBINE	Efficiency = (Ms x (h6-h7)) / (Ms x Δh Isentropic)

2.3.EXERGY ANALYSIS

Exergy is the instrument which shows how far the system has deviated from equilibrium state. It is the measure of total available work until a system reaches equilibrium with its surroundings, it is also called the maximum useable work that is obtained in a process in which a system comes to equilibrium with its surroundings or in dead state.

Hence farther the initial state of system from dead state higher will be the exergy.

By doing first law analysis of we can do quantitate evaluation of the energies in the cycle. While by doing second law analysis or exergy analysis relates to the quality of energy and provides information on the irreversibility's in a process.

The Physical exergy of a working fluid at a state "i" is defined as

$$\epsilon_i = m\,[(hi - ho) - To\,(Si - So)]$$

m = Mass flow rate of the working fluid

hi = Enthalpy of working fluid at state point "i"

ho = Enthalpy of the working fluid at dead state

To = Temperature at dead state

Si = Entropy of the working fluid at state point "i"

So = Entropy of the working fluid at dead state

EXERGY DESTROYED = Xin - Xout

GAS TURBINE	X dest. = Xin Fuel – X out exhaust - Wgt
HRSG	X dest. = Xin GTexhaust – Xout Stack + (Xin Fw – Xout Steam)
STEAM TURBINE	X dest. = Xin Steam – Xout – Wst
TOTAL EXERGY DESTR.	X dest plant = Σ Dest components

2.4.INPUT PARAMETERS

Gas Turbine Inputs

The Calculations done for HRSG and combined cycle is based on GE Gas Turbine 9HA which is the largest and most efficient available heavy duty gas turbine.

Load Condition		Base
Fuel Type		Methane
Fuel LHV	kJ/kg	50044
Output Power	kW	409000
Heat rate	kJ/kWh	8555
Efficiency η		42.07%
Exhaust Gas Flow	Tons Per Hour	3064.7
Exhaust Temperature	ºC	633
Calculated Fuel Flow	kg/s	19.43

Table 1. Gas turbine Input Data

Since Efficiency η = GT Net Power / LHV x Fuel Flow

Thus

Fuel Flow = GT Net Power / LHV x η……………………………………………1

Fuel Flow = 409000 / 50044 x 0.4207

Fuel Flow = 19.43 kg/s

Other Inputs

Cooling water Inlet = 31°C

Cooling water Return = 41°C

Condensing Temperature = 41+3=45°C

Condensing Pressure = 0.1 Bara

Steam Turbine Efficiency = 90%

Generator efficiency = 98.6%

Steam Inlet Temperature = 570°C

Steam Inlet Pressure = 140 Bara

Reheat Pressure = 22 Bara

Steam Properties = IAPWS IF-97

HRSG Pinch Temperature = 20°C

HRSG Approach Temperature = 15°C

2.5. SOFTWARE RESULT VALIDATION BY MANUAL CALCULATION

The tool used for combined cycle analysis is GE GATE CYCLE V. 6.12

Based on GAS Turbine Inputs from above Table and considering a steam parameter of 140 bara and 570°C and condensate return at 140 bara and 46.91°C and specific heat to be 1.145 kJ/kg.

Starting point in the designing of an HRSG is the fixation of pinch and approach points based on this we get TG3 & Tw2 with which we can calculate the mass flow of steam which can be generated.

From Steam Tables

Description		Unit	Value
Saturation Temperature of Steam at 140 Bara	Ts	°C	336.64
Enthalpy of Superheated Steam	hs 1	kJ/kg	3511.15
Enthalpy Of water at Evaporator	hw2	kJ/kg	1466.76
Enthalpy Of water at Economizer inlet	hw1	kJ/kg	208.46

Table 2. Values from steam tables

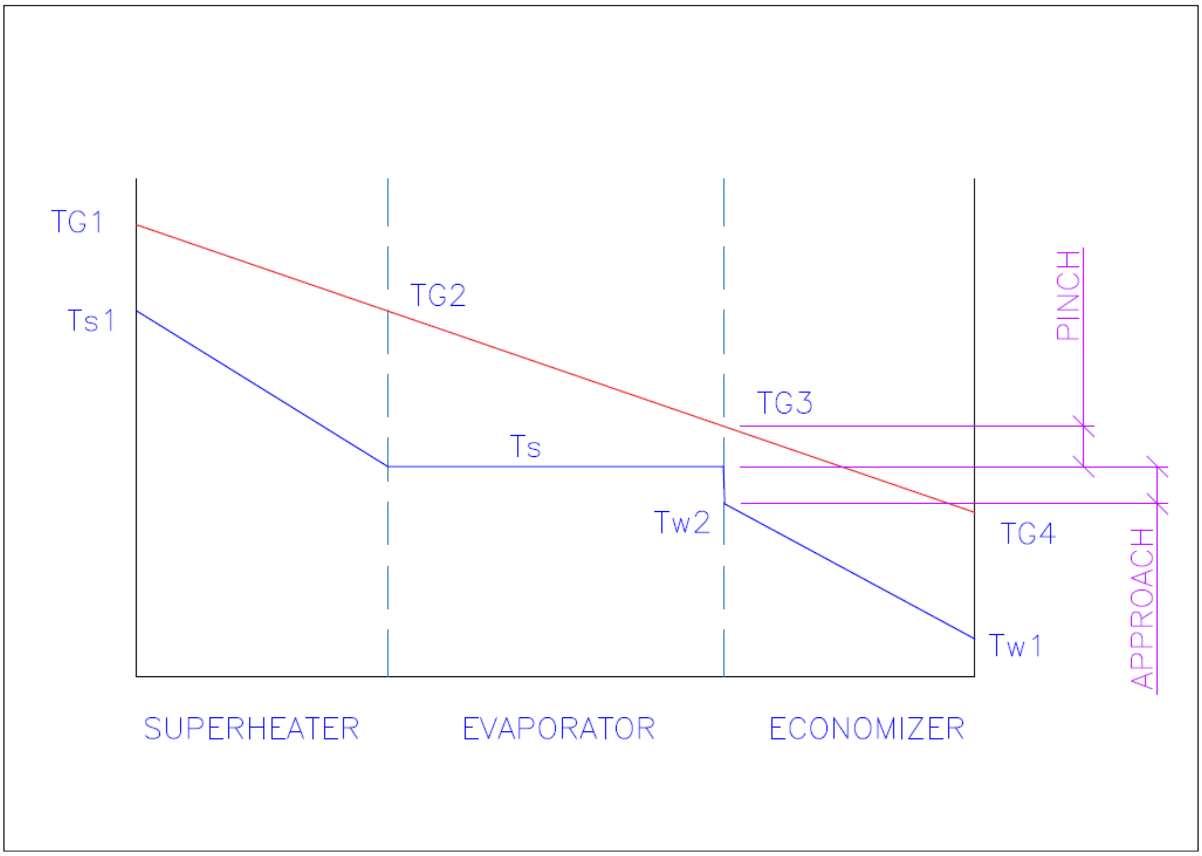


Fig. 9. Single Pressure HRSG Temperature Profile

Exhaust Gas temperature at evaporator outlet (TG3) = 336.64+20 = 356.64 °C

Temperature of feedwater at evaporator Inlet (Tw2) = 336.64-15 = 321.64 °C

Considering Energy Balance of Superheater and economizer

$$\text{Heat Lost by Exhaust Gases} = \text{Heat Gained by Feed water}$$

$$MG \times Cp \times (TG1 - TG3) = Ms \times (hs1 - hw2)$$

$$Ms = MG \times Cp \times (TG1 - TG3) / (hs1 - hw2) \quad \ldots\ldots 2$$

Where Ms = Mass Of steam

MG = Mass Of Exhaust Gas

Cp = Specific heat

hs1 = enthalpy at Ts1

hw2 = enthalpy at Tw2

Putting The Values in above eq. 1

$Ms = (3064.7/3.6) \times 1.145 \times (633 - 356.64) / (3511.15 - 1466.76)$

$Ms = 131.77$ kg/s

For Finding exhaust gas temperature from the HRSG

Doing energy balance for the economizer

$$\text{Heat Lost by Exhaust Gases} = \text{Heat Gained by Feed water}$$

$$MG \times Cp \times (TG3 - TG4) = Ms \times (hw2 - hw1)$$

$$TG4 = TG3 - (Ms \times (hw2 - hw1) / MG \times Cp) \quad \ldots\ldots 3$$

Where hw1 = enthalpy at Tw1

Putting The Values in above eq. 2

$TG4 = 356.64 - (131.77 \times (1466.76 - 208.46) / 851.31 \times 1.145)$

$TG4 = 356.64 - 170.101$

$TG4 = 186\ ^{o}C$

Efficiency calculation

Fuel LHV = 50044

Fuel Flow from Gas Turbine as calculated from eq. 1 = 19.43 kg/s

Hence Net Energy Inputted by the fuel = LHV x Fuel Flow = 50044 x 19.43 = 972354.92 kW

As mentioned in 3.2 for CCPP efficiency steam turbine power should be calculated first.

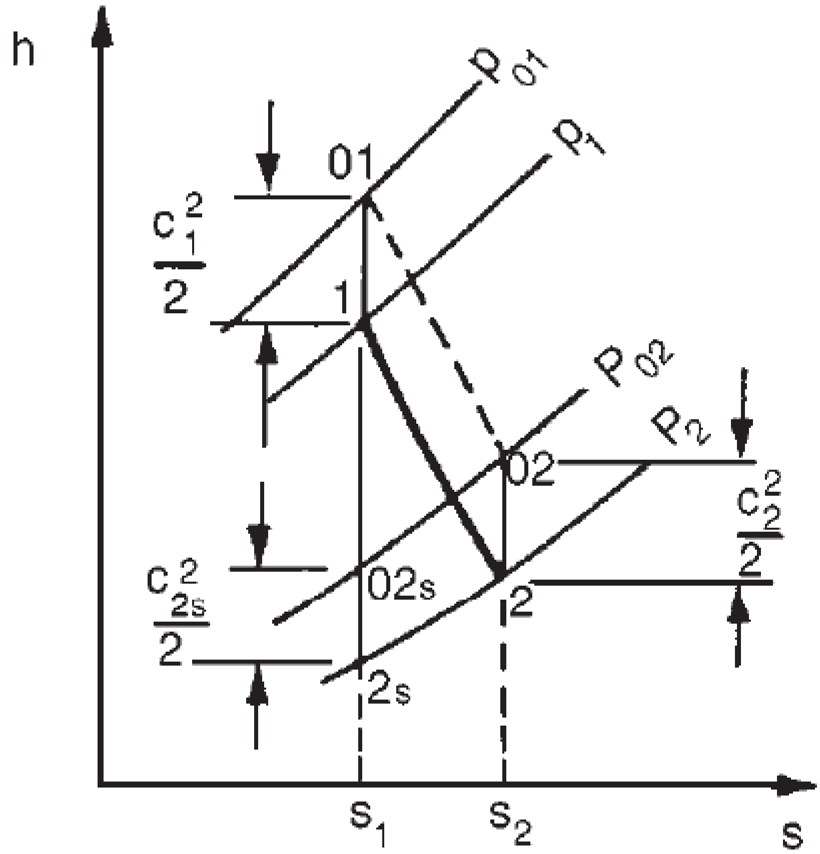


Fig. 10. Steam Turbine expansion h-s diagram

The power of steam turbine hence can be calculated as follows.

Enthalpy of Inlet steam = 3511.15

Entropy of inlet steam = 6.63 kJ/ kg-k

Exhaust pressure of turbine = 0.1 bara

Isentropic enthalpy at exhaust (@0.1 Bara & 6.63 entropy) = 2098.9 kJ/kg

Hence Net available drop in enthalpy = 3511.15 – 2098.9

=1412.25 kJ/kg

Since 90% efficiency is considered

Thus actual available enthalpy drop Δh = 11412.25 x 0.9 = 1271.025 kJ/kg

Now the power from steam turbine

= ((Ms x Δh) – losses) x Generator Efficiency

Net ST Power = (131.77 x 1271.025) x 0.9956 x 0.986

Net ST Power = 164081 kW

Total CCPP electrical Power = GT Power +ST Power

Total CCPP electrical Power = 409000+164081 = 573081 kW

Thus CCPP Efficiency η = P Total / Q Input

η = Total Power / LHV x Fuel Flow

η = 573081 / 972354.92

η = 58.94%

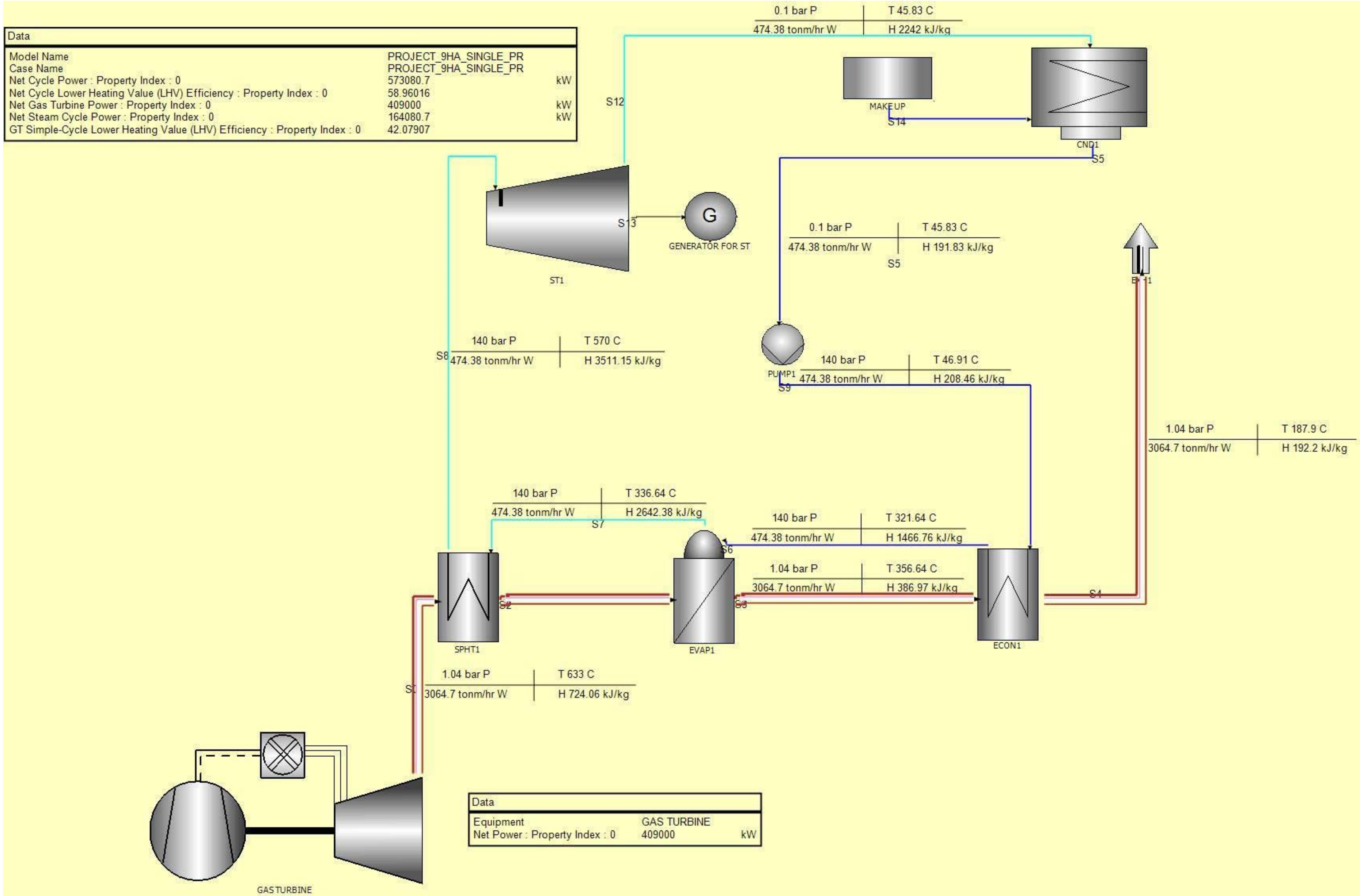


Fig. 11. GATE CYCLE HMBD for Single Pressure HRSG

COMPARITIVE OF GATE CYCLE OUTPUT VALUES AND CALCULATED VALUES

Description		Unit	Gate Cycle	Manual
Saturation Temperature of Steam at 140 Bara	Ts	⁰C	336.64	336.64
Enthalpy of Superheated Steam	hs 1	kJ/kg	3511.15	3511.15
Enthalpy of water at Evaporator	hw2	kJ/kg	1466.76	1466.76
Enthalpy of water at Economizer inlet	hw1	kJ/kg	208.46	208.46
Exhaust Gas temperature at evaporator outlet	TG3	⁰C	356.64	356.64
Temperature of feedwater at evaporator Inlet	Tw2	⁰C	321.64	321.64
Mass of steam	Ms	kg/s	131.77	131.77
HRSG exhaust to atmosphere	TG4	⁰C	187	186
Net Steam Turbine power		kW	164080.7	164081
Total CCPP Power Output		kW	573080.7	573081
Net CPPP efficiency		%	58.96	58.94

Table. 3. Comparative Of Gate Cycle Output Values And Calculated Values

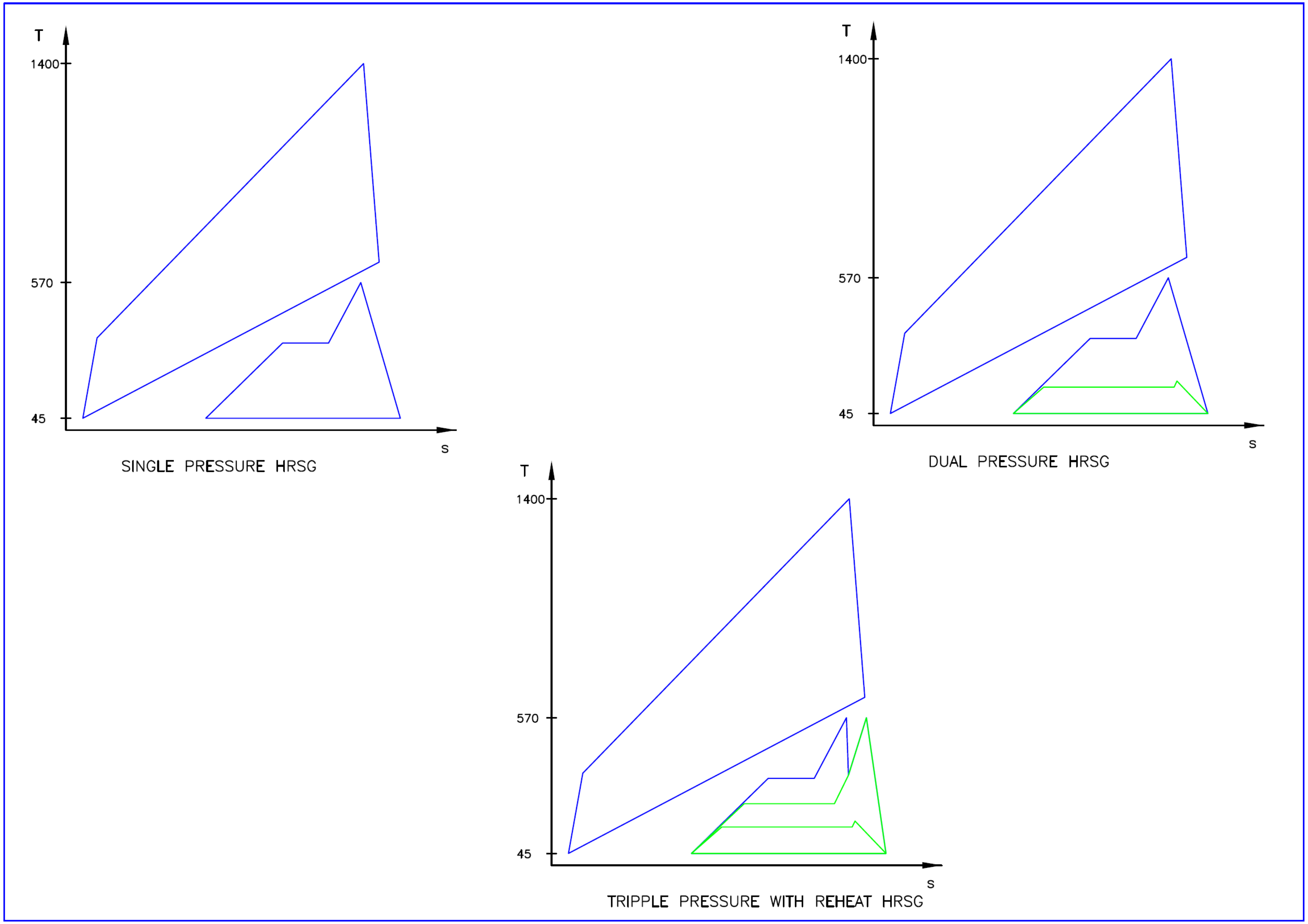


Fig. 12. TS Diagram Single Dual & Tripple pressure with Reheat HRSG

CHAPTER 4

4. RESULTS & DISCUSSIONS

A simple model graph based on varying exhaust stack temperature to show the effect of Stack Loss On Combined Cycle Efficiency. The CCPP efficiency can go upto 63 % efficiency if stack temperatures are reduced to below 80°C.

However, stack temperature cannot be straight forward reduced and different combinations of HRSG are used to achieve this. The Models are simulated on Gate Cycle for to achieve most efficient combination in combined cycle.

1. A single pressure HRSG with straight condensing steam turbine combination.
2. Dual pressure with straight condensing cum Lp injection steam turbine combination
3. Triple pressure with Re-heat Condensing cum Lp injection steam turbine.

Its is known For Rankine cycles to achieve high efficient the steam pressure has to be high as possible and hence most of the steam turbines operating in Combined Cycle plants operate with 140 bar High Pressure steam. At this pressures of 140 bar the saturation temperature will be above 335 °C. The gas exiting the evaporative section will be in the order of 335 °C. Hence absorption of the remaining heat of the flue gas has to be in the economizer. This is not possible and hence high temperature exhaust is let to atmosphere and energy is wasted.

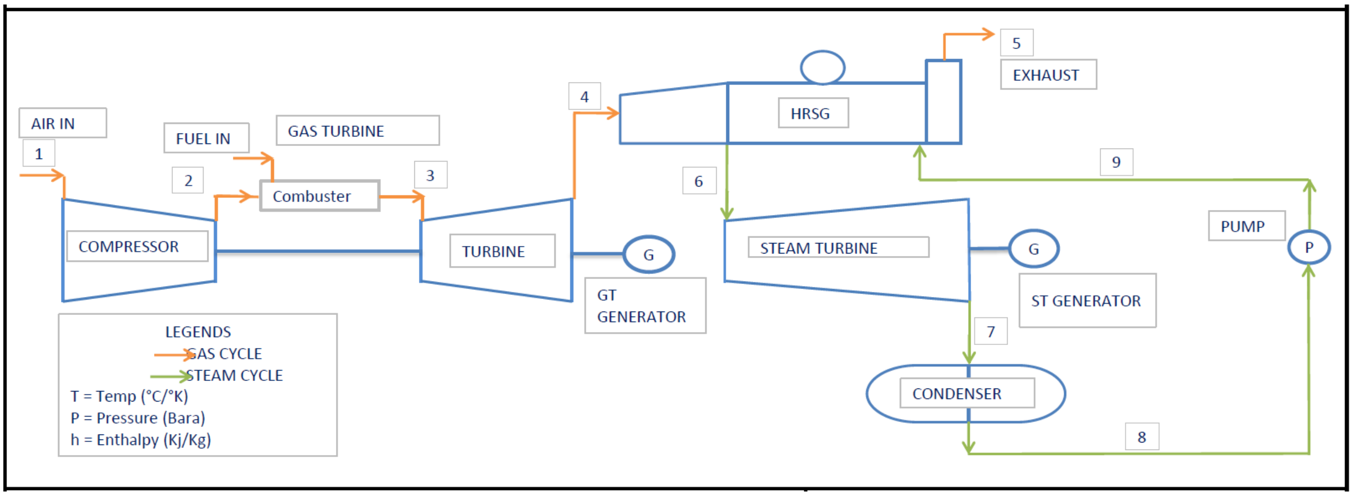


Fig 13. CCPP Flow Schematic

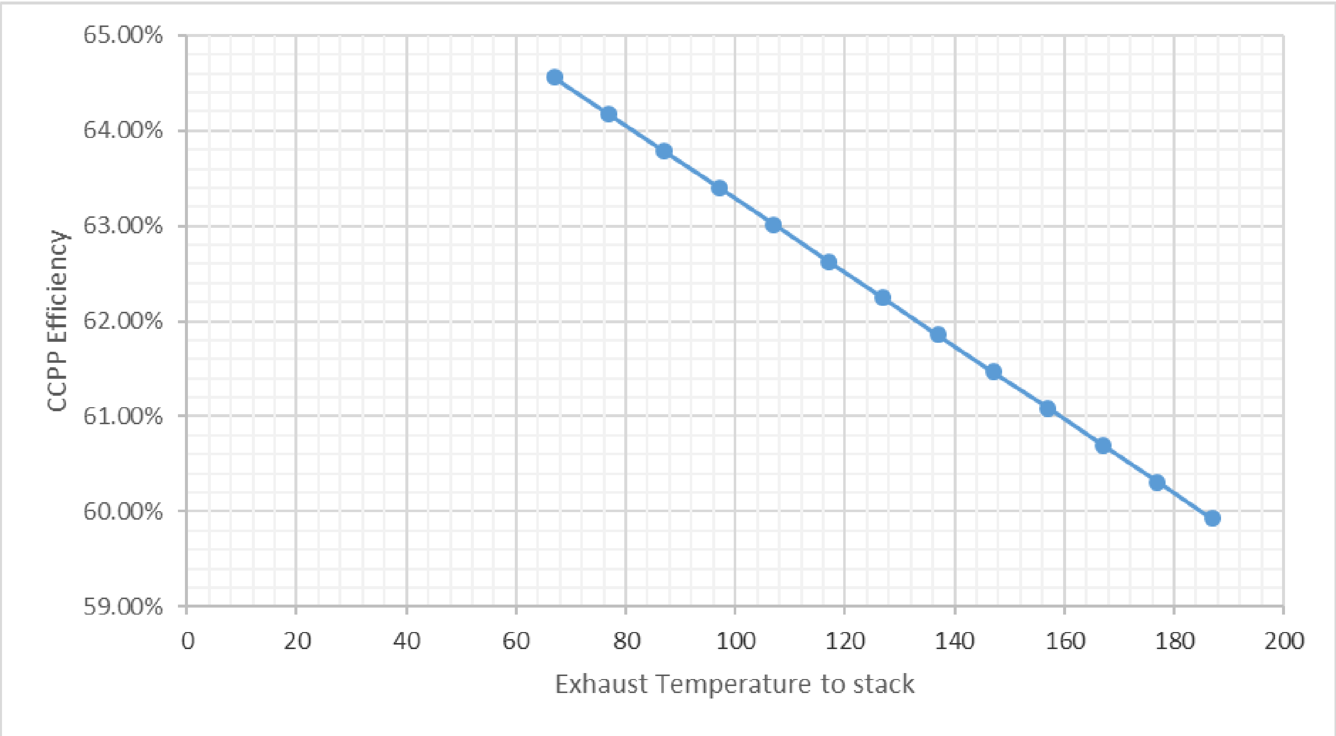


Fig 14. Chart CCPP efficiency Vs Stack Temperature

TABLE 4											
CASE	FUEL LHV kJ/kg	SP. HEAT Cp kJ/kg	GT Eff. %	GT Exh. Flow kg/s	GT Exh. Temp° C	GT Power kW	GT Fuel Flow kg/s	HRSG Stack Temp° C	Steam Prod. kg/s	Steam Temp T6 °C	P6 Bara
1	50044	1.145	42.7	851.3	633.0	409000	19.140	187	131.58	570.0	140
2	50044	1.145	42.7	851.3	633.0	409000	19.140	177	134.53	570.0	140
3	50044	1.145	42.7	851.3	633.0	409000	19.140	167	137.48	570.0	140
4	50044	1.145	42.7	851.3	633.0	409000	19.140	157	140.43	570.0	140
5	50044	1.145	42.7	851.3	633.0	409000	19.140	147	143.38	570.0	140
6	50044	1.145	42.7	851.3	633.0	409000	19.140	137	146.33	570.0	140
7	50044	1.145	42.7	851.3	633.0	409000	19.140	127	149.28	570.0	140
8	50044	1.145	42.7	851.3	633.0	409000	19.140	117	152.23	570.0	140
9	50044	1.145	42.7	851.3	633.0	409000	19.140	107	155.18	570.0	140
10	50044	1.145	42.7	851.3	633.0	409000	19.140	97	158.13	570.0	140
11	50044	1.145	42.7	851.3	633.0	409000	19.140	87	161.08	570.0	140
12	50044	1.145	42.7	851.3	633.0	409000	19.140	77	164.03	570.0	140
13	50044	1.145	42.7	851.3	633.0	409000	19.140	67	166.98	570.0	140

TABLE 4 CONT.											
h6	Cond . Pr. P7	Isentropic "h"at Exit kJ/kg	T7 ° C	h7 kJ/kg	P9 Bar a	T9 ° C	h9 kJ/kg	Power From Steam Turbine kW	Net Cycle Power kW	Total Chem. Power in Fuel kW	CCPP %
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	164990	573990	957845.4	59.93
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	168689	577689	957845.4	60.31
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	172388	581388	957845.4	60.70
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	176088	585088	957845.4	61.08
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	179787	588787	957845.4	61.47
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	183486	592486	957845.4	61.86
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	187186	596186	957845.4	62.24
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	190885	599885	957845.4	62.63
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	194584	603584	957845.4	63.01
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	198284	607284	957845.4	63.40
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	201983	610983	957845.4	63.79
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	205682	614682	957845.4	64.17
3513.9	0.09	2086.62	43.8	2229.3	140	43.8	474.38	209382	618382	957845.4	64.56

Table 4. Calculated values for Chart CCPP efficiency Vs Stack Temperature

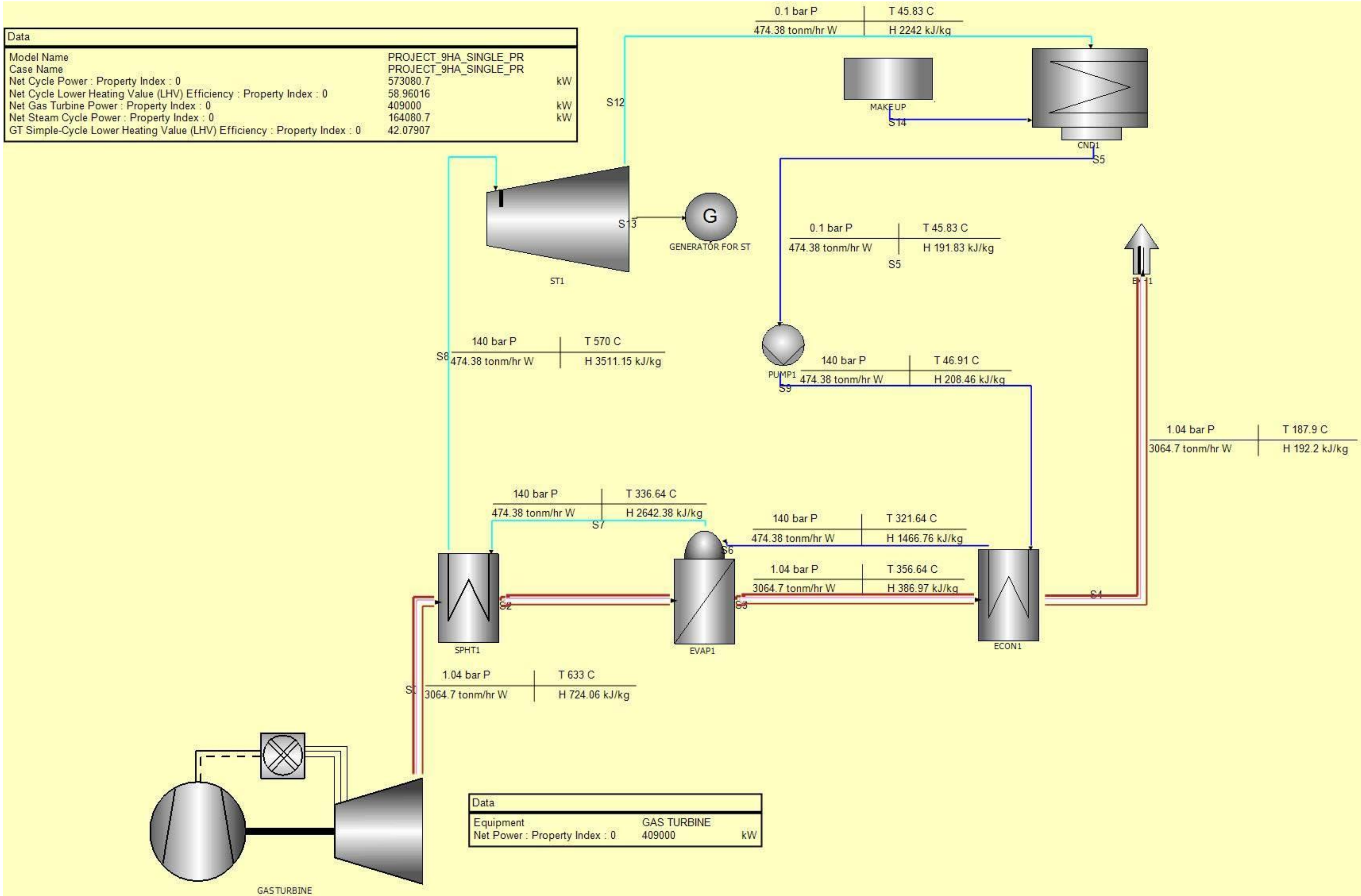


Fig. 15. Single Pressure HRSG HMBD

Table 5. Single Pressure HRSG Stream details

Stream	From	To	Flow	Pressure	Temperature	Enthalpy	Quality
			tonm/hr	bar	C	kJ/kg	
Primary Fuel Inlet	GAS TURBINE	GAS TURBINE	69.9184	20.6843	26.67	22.9493	1
Inlet Air	GAS TURBINE	GAS TURBINE	2994.7813	1.0132	48	34.6081	1
Cooling Water Exit	CND1	CND1	23267.1631	1.0342	25.5557	107.1824	0
Cooling Water Inlet	CND1	CND1	23267.1631	1.0342	15.56	65.3825	0
Blowdown Outlet	EVAP1	EVAP1	0	140	336.6415	1571.6461	0
S0	GAS TURBINE	SPHT1	3064.6998	1.0377	633	724.0643	1
S12	ST1	CND1	474.3842	0.1	45.8328	2241.997	0.8567
S13	ST1	GENERATOR FOR ST	0	0	-273.15	0	0.5
S14	MAKE UP	CND1	0	1.0342	15.56	65.3825	0
S2	SPHT1	EVAP1	3064.6998	1.0377	509.6248	570.7614	0
S3	EVAP1	ECON1	3064.6998	1.0377	356.6416	386.9681	0
S4	ECON1	EXH1	3064.6998	1.0377	187.9015	192.1956	0
S5	CND1	PUMP1	474.3842	0.1	45.8328	191.8333	0
S6	ECON1	EVAP1	474.3842	140	321.6405	1466.7643	0
S7	EVAP1	SPHT1	474.3842	140	336.6415	2642.382	1
S8	SPHT1	ST1	474.3842	140	569.9991	3511.1486	1
S9	PUMP1	ECON1	474.3842	140	46.912	208.4608	0
Expansion Line End	ST1	ST1	474.3842	0.1	45.8328	2241.997	0.8567
Internal Pump Flow	PUMP1	PUMP1	474.3842	0.1	45.8328	191.8333	0

Data		
Model Name	PROJECT_9HA_SINGLE_PR	
Case Name	PROJECT_9HA_SINGLE_PR	
Net Cycle Power : Property Index : 0	573080.7	kW
Net Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	58.96016	
Net Gas Turbine Power : Property Index : 0	409000	kW
Net Steam Cycle Power : Property Index : 0	164080.7	kW
GT Simple-Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	42.07907	

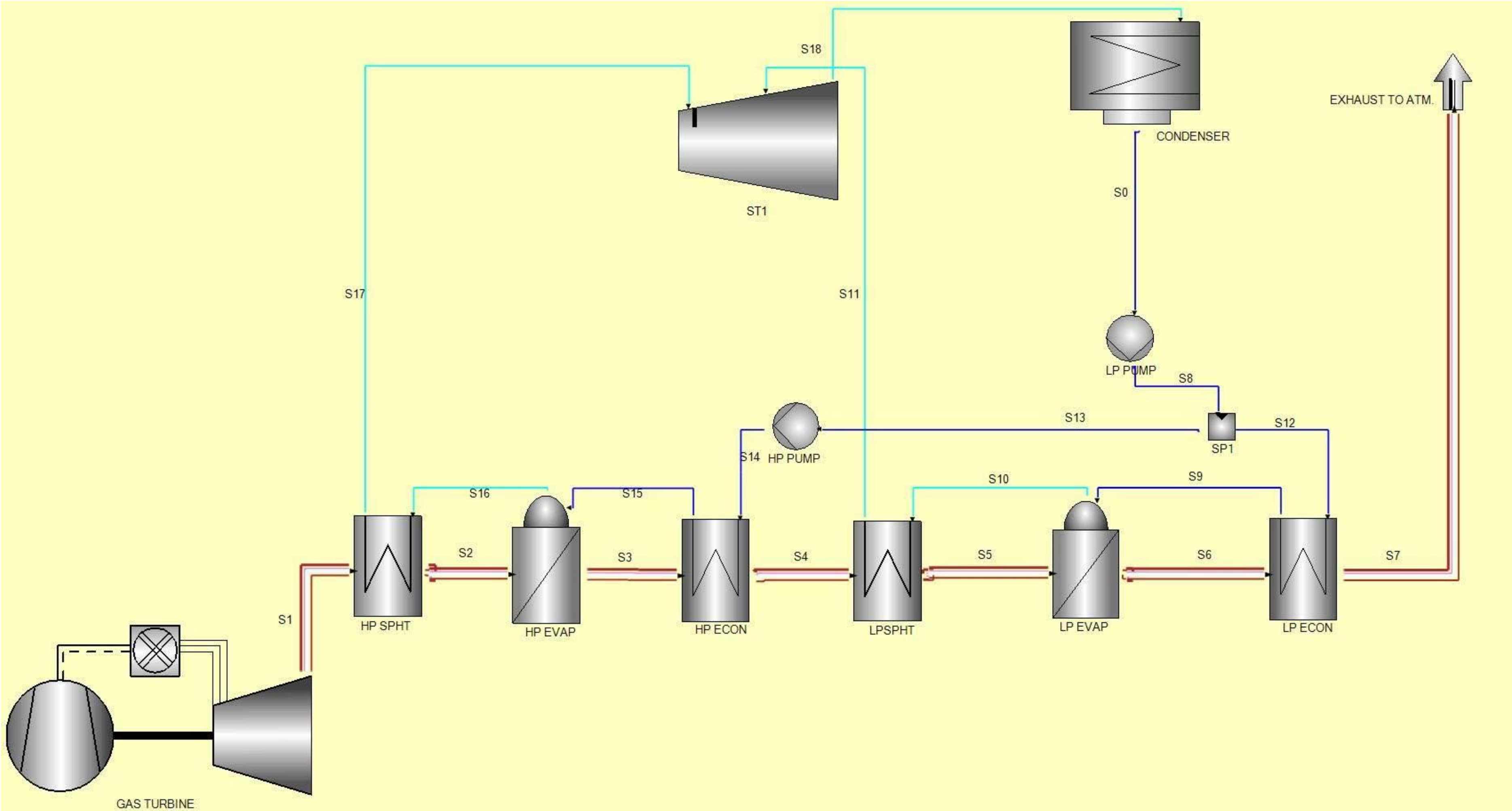


Fig. 16. Dual Pressure HRSG Diagram

Table 6. Dual Pressure HRSG Stream details

Stream	From	To	Flow	Pressure	Temperature	Enthalpy	Quality
			tonm/hr	bar	C	kJ/kg	
Primary Fuel Inlet	GAS TURBINE	GAS TURBINE	69.9184	20.6843	26.67	22.9493	1
Inlet Air	GAS TURBINE	GAS TURBINE	2994.7813	1.0132	35	20.1265	1
Cooling Water Exit	CONDENSER	CONDENSER	30422.5842	3.5585	42.0016	176.1246	0
Cooling Water Inlet	CONDENSER	CONDENSER	30422.5842	4	34	142.7348	0
Blowdown Outlet	HP EVAP	HP EVAP	0	140	336.6415	1571.6461	0
S0	CONDENSER	LP PUMP	496.511	0.1	45.8126	191.7488	0
S1	GAS TURBINE	HP SPHT	3064.6998	1.0377	633	702.8859	1
S10	LP EVAP	LPSPHT	21.7981	4.6119	148.825	2743.9823	1
S11	LPSPHT	ST1	21.7981	4.6119	169.9995	2791.8547	1
S12	SP1	LP ECON	22.1302	4.6119	45.8469	192.285	0
S13	SP1	HP PUMP	474.3808	4.6119	45.8469	192.285	0
S14	HP PUMP	HP ECON	474.3808	140	47.7219	211.8206	0
S15	HP ECON	HP EVAP	474.3808	140	321.6418	1466.7729	0
S16	HP EVAP	HP SPHT	474.3808	140	336.6415	2642.382	1
S17	HP SPHT	ST1	474.3808	140	569.9991	3511.1486	1
S18	ST1	CONDENSER	496.179	0.1	45.8328	2239.0894	0.8555
S19	MAKE UP	CONDENSER	0.332	1.0342	15.56	65.3825	0
S2	HP SPHT	HP EVAP	3064.6998	1.0377	514.2536	559.737	0
S3	HP EVAP	HP ECON	3064.6998	1.0377	356.6416	375.9463	0
S4	HP ECON	LPSPHT	3064.6998	1.0377	181.4871	179.7514	0
S5	LPSPHT	LP EVAP	3064.6998	1.0377	181.185	179.4134	0
S6	LP EVAP	LP ECON	3064.6998	1.0377	166.825	163.6746	0
S7	LP ECON	EXHAUST TO ATM.	3064.6998	1.0377	164.4152	161.035	0
S8	LP PUMP	SP1	496.511	4.6119	45.8469	192.285	0
S9	LP ECON	LP EVAP	22.1302	4.6119	131.825	554.2206	0
Expansion Line End	ST1	ST1	496.179	0.1	45.8328	2239.0894	0.8555
Internal Pump Flow	LP PUMP	LP PUMP	496.511	0.1	45.8126	191.7488	0
Internal Pump Flow	HP PUMP	HP PUMP	474.3808	4.6119	45.8469	192.285	0

Data		
Model Name	PROJECT_9HA_2PR_1	
Case Name	PROJECT_9HA_2PR_1	
Net Cycle Power : Property Index : 0	575718.5	kW
Net Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	59.23155	
Net Gas Turbine Power : Property Index : 0	409000	kW
Net Steam Cycle Power : Property Index : 0	166718.5	kW
GT Simple-Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	42.07907	

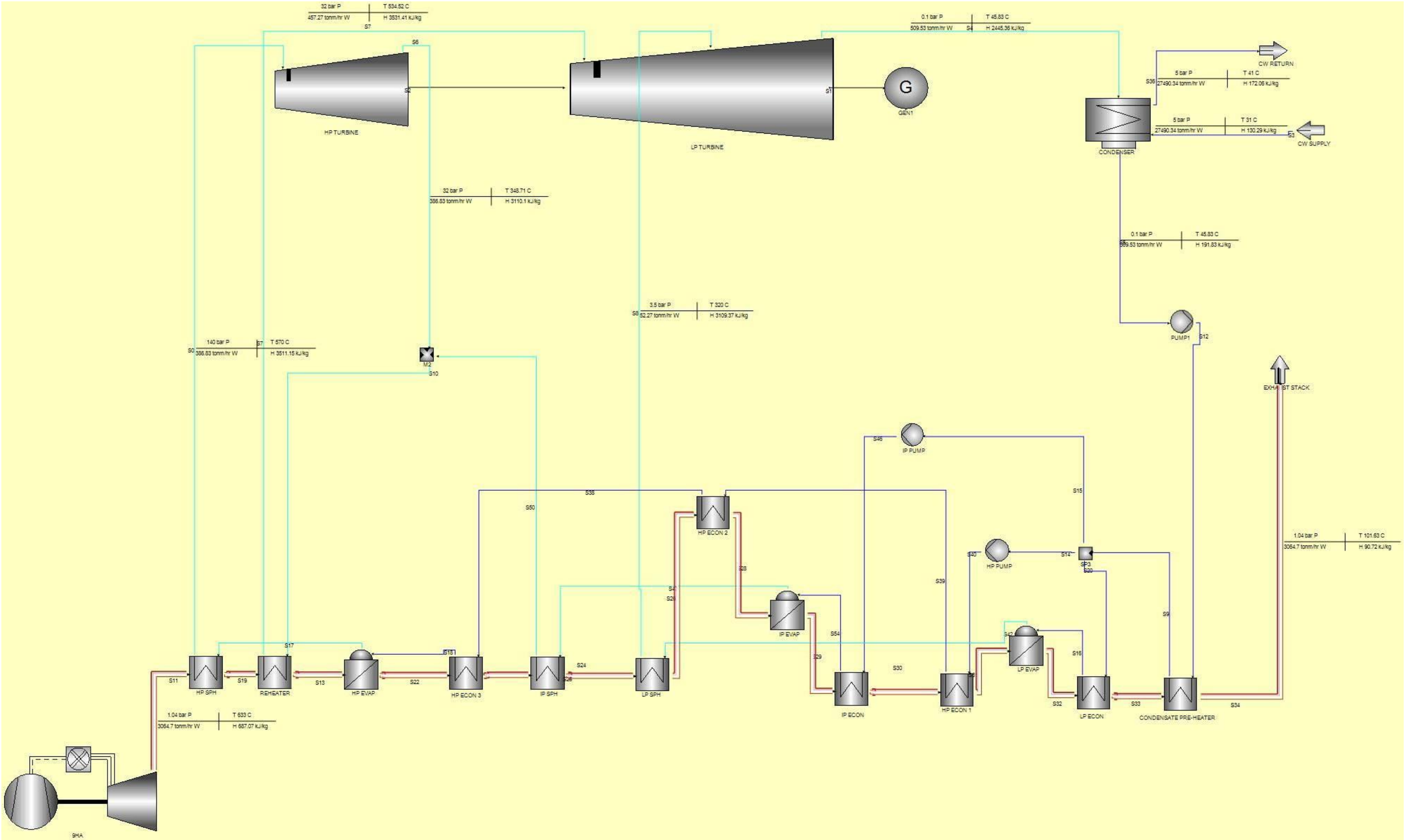


Fig. 17. Triple Pressure HRSG with Reheating Diagram

Table 7. Triple Pressure HRSG with Reheating Stream details

Stream	From	To	Flow	Pressure	Temperature	Enthalpy	Quality
			tonm/hr	bar	C	kJ/kg	
Primary Fuel Inlet	9HA	9HA	69.9184	14	15	-1.1903	1
Inlet Air	9HA	9HA	2994.7813	1.0132	15	-0.561	1
S0	HP SPH	HP TURBINE	386.8326	140	569.9991	3511.1486	1
S1	LP TURBINE	GEN1	0	0	-273.15	0	0.5
S10	M2	REHEATER	457.2651	32	341.5846	3093.0191	1
S11	9HA	HP SPH	3064.6998	1.0377	633	687.0685	1
S12	PUMP1	CONDENSATE PRE-HEATER	509.5304	3.5008	45.8587	192.2375	0
S13	REHEATER	HP EVAP	3064.6998	1.0377	483.7083	511.9961	0
S14	SP3	HP PUMP	386.8326	3.5008	130	546.3623	0
S15	SP3	IP PUMP	70.4326	3.5008	130	546.3623	0
S16	LP ECON	LP EVAP	52.2653	3.5008	138.8808	668.5229	0.0392
S17	HP EVAP	HP SPH	386.8326	140	336.6415	2642.382	1
S18	HP ECON 3	HP EVAP	386.8326	140	326.6418	1499.5145	0
S19	HP SPH	REHEATER	3064.6998	1.0377	540.0748	577.4111	0
S2	HP TURBINE	LP TURBINE	0	0	-273.15	0	0.5
S20	SP3	LP ECON	52.2653	3.5008	130	546.3623	0
S22	HP EVAP	HP ECON 3	3064.6998	1.0377	356.6416	367.741	0
S24	HP ECON 3	IP SPH	3064.6998	1.0377	352.9583	363.6083	0
S25	IP SPH	LP SPH	3064.6998	1.0377	348.8951	359.0908	0
S26	LP SPH	HP ECON 2	3064.6998	1.0377	343.1024	352.6491	0
S28	HP ECON 2	IP EVAP	3064.6998	1.0377	296.1501	300.6259	0
S29	IP EVAP	IP ECON	3064.6998	1.0377	257.4455	258.1752	0
S3	CW SUPPLY	CONDENSER	27490.3387	5	31	130.2932	0
S30	IP ECON	HP ECON 1	3064.6998	1.0377	248.9099	248.8617	0
S31	HP ECON 1	LP EVAP	3064.6998	1.0377	191.6502	186.8647	0
S32	LP EVAP	LP ECON	3064.6998	1.0377	158.8808	151.6802	0
S33	LP ECON	CONDENSATE PRE-HEATER	3064.6998	1.0377	156.9284	149.5969	0
S34	CONDENSATE PRE-HEATER	EXHAUST STACK	3064.6998	1.0377	101.6253	90.7156	0
S36	CONDENSER	CW RETURN	27490.3387	5	40.9989	172.0621	0
S38	HP ECON 2	HP ECON 3	386.8326	140	321.6418	1466.7726	0
S39	HP ECON 1	HP ECON 2	386.8326	140	243.3543	1054.7446	0
S4	LP TURBINE	CONDENSER	509.5304	0.1	45.8328	2445.3622	0.9417
S40	HP PUMP	HP ECON 1	386.8326	140	131.8612	563.545	0
S42	LP EVAP	LP SPH	52.2653	3.5008	138.8808	2731.6475	1
S46	IP PUMP	IP ECON	70.4326	32	130.389	549.9498	0
S47	IP EVAP	IP SPH	70.4326	32	237.4455	2802.345	1
S5	CONDENSER	PUMP1	509.5304	0.1	45.8328	191.8333	0
S50	IP SPH	M2	70.4326	32	303.9983	2999.2109	1
S54	IP ECON	IP EVAP	70.4326	32	222.4455	955.206	0
S6	HP TURBINE	M2	386.8325	32	348.7113	3110.0994	1
S7	REHEATER	LP TURBINE	457.2651	32	534.5187	3531.4106	1
S8	LP SPH	LP TURBINE	52.2653	3.5008	320.0005	3109.3736	1
S9	CONDENSATE PRE-HEATER	SP3	509.5304	3.5008	130	546.3623	0

Data		
Model Name	Project_9HA_3P_1	
Case Name	Project_9HA_3P_1	
Net Steam Cycle Power : Property Index : 0	187459	kW
Net Gas Turbine Power : Property Index : 0	409000	kW
Net Cycle Power : Property Index : 0	596458.9	kW
Net Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	61.36538	
ST Generator Efficiency : Property Index : 0	0.98	
Net Cycle Lower Heating Value (LHV) Heat Rate : Property Index : 0	1401.14	kcal/kW-hr
GT Simple-Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	42.07907	

Table 8. Triple Pressure HRSG with Reheating at Reduced Condensing Pressure

Stream	From	To	Flow	Pressure	Temperature	Enthalpy	Quality
			tonm/hr	bar	C	kJ/kg	
Primary Fuel Inlet	9HA	9HA	69.9184	14	15	-1.1903	1
Inlet Air	9HA	9HA	2994.7813	1.0132	15	-0.561	1
S0	HP SPH	HP TURBINE	386.3482	140	569.9991	3511.1486	1
S1	LP TURBINE	GEN1	0	0	-273.15	0	0.5
S10	M2	REHEATER	462.0039	32	341.1372	3091.9408	1
S11	9HA	HP SPH	3064.6998	1.0377	633	687.0685	1
S12	PUMP1	CONDENSATE PRE-HEATER	523.3989	3.5008	39.049	163.7831	0
S13	REHEATER	HP EVAP	3064.6998	1.0377	483.0389	511.2576	0
S14	SP3	HP PUMP	386.3482	3.5008	130	546.3623	0
S15	SP3	IP PUMP	75.6557	3.5008	130	546.3623	0
S16	LP ECON	LP EVAP	61.395	3.5008	138.8808	617.7649	0.0156
S17	HP EVAP	HP SPH	386.3482	140	336.6415	2642.382	1
S18	HP ECON 3	HP EVAP	386.3482	140	324.6418	1486.2119	0
S19	HP SPH	REHEATER	3064.6998	1.0377	540.1977	577.5483	0
S2	HP TURBINE	LP TURBINE	0	0	-273.15	0	0.5
S20	SP3	LP ECON	61.395	3.5008	130	546.3623	0
S22	HP EVAP	HP ECON 3	3064.6998	1.0377	354.6416	365.5061	0
S24	HP ECON 3	IP SPH	3064.6998	1.0377	352.4607	363.0556	0
S25	IP SPH	LP SPH	3064.6998	1.0377	348.1173	358.2231	0
S26	LP SPH	HP ECON 2	3064.6998	1.0377	341.341	350.6562	0
S28	HP ECON 2	IP EVAP	3064.6998	1.0377	288.3612	292.0475	0
S29	IP EVAP	IP ECON	3064.6998	1.0377	245.4455	245.0911	0
S3	CW SUPPLY	CONDENSER	38892.9449	5	31	130.2932	0
S30	IP ECON	HP ECON 1	3064.6998	1.0377	237.4961	236.4445	0
S31	HP ECON 1	LP EVAP	3064.6998	1.0377	186.4119	181.2017	0
S32	LP EVAP	LP ECON	3064.6998	1.0377	146.8808	138.8544	0
S33	LP ECON	CONDENSATE PRE-HEATER	3064.6998	1.0377	145.5482	137.424	0
S34	CONDENSATE PRE-HEATER	EXHAUST STACK	3064.6998	1.0377	84.0332	72.0938	0
S36	CONDENSER	CW RETURN	38892.9449	5	38.2221	160.4618	0
S38	HP ECON 2	HP ECON 3	386.3482	140	321.6418	1466.7726	0
S39	HP ECON 1	HP ECON 2	386.3482	140	231.9405	1001.7427	0
S4	LP TURBINE	CONDENSER	523.3989	0.07	39.0246	2405.159	0.9305
S40	HP PUMP	HP ECON 1	386.3482	140	131.8612	563.545	0
S42	LP EVAP	LP SPH	61.395	3.5008	138.8808	2731.6475	1
S46	IP PUMP	IP ECON	75.6557	32	130.389	549.9498	0
S47	IP EVAP	IP SPH	75.6557	32	237.4455	2802.345	1
S5	CONDENSER	PUMP1	523.3989	0.07	39.0246	163.3765	0

S50	IP SPH	M2	75.6557	32	303.9983	2999.2109	1
S54	IP ECON	IP EVAP	75.6557	32	210.4455	900.2115	0
S6	HP TURBINE	M2	386.3482	32	348.7113	3110.0994	1
S7	REHEATER	LP TURBINE	462.0039	32	534.6416	3531.6869	1
S8	LP SPH	LP TURBINE	61.395	3.5008	320.0005	3109.3736	1
S9	CONDENSATE PRE-HEATER	SP3	523.3989	3.5008	130	546.3623	0

Data		
Model Name	Project_9HA_3P_1	
Case Name	Project_9HA_3P_1	
Net Steam Cycle Power : Property Index : 0	196254.9	kW
Net Gas Turbine Power : Property Index : 0	409000	kW
Net Cycle Power : Property Index : 0	605254.9	kW
Net Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	62.27033	
ST Generator Efficiency : Property Index : 0	0.98	
Net Cycle Lower Heating Value (LHV) Heat Rate : Property Index : 0	1380.77	kcal/kW-hr
GT Simple-Cycle Lower Heating Value (LHV) Efficiency : Property Index : 0	42.07907	

Table 9. COMPARISON TABLE OF ALL HRSG

		SINGLE Pr.	DUAL Pr.	TRIPPLE Pr. 1	TRIPPLE Pr. 2
Load Condition		Base	Base	Base	Base
Fuel Type		Methane	Methane	Methane	Methane
Fuel LHV	kJ/kg	50044	50044	50044	50044
GT Output Power	Kw	409000	409000	409000	409000
Simply Cycle Efficiency	%	42.07	42.07	42.07	42.07
Condensing Pressure	bara	0.1	0.1	0.1	0.07
Cooling Water	TPH	23267	30422	27490	38893
ST Output Power	**kW**	**164081**	**166718**	**187459**	**196255**
ST Efficiency	%	90	90	90	90
CCPP Efficiency (η)	**%**	**58.96**	**59.23**	**61.36**	**62.27**

CHAPTER 5

5. CONCLUSION

In this thesis an extensive study is done on combined cycle while comparisons of single, dual and triple pressure HRSGs and condensing pressure effects. Conclusions of this analysis are summarized as follows:

1. The Calculated data are in good agreement with software outputs.
2. The increase of pressure levels of steam generation in HRSG increases the heat recovery from the flue gas and as a result, the net energy increases of the combined cycle.
3. There is significant increase in efficiency of triple pressure HRSG with Reheat Steam turbine as reheating increases the Rankine cycle efficiency considerable and the triple pressure HRSG improves the heat recovery from exhaust data hence a cumulative effect is seen.
4. Re-Heating Improves the moisture content in the exhaust of steam turbine which idealizes to use lower condensing pressure and hence more power from the steam turbine simultaneously increasing the cycle efficiency also.
5. Since Re-heated steam turbines have lower moisture content in exhaust the design life and operability of machinery improves.

Based on the thesis and analysis done combined cycle with triple pressure HRSG and a Reheat Steam turbine is a best efficient solution also if this is utilized with lower condensing pressures the CCPP efficiency crosses 62%. Based on the study performed the best solution CCPP efficiency is 62.27%

6. SCOPE FOR FUTURE WORK

1. Further Investigation can be done and by adding additional economizer to IP & LP sections which may result in exhaust temperature to be reduced further.
2. Re-generative Rankine cycle can also be combined to improve the Rankine cycle efficiency
3. Re-heated gas turbines can also be used which will improve the Brayton cycle and combined cycle too.

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