A DISSERTATION ON

MODELLING OF TRIPLE PRESSURE-REHEAT AND SUPPLEMENTARY FIRED COMBINED CYCLE POWER PLANT USING GT PRO

Submitted in partial fulfillment of the requirement

for the award of the degree of

Master of Technology

In

Thermal Engineering

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DECLARATION

I hereby declare that the work, which is being presented in this dissertation, entitled **"Modelling of Triple Pressure-Reheat and Supplementary Fired Combined Cycle Power Plant Using GT Pro"** towards the partial fulfillment of the requirements for the award of the degree of Master of Engineering with specialization in Thermal Engineering, from Delhi Technological University Delhi, is an authentic record of my own work carried out under the supervision of **PROF. R.S. MISHRA & DR. RAJESH KUMAR** Associate Professor, Department Mechanical Engineering, at Delhi Technological University, Delhi.

The matter embodied in this dissertation report has not been submitted by me for the award of any other degree.

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This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

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CERTIFICATE

It is certified that Sachin, Roll no. 2K13/THE/27, student of M.Tech. Thermal Engineering, Delhi Technological University, has submitted the dissertation titled "**Modelling of Triple Pressure-Reheat and Supplementary Fired Combined Cycle Power Plant Using GT Pro**" under my guidance towards the partial fulfillment of the requirements for the award of the degree of Master of Technology in Thermal Engineering.

His work is found to be satisfactory and his discipline impeccable during the course of the project. His enthusiasm, attitude towards the project is appreciated.

I wish him success in all his endeavors.

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ABSTRACT

Due to increase in demand of electricity, dominance of fuel prices, increase in cost of electricity at peak load conditions need development of power plants. Hence requirement of efficient combined cycle power plant comes into picture.

Thermodynamic modelling & analysis of the triple pressure reheat combined cycle power plant with supplementary fired are being conducted using GT Pro software. The objective of the thermodynamic optimization is to enhance the efficiency.

Improve Heat Rate of the CCGT and to maximize the power generation from Brayton & Rankine Cycle, therefore, model of combined cycle power plants is being divided into three sections i.e. Gas Turbine Model, HRSG Model, and Steam Turbine Model in GT Pro software.

For Gas turbine model is influenced by parameters such as ambient temperature, compression ratio and turbine inlet temperature and same to be optimizing for performance. Due to increase in ambient temperature, the performance of combined cycle power plant will reduce.

For Steam Turbine model, the main steam pressure & temperature are one which majorly impact & enhance the performance of Steam turbine model. Similarly in HRSG model, mainly HP, IP, LP drum pressure and pinch point play a major role for optimization of performance. The plant performance will improve due to increase in Steam pressure & temperature.

Performance of all three models has been computed using GT Pro software. The numerical computation has been carried out for find out of impact of ambient temperature, Compression ratio, Gas turbine inlet temperature, steam inlet pressure & temperature on system performance & it was observed that these parameters playing important role in system performance of combined cycle power plant. HRSG performance will increase due to increase in pressure of HP, IP & LP drum.

Overall system performance will improve due to increase in Gas turbine inlet temperature, Compressor ratio & condenser pressure.

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NOMENCLATURE

C_{pa}	The specific heat of the air (kJ/kg·K)
C_{pf}	The specific heat of the fuel (kJ/kg·K)
Cpg	The specific heat of flue gas (kJ/kg·K)
f	The fuel-air ratio
h	Enthalpy (kJ/kg)
h_{1f}	The heat loss factor in the heat recovery steam generator
$\dot{m}_a$	The air mass flow rate (kg/s)
$\dot{m}_f$	The fuel mass flow rate (kg/s)
$\dot{m}_g$	The mass flowrate of the exhaust gases through the gas turbine (kg/s)
$\dot{m}_w$	The water mass flow rate (kg/s)
p	Pressure (bar)
Q_{add}	The heat supplied (kJ/kg)
Q_{av}	The heat available with exhaust gases from gas turbine cycle (kJ/kg)
Q_{cond}	The heat rejected from the condenser (kJ/kg)
P	The net power output of the turbine (MW)
r_p	Compression ratio
T	Temperature (K)
T_1	Compressor inlet air temperature (K)
T_a	The average temperature (K)
T_s	The saturation steam temperature (K)
T_{w1}	The temperature of water entering the economizer (K)
T_{w2}	The temperature of water entering the evaporator (K)
V_f	Specific volume of the water (m3/kg)
W_{Gnet}	The network of the gas turbine (kJ/kg)
W_p	The work of the pump (kJ/kg)
W_{snet}	The work net of the steam turbine cycle (kJ/kg)
Wst	The work of the steam turbine (kJ/kg)
Up_{ccgt}	Uncertainty of the CCGT
W_{Spp}	Least division of the pressure of the superheat steam
W_{rp}	Least division compression ratio
W_{T1}	Least division temperature.
CR	Compression ratio

GREEK SYMBOL

η_t	Turbine efficiency
γ_a	Specific heat ratio of air
γ_g	Specific heat ratio of gases
η_c	Isentropic compressor efficiency
η_{db}	The supplementary firing efficiency
η_m	The mechanical efficiency of the compressor and turbine
η_p	The water pump efficiency
η_{st}	The steam turbine efficiency
η_{th}	The thermal efficiency of the gas turbine.

GLOSSARY OF TERMS

GT	Gas Turbine
ST	Steam Turbine
SG	Steam Generator
HRSG	Heat Recovery Steam Generator
CC	Combustion Chamber
CCPP	Combined Cycle Power Plant
3P	Triple Pressure
TIT	Turbine inlet temperature
PP	Pinch point
DLN	Dry Low Nox
HP	High pressure
IP	Intermediate pressure
LP	Low pressure
CPH	Condensate pre-heater
WHR	Waste heat recovery

CHAPTER – 1

INTRODUCTION

1. INTRODUCTION

Electricity is one of the most vital infrastructure inputs for economic development of a country. The demand of electricity in India has increased rapidly during the last decade. This high electricity requirement is mainly due to the economic development and requirement of better infrastructure. The continuous energy demand could be possible by continuous development in technology & efficient utilization of fuel resources.

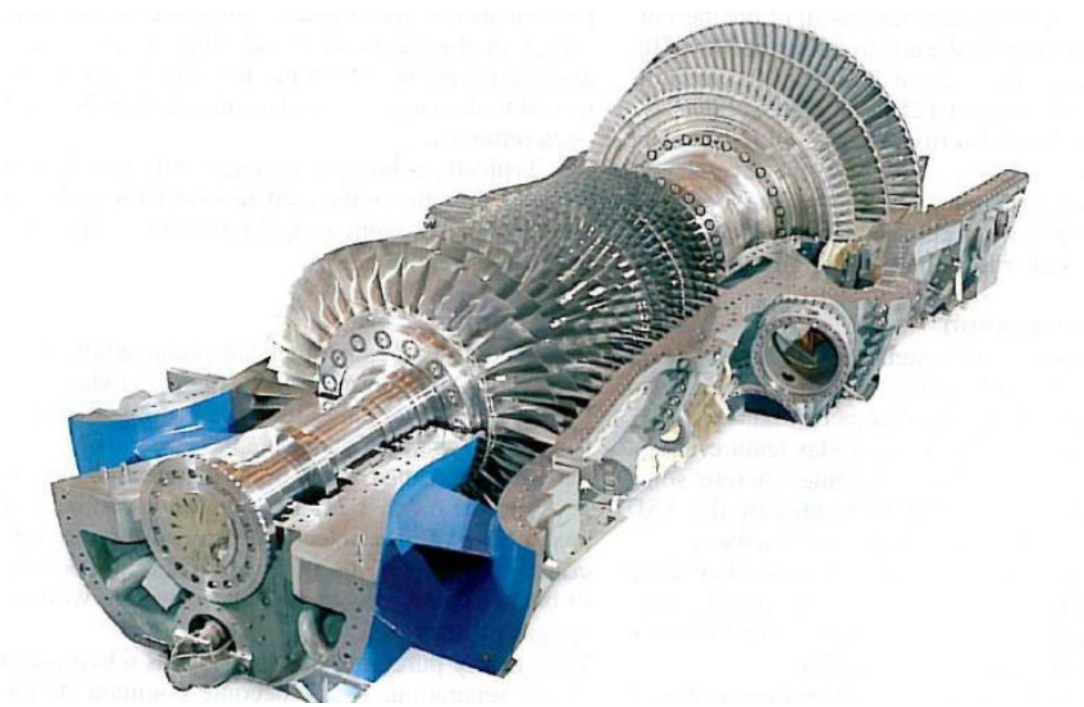


Fig. 1 - Cross Section of Gas Turbine

1.1 Motivation & Objective

The objective of this project is to conduct the optimization of a combined cycle power plant of 220 MW capacity in which natural gas is used as primary fuel. The modernization of technology used in combined cycle power segment moving very rapidly. With the development of technology and improvement in science of Heat Transfer & combustion engineering, promote different variant of Single/Double/Triple Pressure Reheat type with supplementary firing combined cycle power plants to meet the requirement of energy in base as well as peak load hours with economic viability.

1.2 Problem Statement

A GT-Pro Software is being utilized for modelling of Triple pressure reheat type combined cycle power plant with referenced / input design parameters and a relationship is developed to estimate the variation of performance parameters such as main power output, heat rate and plant efficiency w.r.t design point.

The impact of various factors such as such as ambient temperature, Compression ratio, Gas turbine inlet temperature on Brayton cycle & Steam parameters such as pressure & temperature of Rankine cycle has been analysed.

1.3 Expected Outcome

After development of model of 220 MW Triple pressure reheat combined cycle power plant with supplementary firing with GT Pro software, an analysis is being taken to develop a characteristic behaviour of parameters with Plant efficiency, Heat rate and power output etc. of 220 MW module of CCGT (Gas Turbine, Steam Turbine, HRSG).

After that with GT master software the characteristic curve is plotted i.e. variation of plant performance parameter w.r.t. design input

- Effect of Ambient on Power output & Plant efficiency
- Effect of Ambient on Heat rate & Steam rate
- Effect of Compression ratio on Power output & Plant efficiency
- Effect of Compression ratio on Heat rate & Steam rate
- Effect of Gas Turbine Inlet Temperature on Power output & Plant efficiency
- Effect of Gas Turbine Inlet Temperature on Heat rate & Steam rate
- Effect of HP Steam turbine pressure on Power output
- Effect of HP Steam turbine pressure on Plant efficiency
- Effect of Compression ratio on Power output
- Effect of Compression ratio on Plant efficiency
- Effect of HP steam turbine temperature on Power output
- Effect of HP steam turbine temperature on Plant efficiency
- Effect of Gas turbine inlet temperature on Power output
- Effect of Gas turbine inlet temperature on Plant efficiency
- Effect of Steam reheat pressure ratio on Power output

- Effect of Steam reheat pressure ratio on Plant efficiency
- Effect of Deaerator temperature ratio on Power output
- Effect of Deaerator temperature ratio on Plant efficiency
- Effect of pinch point on Power output
- Effect of HRSG high pressure on Power output
- Effect of HRSG intermediate pressure on Power output
- Effect of HRSG low pressure on Power output

1.4 Combined Cycle Power Plant

Due to induction of concept of combined cycle had explored new opportunity in the field of energy sector. Earlier in peak load or in emergency conditions, the gas turbine were used but as technology progressed, the gas turbine starts used for power demand of base load conditions.

The combination of the gas turbine cycle (Brayton cycle) and the steam turbine cycle (Rankine cycle) is designed for improved combined cycle power plants. In Gas turbine cycle the high temperature gas liberated from exhaust of gas turbine for heat energy used in HRSG as prime source of heat energy. There are two fluid termed as Gas & Steam in the cycle functional together. Thus performance of combined cycle depends upon the performance of gas cycle and steam cycle.

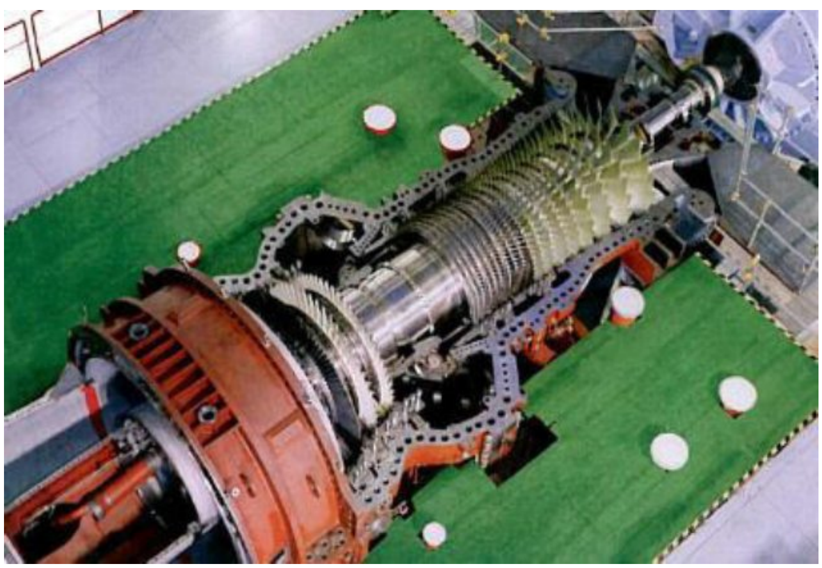


Fig. 2 – Section View of Gas Turbine

The gas and steam turbines are widely accepted to produce electrical power. Both steam and gas turbines have been promptly using in large sector to produce electricity. In this system, gas cycle considered as topping cycle and steam cycle considered as bottoming cycle. As development in advanced cycle, different methodology has been adopted to achieve higher plant efficiency from the turbines, mainly the gas turbine. With widely accepted parameters, the major one is increase the inlet temperature in the gas-turbine and decrease the inlet air temperature of compressor, this way increase the temperature ratio of the peak cycle. As a result of temperature ratio of peak cycle, results high temperature of exhaust gases. Thus many parameters targeted to enhance performance of modelling of Combined Cycle Power Plant. The air and water will act as main fluid for operation of Brayton & Rankine cycle to find reliable, economic power generation with highly efficient manner.

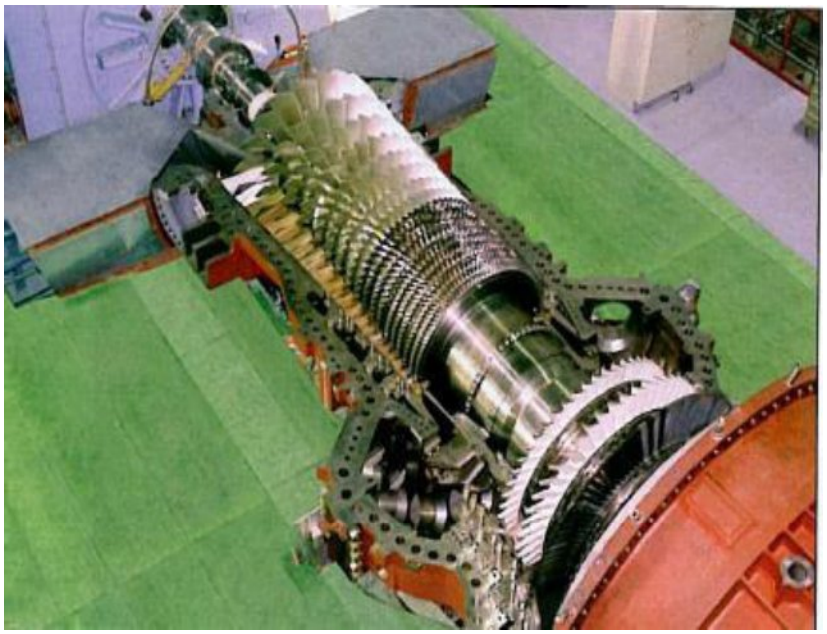


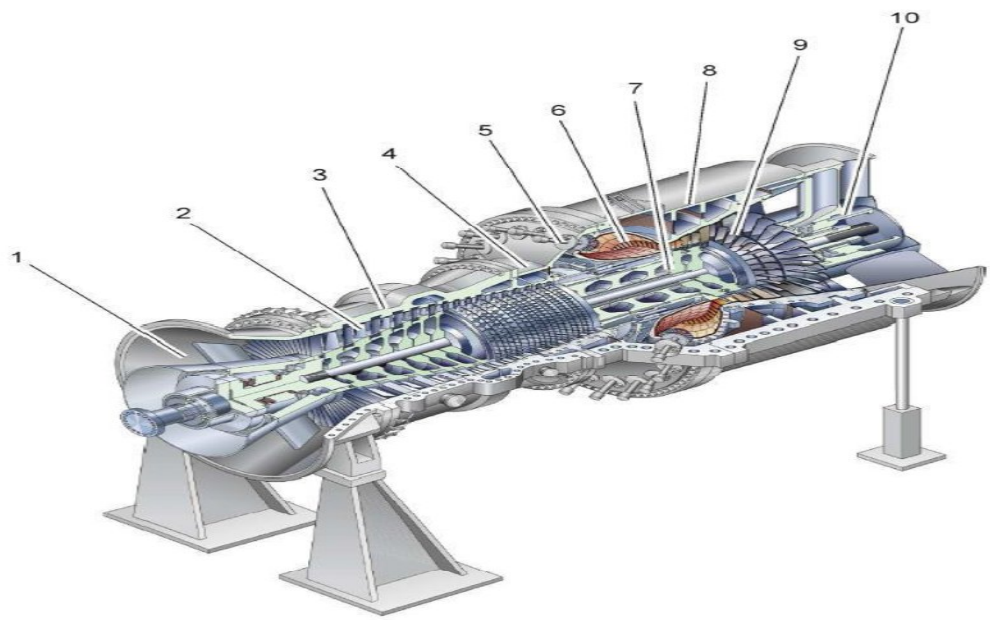
Fig. 3 – Top Sectional View of Gas Turbine

In this scenario, due to technology advancement, the total plant efficiency achieved from CCGT plants touch a figure with lower heating value of 45-55%. However with simple cycle gas turbine it is limited to 35-42%. In opposite to that the CCGT efficiency can be raised upto 60% after incorporating with advanced solution such as higher compression ratio, turbine inlet temperature and ambient condition. Also in Rankine cycle, due to development in steam technology, cause increases in steam turbine cycle temperature and pressure. The improvement in HRSG design and with adoption of newly advanced thermodynamic concept in plant configurations such as two shaft gas turbine, combining intercooler cycle and regenerative cycle may be adopted to improve the plant efficiency.

This Plant Operating Philosophy covers a combined cycle system consisting of a gas turbine, steam turbine, generator and heat recovery steam generator. A power train consisting of one gas turbine, one steam turbine and two generators is called a unit. These procedures, which are typical for a multi-shaft arrangement, may vary with final heat recovery steam generator (HRSG).

1.4.1 Gas Turbine & Its Auxiliary

The gas turbine will be heavy duty, advanced class type each comprising of a multistage axial compressor and a turbine including combustors section. The inlet air system would consist of a filter house with self-cleaning pulse jet type, ducting and silencer. The system would draw atmospheric air into the gas turbine compressor unit.



No.	Designation	No.	Designation
1	Compressor bearing housing	6	Combustion chamber
2	Compressor	7	Rotor
3	Compressor guide vane assembly 1	8	Turbine casing
4	Combustion chamber outer casing	9	Turbine
5	Burners	10	Turbine bearing housing

Fig. 4 – Nomenclature of Gas Turbine

Air intake silencer will suppress the noise in the intake air system. An inlet air guide vane will be turbine will have multiple stages. The exhaust gas from the advance class gas turbines are generally in axial direction of the gas turbine. The gas turbine units will have Dry Low NOx (DLN) combustors suitable for burning natural gas only.

The combustion fuel mixture with air takes place in the combustors and the hot gas will be expanded in the gas turbine, which will drive the generator as well as axial flow air compressor. The gas turbine will have a rated speed of 3000 rpm for direct coupling with generator. The gas turbine generator will be provided with lubrication oil system complete with lube oil pumps, lube oil reservoir, and lube oil coolers. The exhaust system of gas turbine will exhaust the gas into the atmosphere through HRSG. It is general practice with advanced class gas turbines to have a static frequency converter (SFC) to use the generator itself as motor during starting of GT.

1.4.2 Steam Turbine & Auxiliaries

A non-extraction, re-heat, condensing type steam turbine has been considered. The steam entry to the turbine would be through a set of emergency stop and control valves, which would govern the speed / load of the machine. The turbine control system would be of electro-hydraulic type with hydro-mechanical system as a backup. The steam turbine would be complete with lube oil and control oil system, jacking oil system, governing system, protection system and gland sealing steam system. The lube oil system of the STG will be provided with 2x100% online centrifuge system. The gland sealing steam for the steam turbines would be taken from HP steam and will be de-pressurized and de-superheated before supply to turbine glands. The spray water for de-superheating would be taken from IP feed water line. The gland steam header of both power blocks would be interconnected to provide the flexibility during steam turbine start-up and reduce the start-up time.

1.4.3 Condensing Equipment & Auxiliaries

A surface type condenser fixed to the turbine exhaust for condensing the exhaust steam from the steam turbine. The condenser would be of radial or axial or lateral configuration with rigid or spring mounting arrangement. The condenser design will be ensured to prevent sub-cooling of condensate below saturation temperature corresponding to respective condenser backpressure under any of the operating conditions. While deciding the heat duty of the condenser, the heat load during steam dumping will also be considered as one of the operating conditions. Oxygen content of condensate leaving the condenser hot well will be ensured not to exceed 0.03 cc/liter over the entire range of load. A vacuum pumps or steam jet air ejectors will be for maintain the vacuum in the condenser by expelling the non-condensable gases. One vacuum pump would operate during normal working condition and during start-up, both the vacuum pumps may be operate such that, the desired vacuum can be pulled within a shortest possible time. Steam for the ejectors will be supplied from the HP steam header after de-pressurizing and de-superheating. Further, the steam headers of steam jet air ejectors of both the blocks will be interconnected to have the flexibility of operation during start-ups.

1.4.4 Heat Recovery Steam Generator

A heat recovery steam generator (HRSG) carries a vital responsibility for the performance of combined cycle power plant.

It utilizes the thermal energy carried by exhaust gases from the exit of gas turbine. This heat energy is utilized for conversion of feed water in the pressure part of HRSG to steam. The prime mover (steam turbine) is then utilized for extract heat energy from steam (to convert pressure energy to kinetic energy) to convert into mechanical power. HRSG is being utilized for generating steam with required pressure and temperature via extracting sensible heat of gas turbine exhaust for generating.

HRSG as the name designated, is the term mainly for producing steam from cooling the hot gases in the steam generator. The content of heat energy is available free of cost as it is waste gas emitting from exhaust of steam generator, that waste heat is being utilized to increase the efficiency of the combined cycle, via coupled with prime mover of steam turbine with generator. That enhancement in power generation cause cost-effective solution for cycle analysis of CCPP. The incinerator gases, furnace effluent etc can be utilized for regain waste heat available at the exhaust of Gas turbine to HRSG.

As in operation adopted from the heat transfer area could be optimal to extract heat in HRSG, thus as understood the exhaust gases exit in stack should be minimum as possible to avail maximum heat energy to increase the cycle performance. As the name designated in triple pressure HRSG, the performance of CCPP is more compared with single pressure due to more heat extract and intermediate injection of steam in turbine raise the power output. A steam generator composed of number of components such as evaporators, economizers, superheaters, reheater, integral deaerators and preheaters. Different Sections of Heat transfer has been designed for specific task, the selection of material and heat transfer area was based on efficiency requirements & cost of HRSG.

The evaporator section in HRSG used to evaporate the water and generate steam from the section. A bank of finned tubes found in the evaporator section of drum. The feed water initially feed at appropriately pressure to upper part of boiler drum, then feed water circulate in the lower section of drum, then back to upper drum through convection heat transfer in finned part of tube.

The economizers are the section where sensible heat is added to feed water prior entry into evaporator part of the drum. For multi-pressure section, the various heat transfer parts are split in different location of downstream and upstream.

The HRSGs, which considered will be unfired type with horizontal gas, flow, and natural circulation with triple pressure (High, Intermediate and Low pressures) steam generation. The HRSGs will have the dry run capability in order to reduce the black-start power consumption.

Each HRSG will have a separate Superheater, Evaporator and Economizer sections to generate High Pressure (HP), Intermediate Pressure (IP) and Low Pressure (LP) steams. Further, the HRSGs will also have a reheater section where, the cold reheat steam from the HP turbine after integration with IP steam from IP evaporator will be superheated. Steam temperature control at each super heater section will be achieved with spray water attemperation. The spray for attemperators will be tapped-off from HP feed water line.

In each HRSG, a condensate pre-heater (CPH) has been envisaged to recover the thermal energy of the hot gas to the maximum extent. The gas temperature at outlet of CPH is generally governed by dew point temperature of oxides of Sulphur. Though the Sulphur content in the gas is nil, the design exit gas temperature has been limited to 90ºC based on the optimization of the heat transfer area of condensate pre-heater.
It has been envisaged that the Deaerator will be integral part of the HRSG, which will be getting heating steam from the LP evaporator. Vent condenser would be provided with the Deaerator to minimize wastage of steam. The Deaerator will be constant pressure, spray or spray-cum-tray type and will be designed to deaerate all the incoming condensate to keep the oxygen content of the Deaerated condensate below the permissible limit, which generally 0.005 cc/liter and maximum carbon dioxide in Deaerated feed water would be nil. The steam from LP evaporator will be used to peg the Deaerator during plant operation.

Feed water and steam sampling arrangements as required would be provided. Stack height has been arrived to balance the net draft available at stack inlet; however, this will also assist in better dispersion of hot flue gas from HRSG and NOx emission. Steam from the HRSGs would be supplied to a steam turbine through steam piping.
Intermediate-pressure (IP) and Low-pressure (LP) bypass systems of 100% HRSG capacity will be provided for dumping the IP and LP steam to the condenser during startup and turbine trip conditions. During bypass condition, the HP steam will be depressurized and desuperheated to cold reheat steam condition and will be integrated with IP steam before HRSG reheater section. Each bypass station will be provided with pressure reducing valves and attemperators as necessary. The spray water for attemperation would be tapped-off from IP feed water line.

1.4.4.1 Description

As described above, exhaust gas from the gas turbine is passing the heating surfaces of the Heat Recovery Steam Generator (HRSG), where the energy is recovered and superheated steam is generated. Deaerated feedwater is pumped from the feedwater storage tank to the economizer(s) of the HRSG. It has been specific designed for generating high quality superheated steam by pure heat recovery from the exhaust gas leaving a gas turbine. It will be top-supported HRSG with horizontal heating surfaces and vertical gas flow in upward direction.

The water and steam circuit of the HRSG is operated with natural circulation, which reduces the Power consumption and maintenance costs.
Heating surfaces are optimized to achieve the most efficient plant. Heating surfaces are drainable. Due to HRSG casing is made as a fully welded gas tight construction. “Hot” sections are made with corrugated casing. While “Cold” sections are designed with traditional casing made of plain plates with external reinforcement.
The flue gas is passing vertically through the heating surfaces of the HRSG. Heating surfaces are designed with spiral wounded serrated fin tubes in order to obtain maximum plant efficiency. Common for the heating surfaces are the compact bundles of horizontally placed tubes, which are connected to inlet and outlet headers. The advantages of this concept are that tubes can be inspected through access doors and that heating surfaces are drainable. Inspection doors for inspection of the boiler internals are installed between each tube bundle. The heating surfaces will comprise

prefabricated tube banks, each arranged in horizontal loop construction between inlet and outlet headers.

Heating surfaces are designed in such a way that an evenly distributed water and/or steam flow is obtained in order to secure the best and most stable cooling of the heating surfaces under all operating conditions.

The tube diameter and pitching of tube bundles are specifically selected to minimize fouling and permit efficient cleaning. Headers are sized generously, capable of being drained.

1.4.4.2 Steam Drum

The HRSG is provided with HP, IP and LP steam drums, placed on top of the HRSG steel structure. Steam drums are designed in such a way that they can obtain almost all water, displaced by the evaporator systems during start-up. This design minimizes the amount of blowdown at start-up. In addition, steam-purifying equipment, such as cyclones and chevrons, is provided to minimize carry-over and achieve a steam quality.

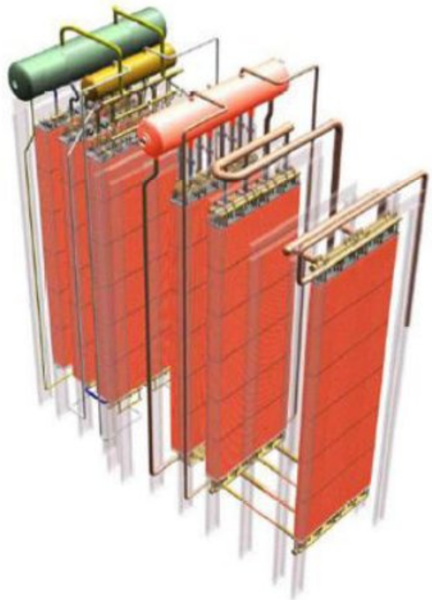
Internal fittings are designed so that no displacement can occur while the unit is in service, but such fittings are not welded directly to the internal surface of the drums. Nozzles on the HP steam drum are set-on type with full penetration. Steam drums are provided with access door and non-illuminated water level gauge.

1.4.4.3 Water and Steam Circulation

Deaerated feedwater is pumped, by others, from the feedwater storage tank to the economizers, where the feedwater temperature increases to the saturation temperature. From the economizers the water is led to the steam drums via internal distribution pipes designed to ensure good mixing of feedwater and boiler water.

The boiler water is circulating naturally between the steam drums and evaporators. Boiler water is led to the inlet of the evaporators by unheated downcomers. Steaming occurs in the evaporator tubes. The water and steam mixture is led to the steam drums by riser tubes. Saturated steam separated in the drum by special steam purifying equipment (chevrons & cyclones), is drawn off at the apex of the steam drum and directed to the superheaters. The HP/IP/LP superheaters are divided based on GT type and operational characteristics.

A spray type desuperheater is placed between or after the HP/LP superheater sections to control the HP/LP steam outlet temperature Drain and vent pipes are included, which together with the design makes it possible to drain and vent the boiler pressure part. The drain line that's drains the superheater and pipeline upstream the main steam stop valve is provided with a motorized drain valve.



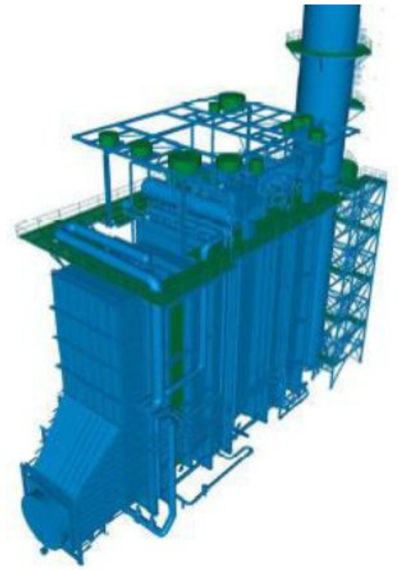


Fig. 5 – Pressure part of HRSG

The HP/IP/LP superheaters are divided based on GT type and operational characteristics. A spray type desuperheater is placed between or after the HP/LP superheater sections to control the HP/LP steam outlet temperature Drain and vent pipes are included, which together with the design makes it possible to drain and vent the boiler pressure part. The drain line that's drains the superheater and pipeline upstream the main steam stop valve is provided with a motorized drain valve.

1.4.4.4 Exhaust Gas System Upstream & Downstream HRSG

HRSG inlet duct - A duct is installed upstream the HRSG. The duct is made of corrugated steel plates without external reinforcement. The duct is designed for being externally insulated. The duct is provided with one access door at bottom level. The duct is fixed to the HRSG and expands against ground, as the HRSG is top supported. An expansion bellow is installed at the duct inlet in order to accommodate thermal expansions. The "body" of the bellow is made in steel. The gab that shall obtain the movements is sealed with mineral wool and covered externally by a suitable fabric.

1.4.4.5 Desuperheater

Included is one desuperheater (spray atomizing type) to control the HP steam outlet temperature. The spray nozzle is designed to secure proper evaporation of the spray medium. Boiler feedwater is used as spray medium.

1.4.4.6 Blowdown System

Continuous blowdown: The function of the continuous TDS blowdown valve(s) is to secure the correct amount of blow down automatically controlled by the TDS level in the steam drum(s). The blowdown valve(s) has a capacity equal to approx. 3% blowdown.

1.4.4.7 Blowdown Vessel

Included is a blowdown vessel for collecting the different drains, etc. The tank is designed as a non-pressurized vessel. It is provided with external lagging and cladding for personnel protection.

1.4.5 Description of the Triple-Pressure Reheat Combined Cycle

The Working of Triple pressure-Reheat with supplementary fired combined cycle is summarised below:

Figure 6 shows a schematic diagram of the combined cycle power plant with three pressure reheat type GT cycle with consideration, in which Turbine inlet temperature is 1322°C & 220 MW power generation taken place.

Initially air after filtration in intake air filter at 1 is compressed to a higher pressure (HP) at 2, from where the air enters the combustion chamber (CC), the fuel (natural gas) is supplied in combustion chamber, and the compressed air is added with fuel, resulting in a combustion gas at 3. The high energy gas at 3 expands in the GT, to chimney or HRSG at 4.

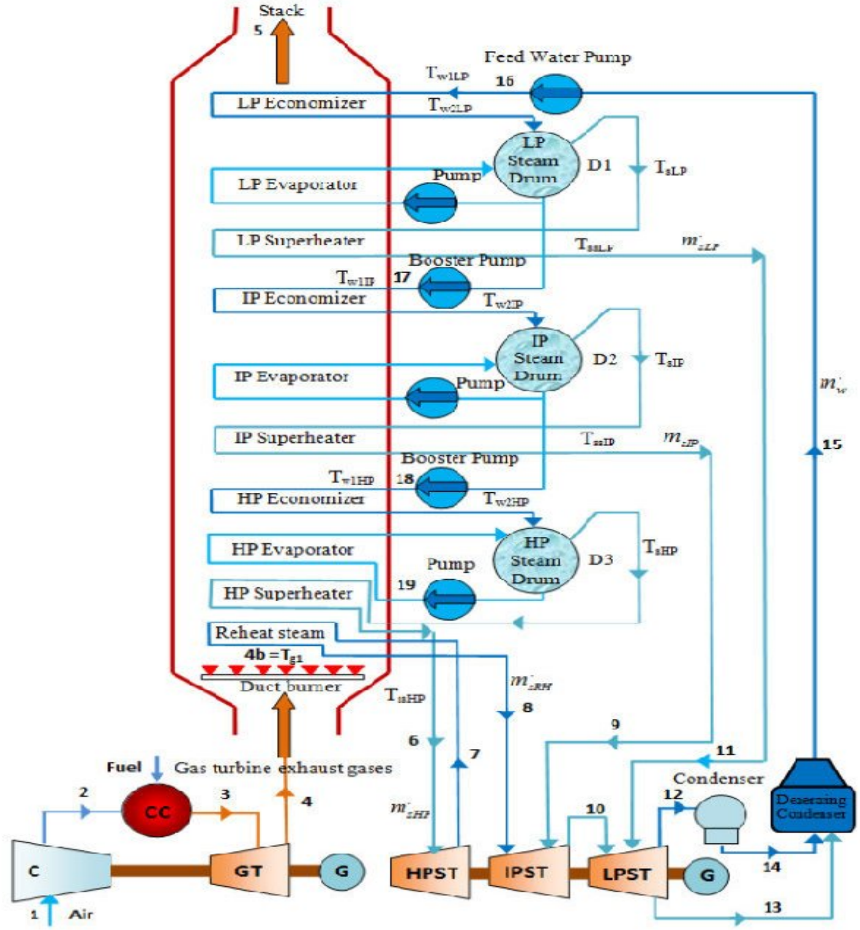
The exhaust gas from the outlet of Gas Turbine at 4 enters the HRSG to transfer heat to steam and exits at the stack temperature at 5. From HRSG when feed water converted into steam, outlet of HP superheater at 6 expands in the HP steam turbine (HPST) to a lower pressure (LP) and temperature at 7. From the outlet of HP steam turbine, Steam at 7 is reheated in the reheat (RH) section to a higher temperature at 8 where steam expands further in the intermediate-pressure (IP) drum of steam turbine (IPST) to the LP at 10. The superheated steam at the outlet of IP section of the HRSG at 9 expands in the IPST to a LP and temperature at 10 where enter to LP steam turbine (LPST). The superheated steam at the outlet of LP section of the HRSG at 11 and steam at 10 expands further in the LPST to LP and temperature at 12.

Steam is extracted from the LPST and fed to the open feed water heater (deaerator) at 13. The steams with LP and low-temperature at 12 will condensate in the condenser at 14 to saturated water. The saturated water out of the condenser at 14 mixed with the steam at 13 inside the deaerating condenser, the resulting water at 15. The saturated water out of the deaerating condenser at 15 is pumped to a HP at (Tw1LP). The water at (Tw1LP) is heated in LP economizer section of the HRSG, resulting in saturated water at (Tw2LP) where the water enters the LP steam drum (D1). The saturated steam at the outlet of drum D1 at (TsLP) is superheated in LP superheater of the HRSG, resulting in superheated steam at (TssLP).

Steam at (TssLP) enters the LPST where the steam expanded to the condenser pressure at 14. The saturated water at the outlet of drum D1 is pumped to the pressure of drum D2 at (Tw1IP). The water at (Tw1IP) is heated in intermediate-pressure economizer section of the HRSG to the saturated water condition. The saturated water at (Tw2IP) is heated and partially evaporated in IP evaporator section of the HRSG.

The saturated vapor at the top of drum D2 at (TsIP) is superheated to a higher temperature at (TssIP) in intermediate-pressure superheater section of the HRSG. Steam at (TssIP) enters the IPST where the steam expanded to the condenser pressure at 14. The saturated water at the outlet of drum D2 is pumped to the pressure of drum D3 at (Tw1HP).

The water at (Tw1HP) is heated in HP economizer section of the HRSG to the saturated water condition. The saturated water at (Tw2HP) is heated and partially evaporated in HP section of the HRSG. The saturated vapor at the top of drum D3 at (TsHP) is superheated to a higher temperature at (TssHP) in HP superheater section of the HRSG. The superheated steam at (TssHP) enters the HPST where the steam expanded to the reheat section at 7. The steam at 7 is superheated to a higher temperature at 8 in reheat section of the HRSG with effect the duct burner. All the steam at 12 will condensate in the condenser to water at 14, and then pumped to 16.



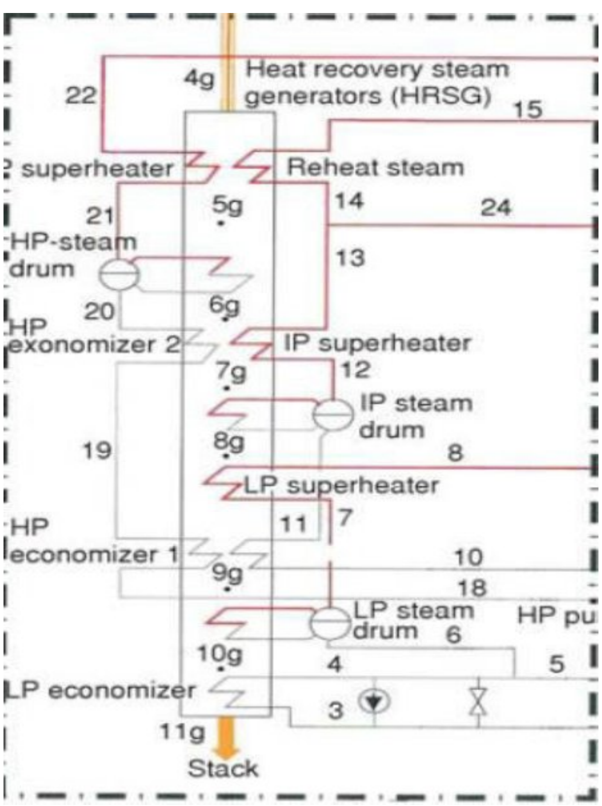


Fig. 6 – A schematic diagram of the supplementary firing triple-pressure steam-reheat combined cycle power plant [6] & [1]

CHAPTER – 2

LITERATURE REVIEW

2. LITERATURE REVIEW

Some recent work with similar nature of Projects is summarized below:-

Alus and Petrovic (2012) conducted a simulation on the triple pressure combined cycle power plant (CCPP) & the production cost of electricity was minimizing during optimization process of CCPP power plant based on economic analysis.

Ghazl (2012) carried out an simulation study to optimize the best design parameters (pinch point temperature difference, high and low degree pressures, the duct burner fuel consumption flow rate and steam mass flow rates) of a dual pressure combined cycle power by using objective function in terms of Total cost per unit of produced steam of power plant.

Zhixin (2012) formulate a condition of a single pressure waste heat recovery system when the flow rate of exhaust gas or temperature fluctuates, by using system optimization for generating more power at the upper boundary of fluctuation range of exhaust gas.

Mohammad (2012) performed an economic & exergetic evaluation of HRSG configurations on the system performance of combined cycle power plants by using exergy destruction rate in HRSG & observed increase in the number of pressure levels.

Manassaldi (2011) developed non-linear function model by considering net heat transfer maximization of HRSG of combined cycle power plant.

Srinivas (2009) formulate the location of a deaerator in a triple pressure HRSG of combined cycle power plant.

Mohagheghi and Shayegan (2009) is also developed computer code by formulating thermodynamic analysis & optimization of the HRSG to obtain power in the steam cycle.

Som and Datta (2008) performed exergy & irreversibility analysis for combined cycle power plant for the gaseous, liquid and solid fuels.

Polyzakis et al. (2008) work contains an optimized three thermodynamic cycle i.e. Gas Turbine cycle, intercooled cycle, Reheat and intercooled cycle and found reheat cycle considered as optimum Gas Turbine cycle in the CCPP.

Borelli and Junior (2008) have studied the exergy-based method for analyzing the composition of the electricity cost generated in gas-fired combined cycle plants. They proposed a method to analyze the composition of the cost of electricity is based on the energy conversion processes and the destruction of the exergy through the several thermodynamic processes that comprise a combined cycle power plant.

Ameri et al. (2008) have modeled exergy analysis of a 420MW CCPP and calculated irreversibility of each components of Neka CCPP. The results show that the HRSG, Gas turbine, Combustion chamber, and duct burner are the main components of irreversibility & result shows 83% of exergy losses.

Sengupta et al. (2007) have studied the exergy analysis of a coal-based 210 MW thermal power plant. It helps to find out the contributions of different parts of the plant towards exergy destruction. The exergy efficiency is calculated using the operating data from the plant at different conditions, viz. at different loads, different condenser pressures, with and without regenerative heaters and with different settings of the turbine governing.

Butcher and Reddy (2007) modeled a WHR Power plant based on second law analysis.

Khaliq and Choudhary (2007) have studied the combined first and second-law analysis of gas turbine cogeneration system with inlet air cooling and evaporative aftercooling of the compressor discharge. They performed Computational analysis to investigate the effects of the overall pressure ratio, turbine inlet temperature, and ambient relative humidity on the exergy destruction in each component, first-law efficiency, power-to-heat ratio, and second-law efficiency of the cycle. They found that the thermodynamic analysis indicates that exergy destruction in various components of the cogeneration cycle is significantly affected by overall pressure ratio and turbine inlet temperature, and not at all affected by the ambient relative humidity.

Xiang and Chen (2007) carried out a thermodynamic analysis for a triple pressure HRSG to optimize the combined cycle system.

Sanjay et al. (2007) have performed research work on energy and exergy analysis of steam cooled reheat gas steam combined cycle. They have compared the blade cooling techniques and found that closed-loop-steam-cooling to be superior to air-film cooling. The reheat gas–steam combined cycle plant with closed-loop-steam-cooling exhibits enhanced thermal efficiency (around 62%) and plant specific work as compared to basic steam–gas combined cycle with air-film cooling as well as closed-loop-steam cooling.

Butcher and Reddy (2007) analyzed Second law analysis of a waste heat recovery based power generation system and improvement the performance of a waste heat recovery power generation system for various operating conditions. The temperature profiles across the heat recovery steam generator (HRSG), network output, second law efficiency and entropy generation number are simulated for various operating conditions.

Bassily (2007) modeled combined cycle power plant to reduce the irreversibility of the HRSG and compared the optimized results with triple pressure reheat combined cycle.

Dellenback (2006) studied a reassessment of a alternative regenerative cycle and found the alternative regenerative cycle in more efficient than conventional cycle & optimized the pressure ratio at all points for maximum possible thermal efficiency.

Kumar and Krishna (2006b) performed the first & second law analysis of Gas turbine power plant with alternative regeneration and compares it with conventional regeneration configuration and found that alternative regeneration proved to be efficient than the conventional configuration.

Bassily (2005) presented the effects of varying the inlet temperature of the gas turbine & PP on the performance of a dual pressure reheat combined cycle.

Korakianitis (2005) performed analysis of combined cogeneration power plant for efficiency enhancement. With computing the exergy and exergy destruction at different points of combined cogeneration plant and optimized thermal performance by enhancing efficiency; power output, plant effectiveness and exergy rate.

Yadav (2005) analyzed Gas & steam combined cycle power plant for different type of stage cooling. Steam coolant is bled from heat recovery steam generator. Influence of different type of coolant upon the performance of topping, bottoming, combined cycle and HRSG has been analyzed and improvement thermal performance.

Ramaprabhu and Roy (2004) developed a computational model of a combined cycle power generation unit and developed thermodynamic model to a plant and compare the simulation results with available plant test data at rated load for evaluating thermal performance.

Khaliq and Kaushik (2004a) carried second-law analysis of the combined power-cycle with reheat and compared the exergy destruction in the components of the combined cycle and with the effects of pressure ratio, temperature ratio and number of reheat stages on the cycle performance.

Khaliq and Kaushik (2004b) presented systematic methodology based on first and second law for the thermodynamic performance evaluation, of combustion gas turbine cogeneration system with reheat. He considered the cases of combustion gas turbine cogeneration for analysing the energetic and exergic performance and concluded reheat expansions gives significant improvement in first- and second-law efficiencies.

Valdes (2003) performed a thermos-economic optimization of combined cycle gas turbine power plants using a genetic algorithm. They proposed two different objective functions; aimed at minimizing the cost of production per unit electricity and maximizing the annual cash flow.

Reddy et al (2002) carried out exergy analysis for WHR boiler in thermal power plant by minimizing entropy of thermal plant.

Casarosa (2002) proposed an analysis of combined cycle power plant to increase the plant efficiency and evaluate 60 percent efficiency without resorting to a new gas turbine technology.

Attala (2001) optimize a dual pressure level combined cycle gas turbine & developed a simulation programme which include three models & input simulates the cycle, then evaluates the thermodynamic and thermo-economic parameters of optimization model of combined cycle.

Felster et al. (2001) analysed the effect the dual pressure and triple pressure with & without reheat and compressor air intercooling, water injection and reheating of combined cycle power plant.

Facchini (2000) performed the detailed study of reheat gas turbine/combined cycle and close loop steam gas cooled gas turbine. By using detailed exergy balance to compare the performance of plant sections to improve plant efficiency.

Ali (1997) carried out studied on simple Gas turbine system and analysed different cycle with evaporative cooling and adding to improve overall cycle.

Ongiro et al. (1997) developed a numerical method to predict the thermal performance of the HRSG system for keeping computational complexity manageable.

Moran (1996) established a design methodology for Gas turbine cogeneration system. This simple system is integrated with Regenerator and a HRSG is attached to utilize the waste heat. A 30 MW Gas turbine is designed on the basis of analysis of enthalpy and entropy of air and gas. Methane has been used as fuel. Simultaneously, the exergy and energy destruction at different points are also tabulated. The waste heat from turbine (exhaust gases) has been used to produce steam for other processes. Steam is formed at 20 bar and 14 kg/s, which is a assumption to design heat recovery steam generator.

Dock (1996) carried out the exergy analysis of Gas turbine cogeneration system and examined second law. The performance of Gas turbine cogeneration systems in terms of exergy destruction in each component for part and full load conditions. In addition, the effect of inlet air temperature, humidity of the inlet air, water and steam injection on the performance of the system is analyzed.

Subrahmanyam et al (1995) discussed about the various factors affecting the HRSG design for achieving the highest combined cycle efficiency with cheaper, economical and competitive designs and with the highest requirement to meet the shorter deliveries.

Pasha & Sanjeev (1995) studied for different type of circulation for the design of HRSG and its importance for the performance analysis.

Horlock (1995) based on thermodynamic considerations, outlined developments of 1970s, 1980s and future prospects of combined-cycle power plants. The main focus was on (i) raising the mean temperature of heat supply; (ii) minimizing the irreversibility within the heat recovery steam generator; (iii) keeping the heat loss between the two plants as low as possible. The optimum pressure ratio for the combined plant (CR=18) is less than that for the gas turbine alone (CR=30) although it is still greater than the pressure ratio, which gives maximum specific work in the higher plant (CR=11).

Subhramanyam (1995) considered the computational complexity of the HRSG and how it could be model. The proposed method calculates the velocity and temperature fields by discretisation and the solution of conservation equation derived for a HRSG of particular geometry and duty.

Najjar and Akyurt (1994) reviewed various types of combined cycles, including repowering, integrated gasification and other advanced systems. According to this study: 1). Combined cycles boost power output and efficiency to levels that are considerably above those of steam power plants 2). Repowering, when converting an existing steam plant to combined cycle, offers savings in capital cost as compared to new construction 3). Combined cycle, when integrated with coal gasification, holds promise in converting coal into electric power in an efficient, economical and environmentally acceptable manner.

Bruckner and Emsperger (1989) studied the retrofitting of a conventional steam power plant by adding the gas turbine, where good fuel efficiency, low power generation cost and reduction in pollutants emissions are major factors.

Takeya and Yasui (1988) presented the performance of combined cycle with reheat gas turbine with an intercooler and water spray type heat exchanger, coupled to dual pressure and triple pressure reheat waste heat recovery boiler and found plant efficiency is increased upto 54%.

Cerri and Sciubba (1987) A parametric for a plant equipped with a gas generator and with steam injection into an after-burner placed upstream of the power turbine. The steam was to be produced by a waste-heat recovery section made up of a boiler and distillation plant fed by the gas turbine exhaust. The results showed a 13% improvement in plant efficiency and a double the specific work output when compared with a standard gas turbine cycle with full reheat and optimal steam injection.

CHAPTER – 3

MATHEMATICAL MODELLING

3. MATHEMATICAL MODELLING

GAS TURBINE MODEL

The network of the GT (W_{Gnet}) is calculated from the equations:

$$W_{Gnet} = C_{pg} * TIT * \eta_t \left(1 - \frac{1}{r_p^{\left(\frac{\gamma_g - 1}{\gamma_g}\right)}} \right) - C_{pa} * T_1 \left(\frac{r_p^{\left(\frac{\gamma_a - 1}{\gamma_a}\right)}}{\eta_m \eta_c} \right)$$

Where C_{pa}, is the specific heat of air which can be fitted below mentioned eq for the range of 200K<T<800K & is the mech. efficiency of the compressor and turbine:

$$C_{pa} = 1.0189 * 10^3 - 0.13784 T_a + 1.9843 * 10^{-4} T_a^2 + 4.2399 * 10^{-7} T_a^3 - 3.7632 * 10^{-10} T_a^4$$

Where, $T_a = \left(\frac{T_2 - T_1}{2}\right)$ in Kelvin

The specific heat of flue gas (C_{pg}) is given by

$$C_{pg} = 1.8083 - 2.3127 * 10^{-3} T + 4.045 * 10^{-6} T^2 - 1.7363 * 10^{-9} T^3$$

The Output power from the turbine (P) is expressed as below :

$$\mathrm{P} = \dot{\mathrm{m}}_{\mathrm{g}} * \mathrm{W}_{\mathrm{Gnet}}$$

Where ṁg is the mass flow of the exhaust gases through the GT, and it expressed as below :

$$\dot{\mathrm{m}}_g = \dot{\mathrm{m}}_a + \dot{\mathrm{m}}_f$$

The specific fuel consumption (SFC) is determined by below mentioned equation

$$SFC = \left(\frac{3600 f}{W_{Gnet}}\right)$$

The heat supplied is also expressed as:

$$Q_{add} = C_{pgm} * \left[TIT - T_1 * \left(1 + \frac{r_p^{\left(\frac{\gamma_a - 1}{\gamma_a}\right)} - 1}{\eta_c} \right)\right]$$

The GT efficiency (η_{th}) can be determined by eq mentioned below η_{th} :

$$\eta_{th} = \frac{W_{Gnet}}{Q_{add}}$$

STEAM TURBINE MODEL

By performing the energy balance for ST as shown in fig.6, the following relation is obtained:

$$W_{ST} = \dot{m}_{sHP}h_6 - \dot{m}_{sHP}h_7 + \dot{m}_{sRH}h_8 + \dot{m}_{sIP}h_9 + \dot{m}_{sLP}h_7 - \dot{m}_w h_{12}$$

Where m_w is the water mass flow rate and it is determined from equation below:

$$\dot{m}_w = \dot{m}_{sHP} + \dot{m}_{sIP} + \dot{m}_{sLP}$$

CONDENSER MODEL

The heat reject from condenser is:

$$Q_{cond} = \dot{m}_w(h_{12} + h_{14})$$

DUCT BURNER MODEL

The additional fuel is burn in the HRSG section known as Duct burner. This is designed to increase the temperature of gases in the heat transfer zone of the HRSG.

$$\begin{aligned} &\dot{m}_4 * C_{PG4} * T_4 + \dot{m}_{fdb} * LHV \\ &= \left(\dot{m}_4 + \dot{m}_{fdb}\right) * C_{pg4b} * T_{4b} + (1 - \eta_{db}) * \dot{m}_{fdb} * LHV \end{aligned}$$

Where η_{db} is the duct burner efficiency, considered as 93%.

$$\dot{m}_g = \dot{m}_4 + \dot{m}_{fdb}$$

$$T_{g1} = T_{4b}$$

HERAT RECOVERY STEAM GENERATOR MODEL

The analysis of HRSG unit is based on the pinch and approach point. The pinch point is simply define as, the difference between the saturation temperature of water and the gas temperature prior to entry to the economizer, which is denoted as T_{pp} and T_{ap}, as shown in Fig. 6. The greater the pinch point, the smaller the surface area available for thermal energy transfer from the hot exhaust gas to the steam. Therefore, this would result in a higher HRSG exit temperature.

Approach point is used in the sizing of the economizer. As demonstrated in Figure 6 the approach point is the difference between the economizer water outlet temperature (T_{w2}) and the saturation temperature (Ts) of the steam. This temperature will guarantee that no steaming will occur in the economizer section.

From fig. 6 it is clear that the energy balance for the high-pressure steam evaporator side of the HRSG gives:

$$T_{g4} = T_{sHP} + T_{pp}$$

$$T_{w2HP} = T_{sHP} + T_{ap}$$

The temperature of the exhaust gases exit from the low-pressure economizer of the HRSG could be found by considering the energy balance of the low-pressure economizer fig. 6 yields;

$$T_{g11} = \frac{C_{pg10}T_{g10}}{C_{pg11}} - \frac{\dot{m}_{sLP}(h_{sLP} - h_{w2LP})}{\dot{m}_g C_{pg11} h_{1f}}$$

Heat available with exhaust gases from GT calculated as

$$Q_{av} = \dot{m}_g (C_{pg1}T_{g1} - C_{pg11}T_{g11}) h_{1f}$$

After the thermodynamics properties of steam–water in all steam cycle points have been calculated, the mass flow rate of steam generation in the HRSG can be determined by applying the energy balance for flow at each single pressure.

Energy Balance equation for the HP part of HRSG

$$\dot{m}_{GT,out} \eta_{HRSG} (h_{4g} - h_{6g}) = \dot{m}_{ST,HP}[(h_{22} - h_{20}) + (h_{15} - h_{24})] + \dot{m}_{ST,IP}(h_{15} - h_{13})$$

Energy Balance equation for the IP part of HRSG

$$\dot{m}_{GT,out} \eta_{HRSG} (h_{6g} - h_{8g}) = \dot{m}_{ST,HP}(h_{20} - h_{19)} + \dot{m}_{ST,IP}(h_{13} - h_{11})$$

Energy Balance equation for the LP part of HRSG

$$\dot{m}_{GT,out} \eta_{HRSG} (h_{8g} - h_{10g}) = \dot{m}_{ST,HP}(h_{19} - h_{18)} + \dot{m}_{ST,IP}(h_{11} - h_{10}) + \dot{m}_{ST,LP}(h_8 - h_6)$$

Mass Balance Equation for the HRSG

$$\dot{m}_{ST} = \dot{m}_{ST,HP} + \dot{m}_{ST,IP} + \dot{m}_{ST,LP}$$

Temperature of Gas Leaving Section 1 of the HRSG

$$h_{5g} = h_{4g} - \frac{\dot{m}_{ST,HP}[(h_{22} - h_{21}) + (h_{15} - h_{13})] + \dot{m}_{ST,IP}(h_{15} - h_{24})}{\dot{m}_{GT,out} \eta_{HRSG}}$$

Calculation of the Heat Transfer Area

A counter flow type heat exchanger has been considered for modelling with HRSG. The optimal area selection “A” of HRSG was calculated based upon value of PP. We hereby assumed that pinch point is equal for all three sections of pressure parts.

The temperature of the gas entering the LP economizer, IP economizer, and HP economizer given as:

$$T_{6g} = T_{20} - PP$$

$$T_{8g} = T_{11} - PP$$

$$T_{10g} = T_6 - PP$$

After calculating the thermodynamic property of steam-water, the mass flow arte of steam generation in HRSG determined from energy balance for flow at each single pressure.

Now, the heat transferred through each area of HRSG is given by:

$$\dot{Q}_i = U_{m,i} A_i LMTD_i$$

Where, A refers to the heat exchange area and U_m is average overall heat transfer coefficient. The log mean temperature difference (LMTD) is calculated as:

$$LMTD_i = \frac{\Delta T_{1,i} - \Delta T_{2,i}}{\ln \frac{\Delta T_{1,i}}{\Delta T_{2,i}}}$$

Where, ΔT_1 is the temperature difference between gas and steam at the inlet of the heater and ΔT_2 - the temperature difference between gas and steam at the exit of the heater.

The HRSG net heat transfer area is calculated as sum of the necessary heat transfer area for each steam pressure level:

$$A_{HRSG} = \sum_E A_E + \sum_V A_V + \sum_{SH} A_{SH} + \sum_{RE} A_{RE}$$

The condenser heat transfer area A_{cond} was calculated according to:

$$A_{cond} = \frac{\dot{Q}_{cond}}{U_{cond} LMTD_{cond}}$$

Where, Q_{cond} is the heat transferred and U_{cond} – the heat transfer coefficient in condenser.

The condenser log mean temperature difference, $LMTD_{cond}$, is defined as:

$$LMTD_{cond} = \frac{TR}{\ln \frac{1}{1 - \frac{TR}{ITD}}}$$

Where, TR is the temperature rise of cooling water in the condenser and ITD – the difference between the steam temperature and cooling water inlet temperature (the initial temperature difference).

PUMP MODEL

The condensate water from the condenser is extracted by the pump and is raised to the economizer pressure. The corresponding work for 3 pressure levels is given by;

$$W_p = \dot{m}_w v_{f16}(p_{shLP} - p_c) + (\dot{m}_w - \dot{m}_{sLP}) v_{f17}(p_{shIP} - p_{shLP}) + (\dot{m}_w - \dot{m}_{sIP} - \dot{m}_{sLP}) v_{f18}(p_{shHP} - p_{shIP})$$

Therefore the net-work for the ST power plant is:

$$W_{net} = W_{st} - W_p$$

The performances of combined cycle gas turbine power plant, including the thermal efficiencies for GT cycle, ST cycle, and overall efficiency are calculated as given below respectively [2]:

The efficiency for the steam turbine power plant is;

$$\eta_{st} = \frac{W_{snet}}{Q_{av}}$$

The overall plant efficiency for the combined cycle power plant is:

$$\eta_{all} = \frac{3W_{Gnet} + W_{snet}}{3Q_{add}}$$

The overall cycle efficiency can be obtained from the steam and gas turbine powers, which are dependent not only on the HRSG area, but also on many other variables such as condenser pressure, pressure drum, and ambient temperature:

$$\eta_{CCGT} = \frac{W_{ST}(\dot{m}_{ST}, h_{cond}, \eta_{ST}) + W_{GT}}{\dot{m}_f H_t}$$

The total heat rate is:

$$HR_t = \frac{3600}{\eta_{all}}$$

CHAPTER – 4

INPUT DATA

4. INPUT DATA

Table 1 : Natural Gas Characteristic

Composition	Volume
N_2	73.54%
O_2	12.54%
CO_2	3.717%
H_2O	9.316%
Ar	0.8856%
Total	100%
Fuel gas temp. @ FG1, base load	46°C
Fuel Low Heating Value	413009 kWth

Table 2 : Design Input for CCGT

Atmospheric condition	1.01325 bar & 25°C
Gas cycle pressure ratio	14.7
Turbine Inlet temperature	1322°C
Inlet pressure HP steam turbine	110 bar
Inlet temperature HP steam turbine	565°C
Approach point (AP) in LP and HP economizers	15°C
Condenser pressure	0.104 bar
Steam reheat pressure	25% of HP pressure
Pinch points in HP, IP, and LP evaporators (minimum temperature difference between the flue gas and the saturated steam)	15°C
TTD in the HP, IP, and LP super heaters (temperature difference between flue gas and superheated steam)	20°C
Degree of superheat (DSH) in superheater	50°C
Polytrophic stage efficiency for compressor and gas turbine	85%
Isentropic efficiency of Compressor & GT are range	75% to 95%
Isentropic efficiency of steam turbine	90%
Generator efficiency	95%
Pressure drop in combustion chamber Pressure drop in HRSG, deaerator and condenser	Neglected

Heat loss in HRSG turbine condenser, and deaerator	
All processes	Steady state & steady flow
Fuel type	Natural Gas

CHAPTER – 5

THERMODYNAMIC MODELLING PROCEDURE

5. THERMODYNAMIC MODELLING PROCEDURE

5.1 Details of Software (GT Pro)

GT PRO automates the process of designing a combined cycle plant. GT PRO is particularly effective for creating new designs and finding their optimal configuration and design parameters.

The user inputs design criteria, assumptions and the program computes heat and mass balance, system performance, and component sizing. Most key inputs are automatically created by intelligent design procedures that help the user identify the best design with minimal time and effort, while allowing the flexibility to make any changes or adjustments.

GT PRO is truly easy to use, typically requiring only a few minutes to create a new plant design. It normally computes a heat balance and simultaneously designs the required equipment in under minimal time requirement.

When run in conjunction with the optional GT Pro module, the programs provide extensive engineering and cost estimation details.

The variety of gas turbine based power plant configurations which could design in GT PRO is virtually endless. From gas turbines in simple cycle, to engines exhausting into waste heat recovery boilers for process steam, & condensing, non-condensing, or reheat steam turbines, with any sort of cooling system, are all easily accommodated in GT PRO. The following is a small sampling of the variety of plants which can generate.

5.2 Procedures

To develop the model of 220 MW Triple pressure, reheat type with supplementary fired combined cycle power plant, the following minimal input parameters are being feed. Steps are follows:

1. Initial select the basic configuration of the plant i.e. Approx. Plant output in terms of MW, type of primary fuel (as Natural Gas in our case) for Gas turbine, Plant configuration such as GT, HRSG, Condensing reheat Steam Turbine etc. A table for design input parameters will be ready while filling the data in Programed module of the GT Pro software.

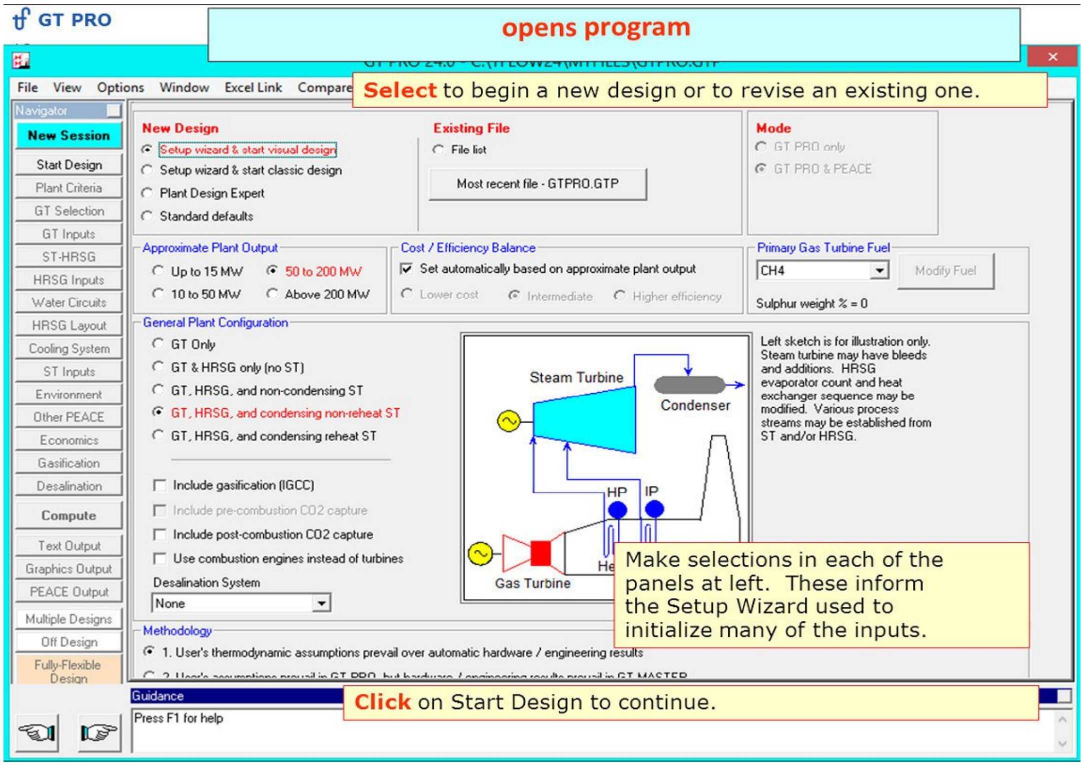


Fig. 7 – Selection of basic configuration CCPP model

2. Then select the basic configuration of the plant i.e. 3P HRSG Evaporator, LP evaporator feed to deaerator etc in the software module.

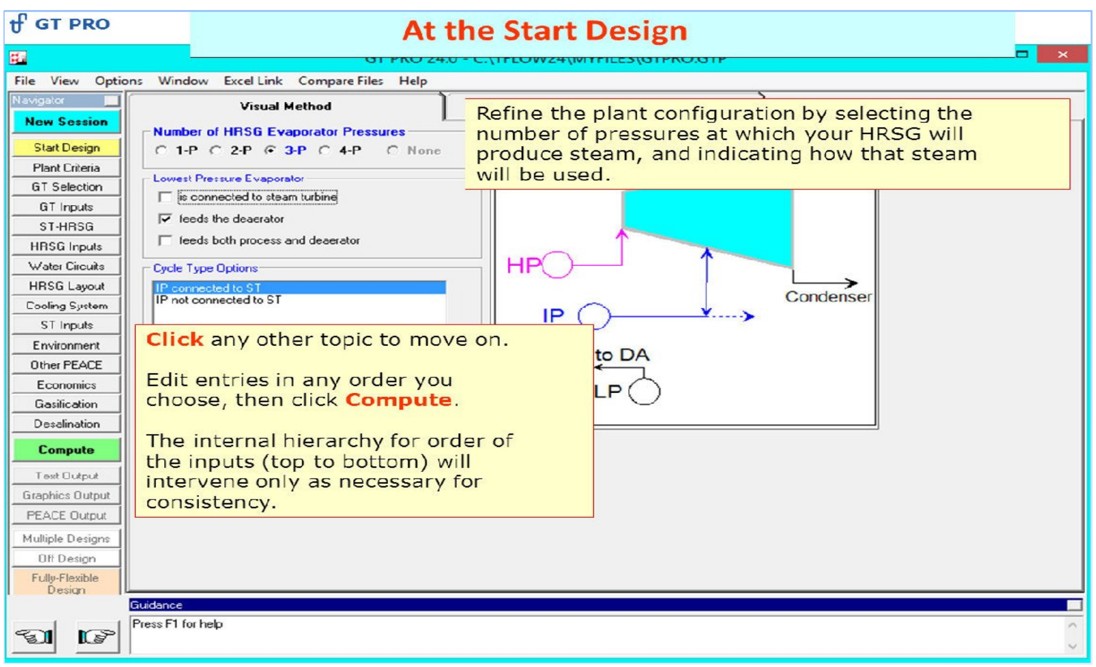


Fig. 8 – Selection of HRSG model of CCPP

3. Then Ambient conditions such as Wet bulb temp, relative humidity, Altitude, frequency of the plant, Water cooled condenser etc will be in next scrolled page.

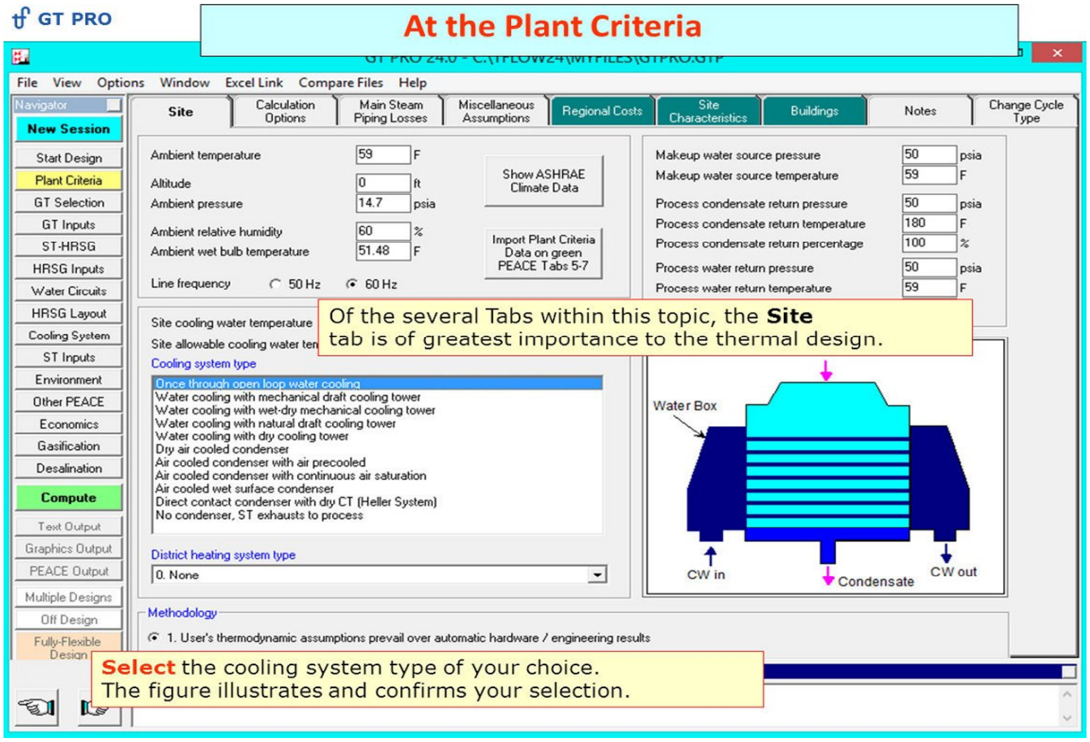


Fig. 9 – Selection of Condenser for CCPP model

4. Then select the number of Gas Turbine, Power output range of Gas Turbine, Model of Gas Turbine in the available list of GT.

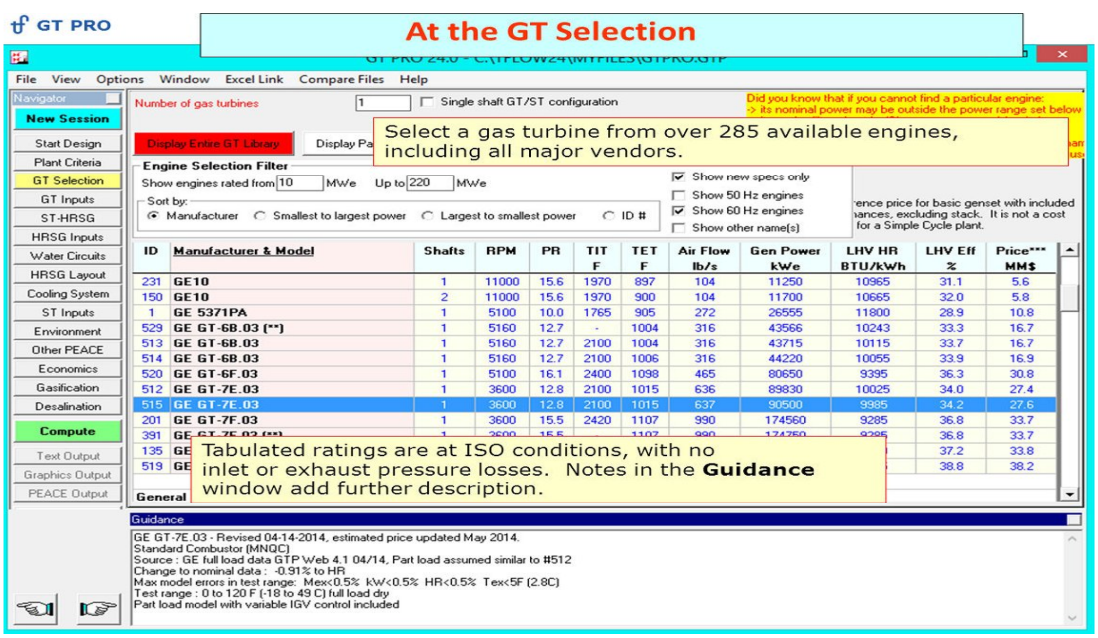


Fig. 10 – Selection of Gas Turbine for CCPP model

5. Input of Combustion chamber put the type of fuel needed such as Natural Gas, Fuel line pressure, assumed pressure drop in air filter, duct & draft pressure drop in H2O.

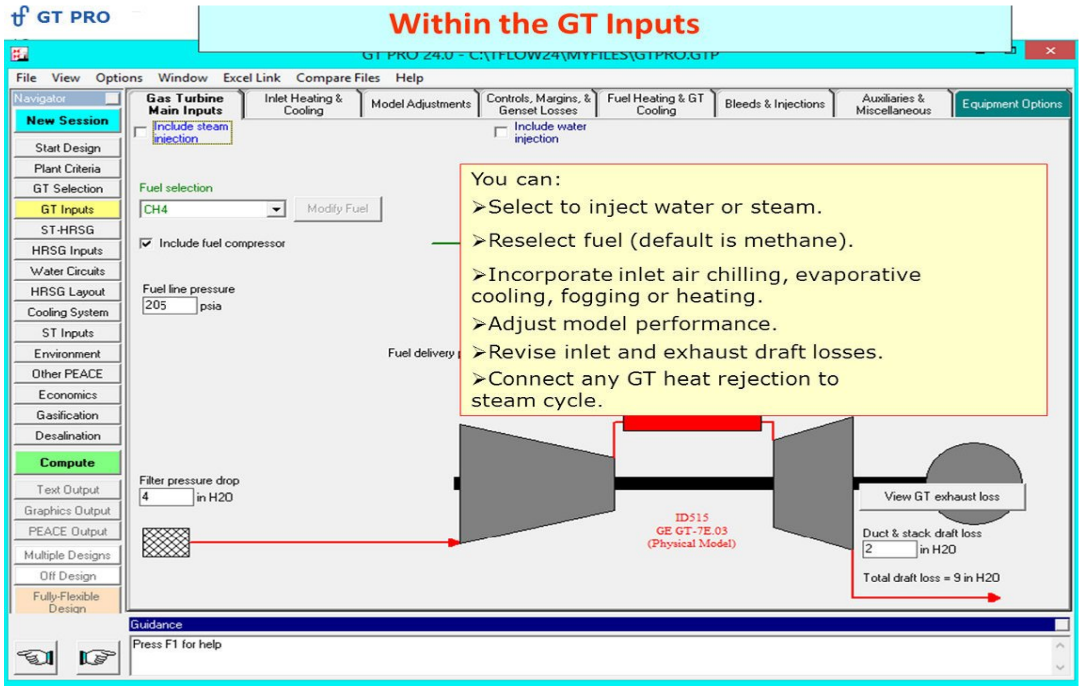


Fig. 11 – Selection of Gas Turbine input for CCPP model

6. Then, main steam parameters at different section of the Steam Turbine need to be defined. These are to be evaluated in terms of internal efficiency of Steam Turbine.

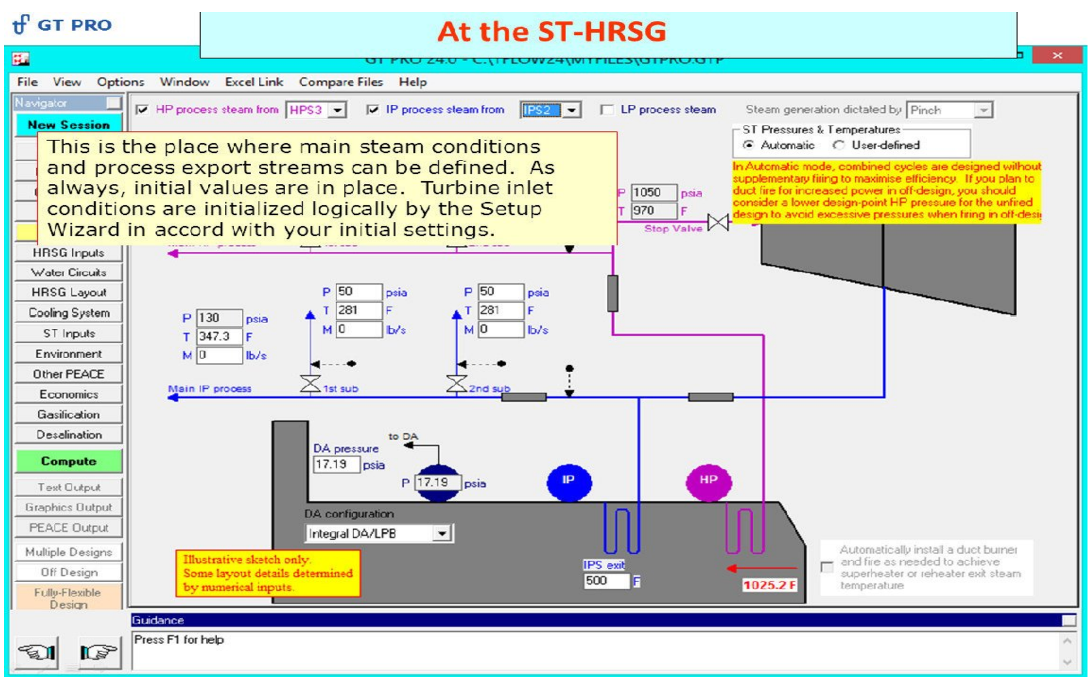


Fig. 12 – Selection of Steam parameters for Steam turbine

7. Then, input parameters of the HRSG to be defined such as type of circulation, HRSG draft loss, circulation loss, direction of flow, pinch point, approach sub-cooling, blow down percentage, evaporator exit temperature specification etc.

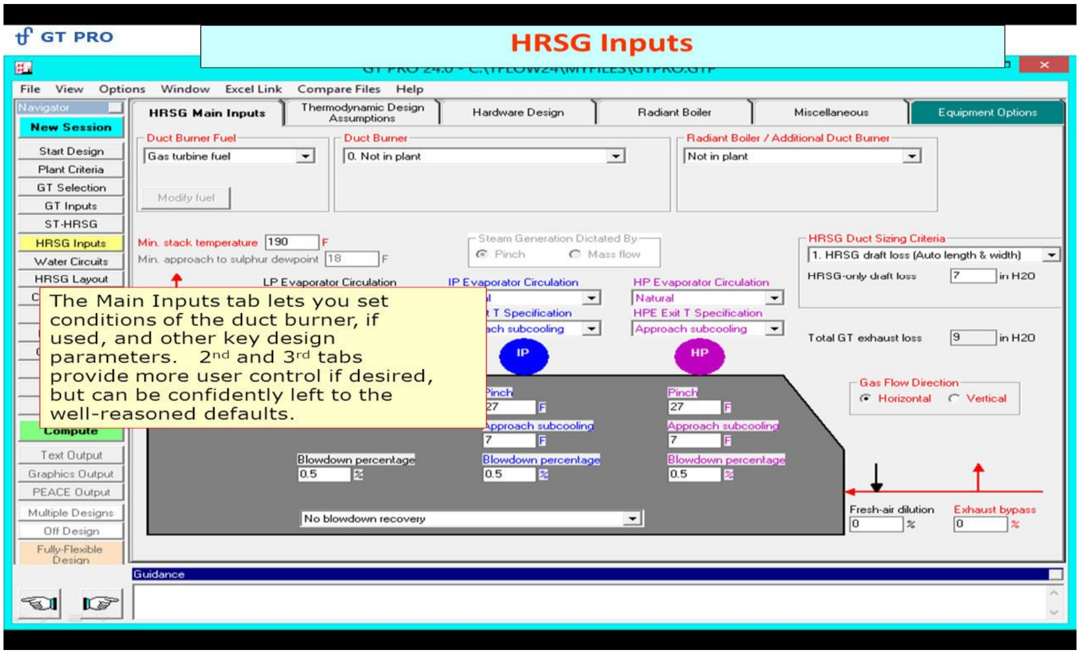


Fig. 13 – Selection of design parameters of HRSG model

8. Then, input parameters of CW circulation such as temp of cooling water, water temp rise in condenser, return water approach to wet bulb temp, condenser pressure etc.

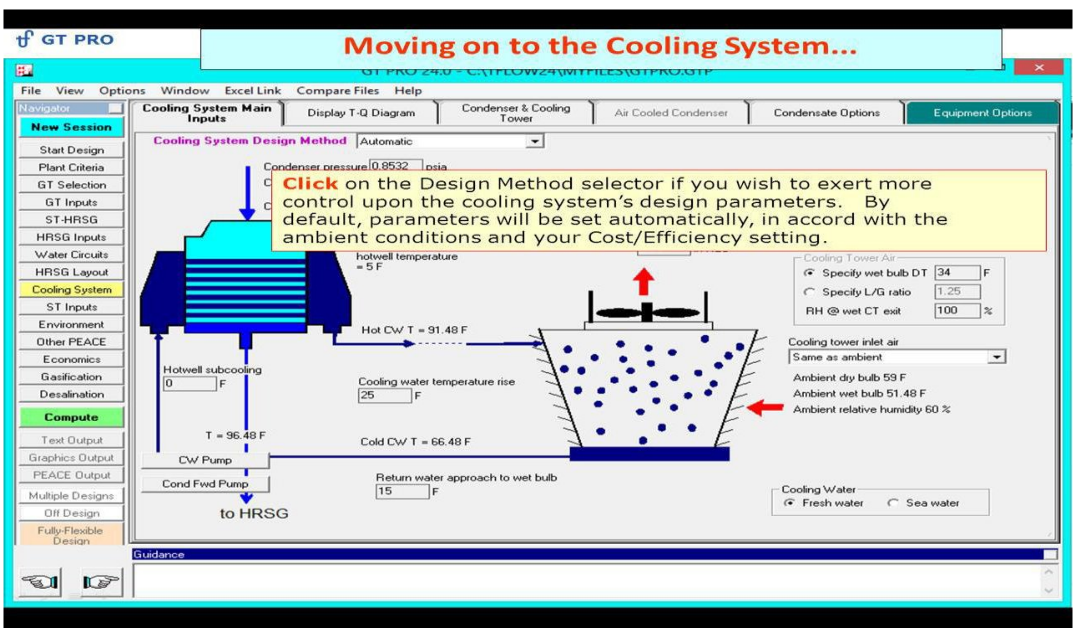


Fig. 14 – Selection of cooling water re-circulation system of CCPP model

9. In this step the input parameters of the Steam Turbine, pressure drop in piping of the steam turbine, requirement of bleed if any.

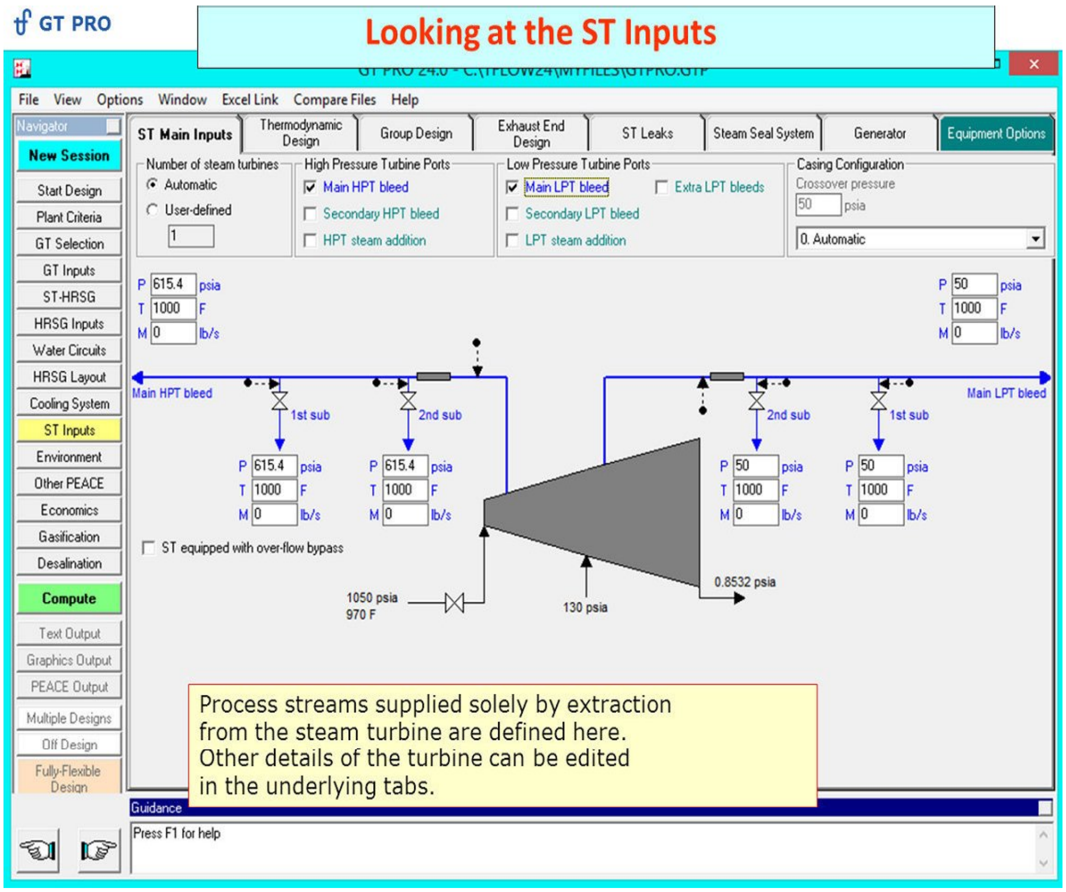


Fig. 15 – Selection of Steam process parameters of CCPP model

5.3 Procedure for GT Master software

GT MASTER Simulates expected performance of a given plant at different operating conditions, such as different ambient and loads. A multiple of varieties of inputs which define plant hardware and main control set-points can all be initialized for a plant. This makes it easy to generate correction curves from the GT MASTER model to help to calculate fuel consumption and electricity production, Heat Rate, Steam rate etc. The extensive engineering facilitates optimization of design details, such as pipe sizing or pump selection.

The steps for part load / off-design are as follows:

1. Initial for an example of impact of ambient temperature on the performance of CCGT model is shown below. The actual ambient conditions (temperature) are selected.

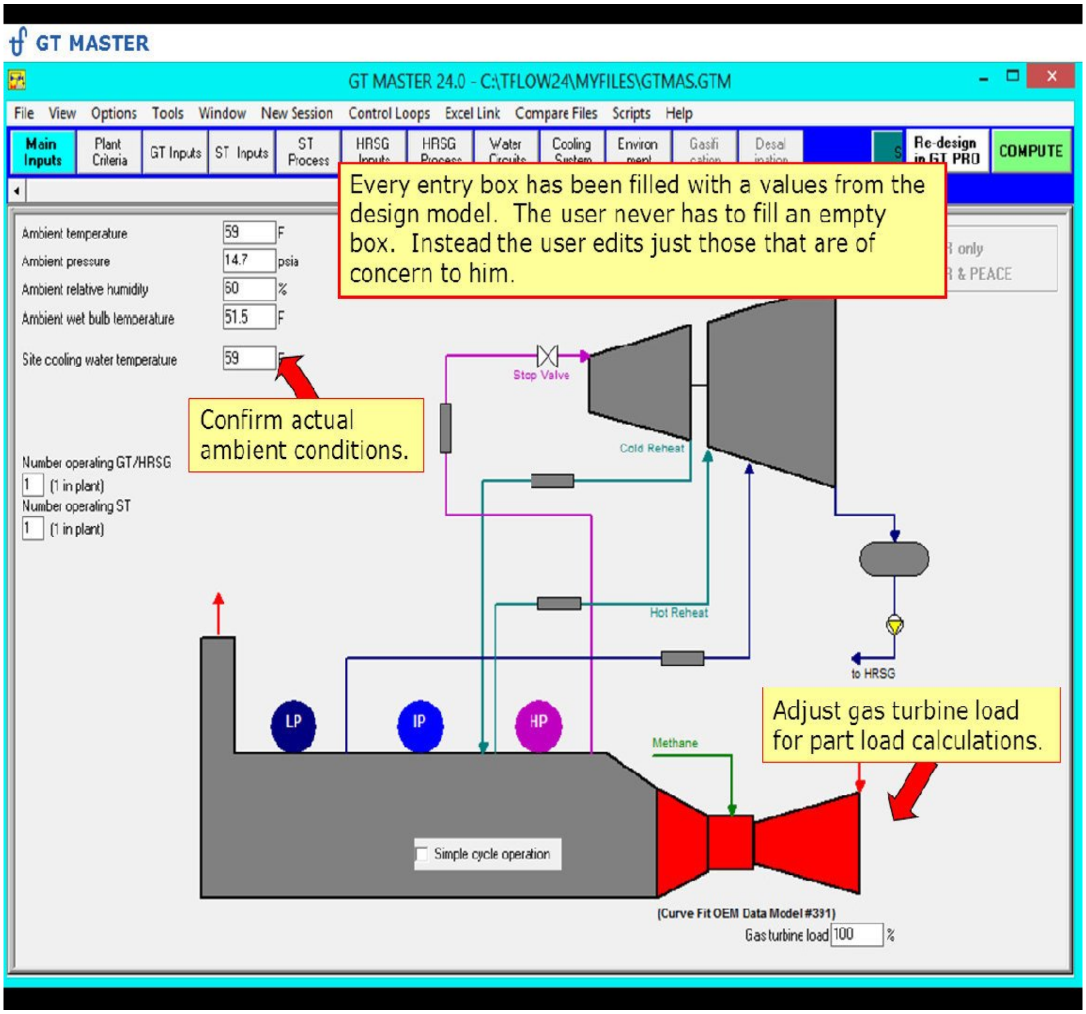


Fig. 16 – Off-design input for ambient temperature in GT Master

2. As we know, GT MASTER enables for evaluate how those designs perform at off-design conditions. GT MASTER computes expected plant performance when adjust site ambient conditions, equipment load setpoints, equipment control logic, hardware definitions, degradation levels, fuel characteristics, etc. It will quickly map out expected plant performance envelope under the ambient, dispatch, and operating conditions the plant will run in.

After development of Model the CCGT of 220 MW Triple pressure, reheat type with supplementary fired combined cycle power plant will be off-design with different set point conditions such as effect of ambient temperature, compression ratio, turbine inlet temperature on Overall efficiency, Power output, Heat rate and steam rate.

If there is off-design due to variation of main steam parameters (such as steam pressure & temperature), then steam parameters as shown below will be selected.

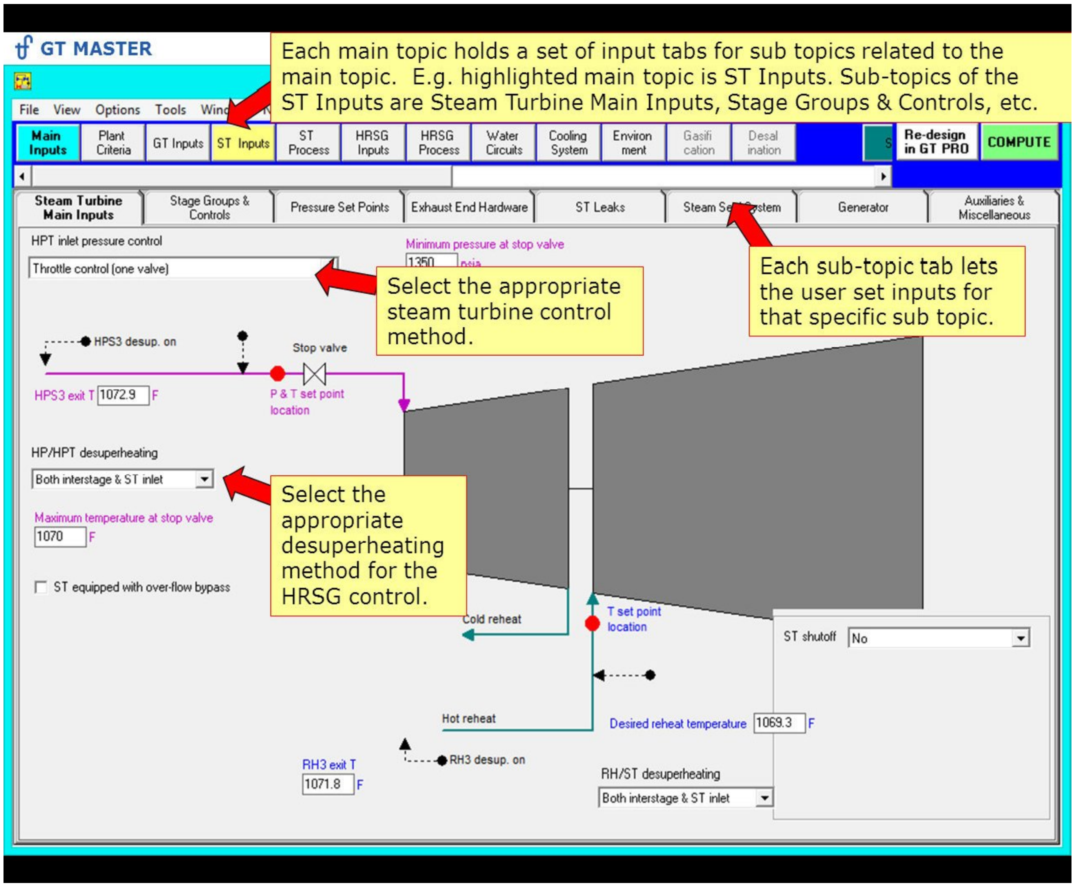


Fig. 17 – Off-design input for Steam parameters in GT Master

3. Off-design of the Model due to variation in pinch point of the HRSG the following parameters as shown will be selected.

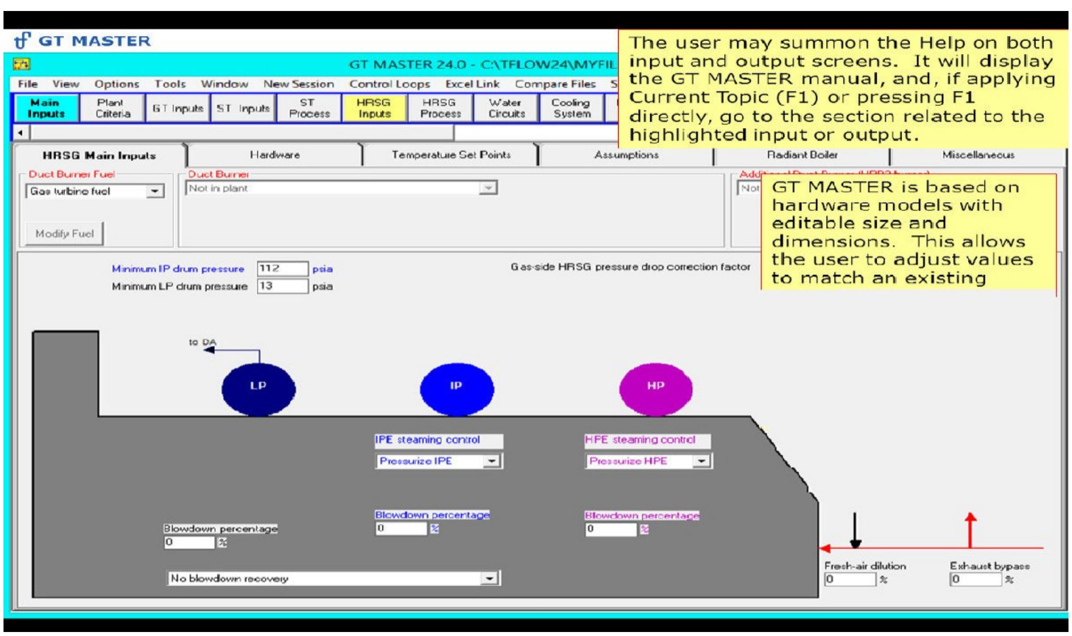


Fig. 18 – Off-design input for design parameters of HRSG in GT Master

4. The impact of change in heat transfer area of the pressure parts of the SG, the following parameters can be played to measure the overall efficiency of the CCGT.

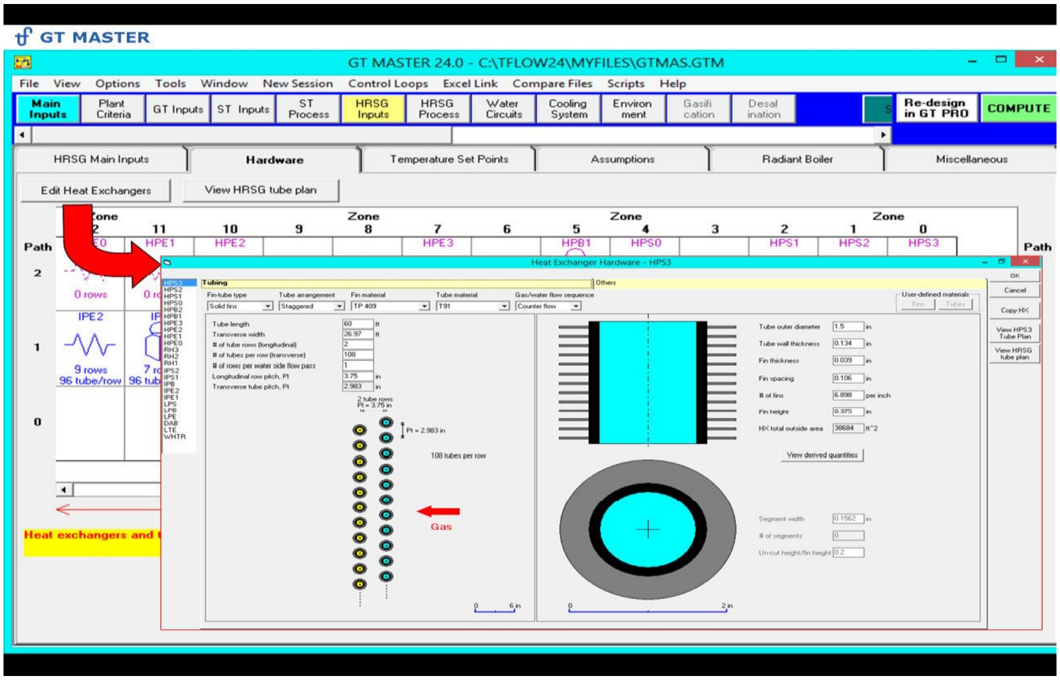


Fig. 19 – Off-design input for heat transfer area in GT Master

5. The variation of cooling cycle parameter such as cooling water temperature, difference in cooling water flow & its impact on steam cycle will be reflected as below.

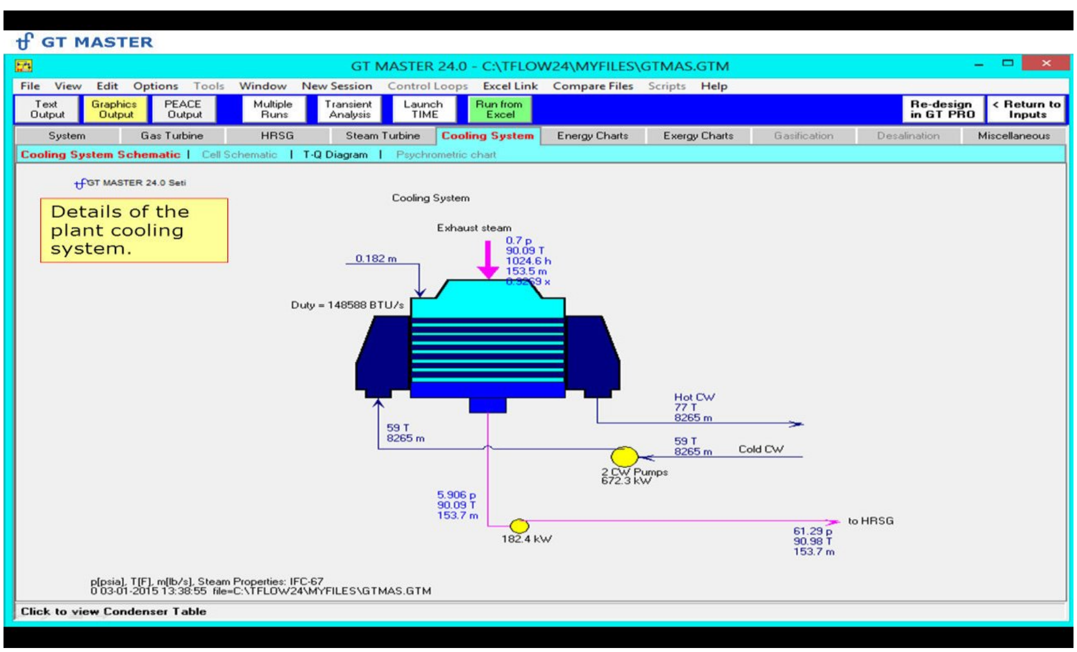


Fig. 20 – Off-design input for cooling water system in GT Master

5.4 Uncertainty Analysis

During the modelling of CCPP & it’s off design characteristic show a kind of linear correlation of impact of input parameters on the performance of Plant cycle. These results may not be linearly dependent because after a certain range of variation of critical input parameters, there is abrupt change in the characteristic of the performance parameters. However to measure the uncertainty in the result of the cycle model, the numerical correction has also been adopted.

CHAPTER – 6

RESULT & DISCUSSION

6. RESULT & DISCUSSION

6.1 Thermodynamic model of Brayton Cycle & Rankine Cycle

S.No.	Descriptions	Output Parameters
1	Nos. of Gas Turbine	2 (GE 6111FA)
2	Nos. of Steam Turbine & Condenser	1
3	HP Steam Pressure	110 bar
4	HP Steam Temperature	565° C
5	HP Steam Flow	48.94 kg/s
6	Reheat Pressure	27.62 bar
7	Reheat Temperature	366° C
8	Reheat Steam Flow	48.25 kg/s
9	IP Steam Pressure	24.1 bar
10	IP Steam Temperature	565° C
11	IP Steam Flow	55.39 kg/s
12	LP Steam Pressure	1.055 bar
13	LP Steam Temperature	292° C
14	LP Steam Flow	8.411 kg/s
15	Atmospheric pressure & Temp	1.01 bar & 25°C
16	Relative humidity	75%
17	Type of HRSG	Horizontal with Supplementary fired
18	Nos. of HRSG	1
19	Nos. of Evaporator drum	3
20	Type of Fuel Fired	Natural gas
21	Natural Gas LHV input	413009 kWth
22	HRSG feed water temperature	46°C
23	Deaerator temperature	104°C
24	Condenser (Vacuum pressure)	0.104 bar
25	Condenser flow	64.46 kg/s
26	Condensate temperature	46.6°C

Table 3 : Brayton Cycle & Rankine Cycle Output

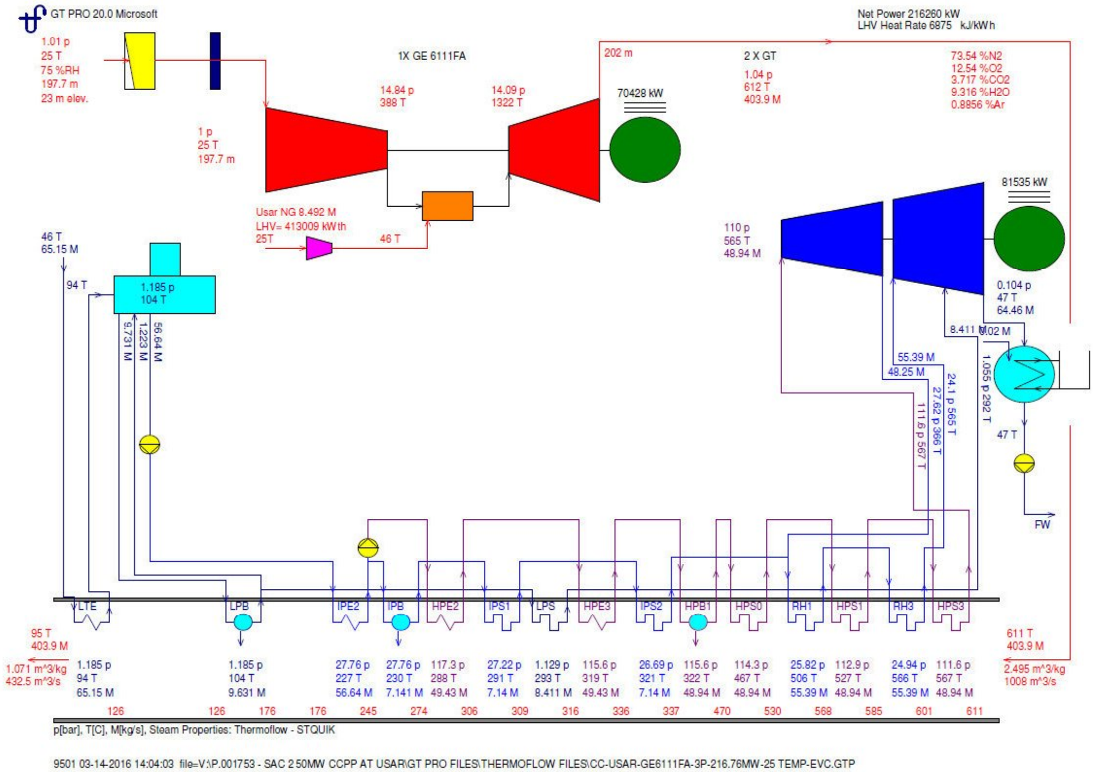


Fig. 21 – Thermodynamic model of Brayton Cycle of CCPP

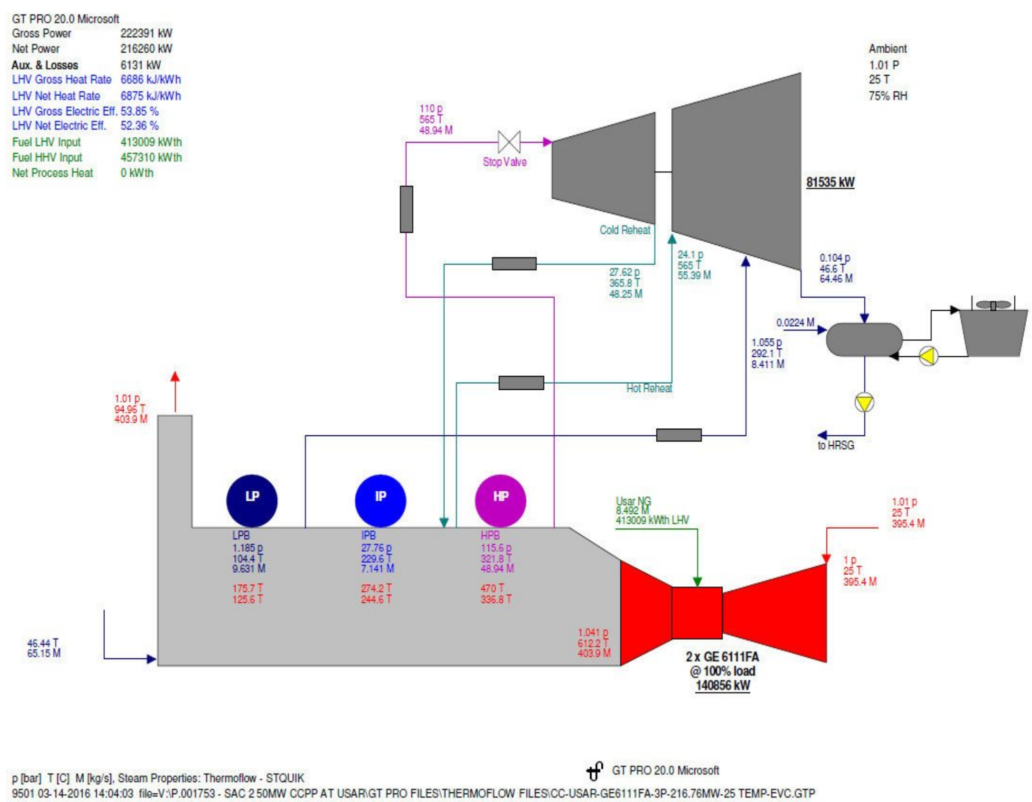


Fig. 22 – Thermodynamic model of Rankine Cycle

6.2 Result

In this section of the dissertation, the performance result of 220 MW triple pressure-reheat type with supplementary fired combined cycle power plant has been tabulated.

S.No.	Descriptions	Output Parameters
1	Total Power Output from Gas Turbine	**70428 kW from each (Total 140856 kW**
2	Total Power Output from Steam Turbine	**81535 kW**
3	Gross Power Output	**222391 kW**
4	Net Power Output	**216260 kW**
5	Aux. & Losses	**6131 kW**
6	LHV Net Heat Rate	**6875 kJ/kWh**
7	LHV Net Electrical Efficiency	**52.36%**
8	Fuel Quantity	**2.359 TPH**
9	Total compressed air required	**395.4 kg/s**
10	Flue gas exit quantity	**403.9 kg/s**
11	Flue gas exit temperature to atmosphere	**95° C**

Table 4 : Thermodynamic Model Output

6.3 Graph

This characteristic curve of 220 MW CCPP is being plotted with various design input parameters. The impact of Compressor ratio, Turbine inlet temperature, ambient temperature of Brayton cycle, Steam pressure & temperature of Rankine cycles, Reheat pressure ratio, deaerator temperature ratio, Pinch point on the performance of model in term of Power output, overall efficiency of the cycle, Heat Rate & Steam rate of the Rankine cycle is carried out.

These characteristic curves will explain the impact of key parameters on the cycle performance. Also during part load condition how the result can be estimated. The plot & its explanation are mentioned in the following section.

6.3.1 Effect of Ambient Temperature on Combined Cycle Power Plant Performance

The change of ambient temperature effect the both Steam Turbine & Gas Turbine performance which will ultimately effect performance of 220 MW CCPP. During variation of temperature of ambient, the design point of the cycle will change and it will impact the process of heat transfer to the Heat Recover Steam Generator. At the end, it also cause change in mass flow rate and exhaust temperature of gas.

If the air temperature decreases, from its design point 298°K cause an increase higher air density due to which the compressor work will reduce and this will increase power output of GT. Also due to decrease in the cooling water temperature of cooling water circuit which will create positive impact on working on water cooled condenser & thus Power output of steam cycle will increase.

As shown in figure no. 23 when ambient temperature decrease in 298°K to 288°K, the power output will increase from 220 MW to 225 MW. It is mainly due to reduction in the auxiliary power consumption of CCPP (compressor work).

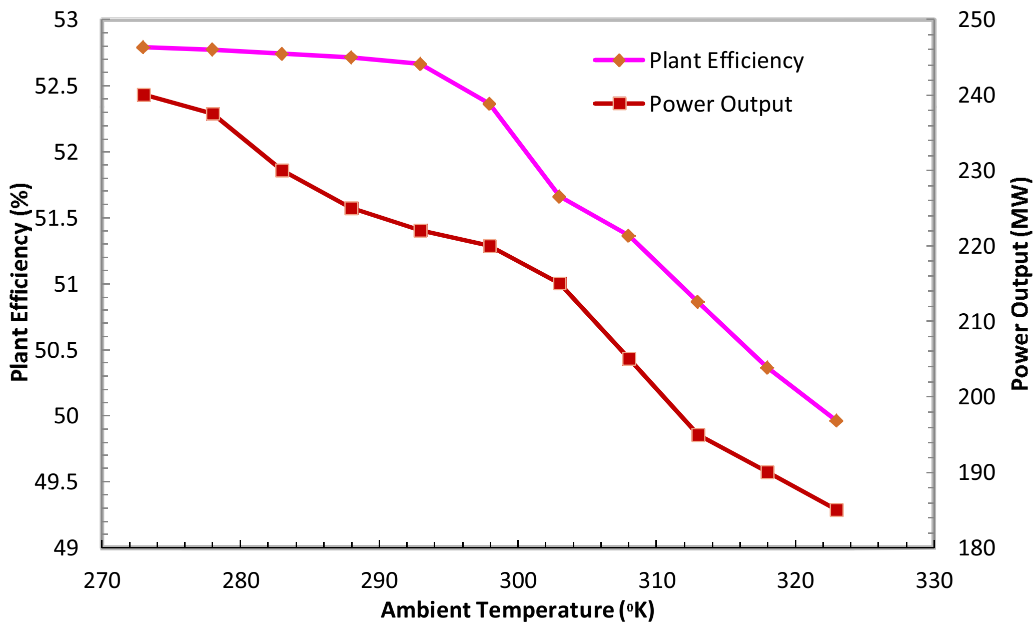


Fig. 23- Effect of ambient on Power output & overall efficiency of 220 MW CCPP

Above figure no. 23 explain the impact of ambient temperature on the Plant efficiency of the combined cycle power plant. The efficiency of CCPP will increase with decrease in ambient temperature. It is due to decrease of compressor work of plant efficiency of GT. The plant efficiency will also improve due to decrease in the exhaust gases losses. It is noticed that when ambient temperature will decrease from 298°K to 288°K, the plant efficiency will improve from 52.36% to 52.71%.

Also, when temperature of ambient will change, from 298°K to 288°K the heat rate (kJ/kWh) of combined cycle will improve from 6875 kJ/kWh to 6805 kJ/kWh. As explained above due to decrease in air density the compressor work will reduce, so for the same heat input to cycle when power output will increase, thus the Heat rate will get improved.

And, as shown in below figure 24 when ambient temperature will change from 298°K to 288°K, the steam rate will also reduce from 63.8 kg/s to 63.35 kg/s. As we know due to increase in overall efficiency of CCPP, when the same amount of steam generations is happening in HRSG & feeding to steam turbine, the steam rate will decrease.

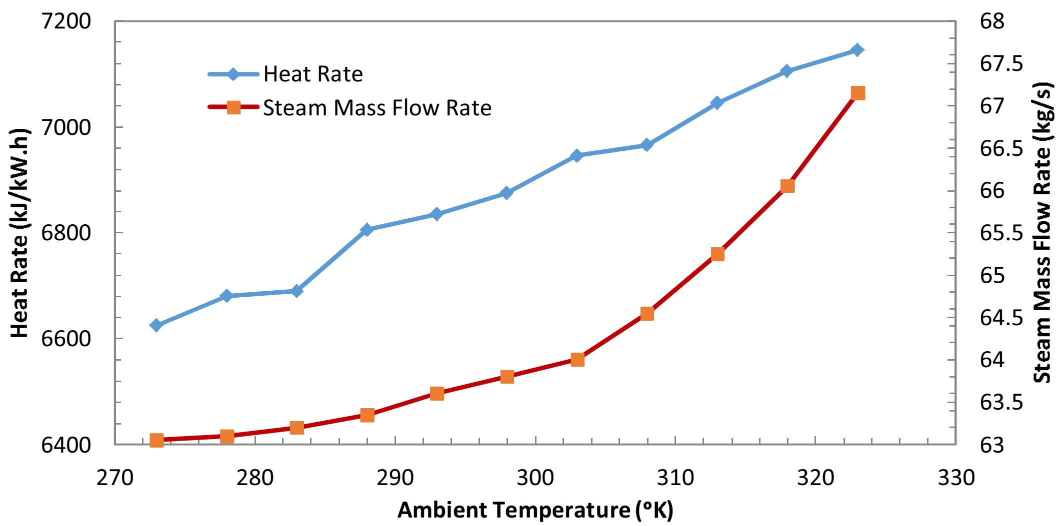


Fig. 24 - Effect of ambient on heat rate & steam rate of 220 MW CCPP

6.3.2 Effect of Compression Ratio on Combined Cycle Power Plant Performance

The impact of compression ratio effect the both Steam Turbine & Gas Turbine performance which will ultimately effect performance of 220 MW CCPP. During variation of compression ratio, the design point of the cycle will change due to off-design which will impact design power required by compressor needed during cycle operation. It is been noticed that cycle is modelled for compression ratio of 14.7 for 220 MW capacity. Now it is observed from compression ratio vary from 14.7 upto 21.1 the plant efficiency of the cycle will improve. However after CR 21.1, the efficiency will start reducing.

Also, seen that when compression ratio will increased, the power output of the cycle will reduced from design condition of CR 14.7 of 220 MW CCPP.

It is been due to when compression ratio will increase, the power consumption by compressor will increase, thus as compressor is coupled with Gas turbine, so the power output produced from Gas turbine will take by compressor for its work done. This will cause reduction in the Gross power output.

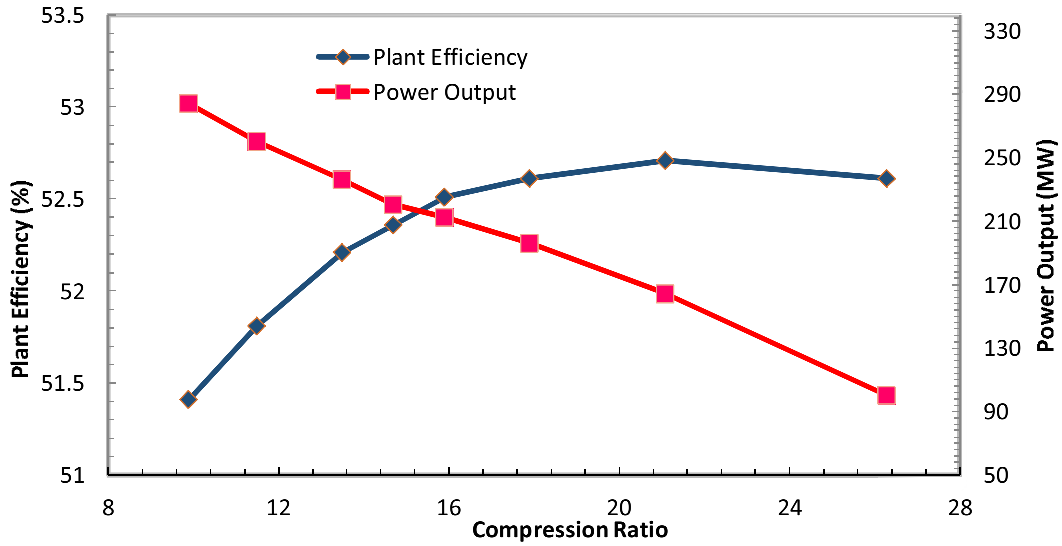


Fig. 25-Effect of compression ratio on Power output & overall efficiency of 220 MW CCPP

As we know, the power output of steam turbine will decrease when compression ratio will increase because increase in compression ratio cause negative impact on the steam generation in the bottoming cycle.

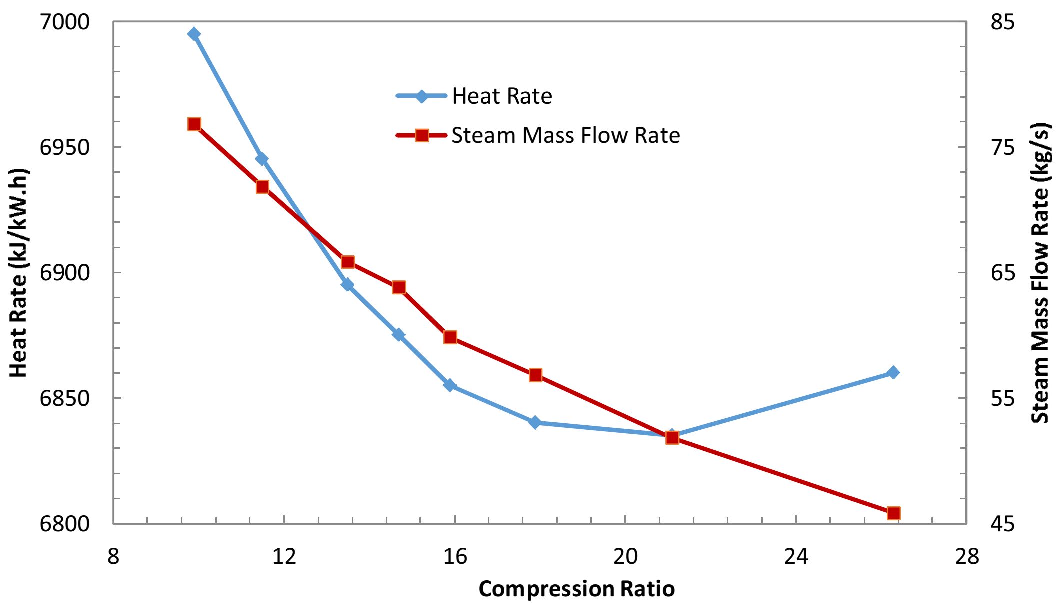


Fig. 26 - Effect of compression ratio on heat rate & steam rate of 220 MW CCPP

As shown in below curve when compression ratio will change, from 14.7 to 18 the steam flow rate will reduce from 63.8 kg/s to 56.8 kg/s. The power output of Gas turbine in terms of magnitude is greater than Steam turbine output. As seen when compression ratio will increase, the improvement in the plant efficiency the heat rate will improved from 6875 kJ/kWh to 6840 kJ/kWh but after a threshold point after which there is negative impact of rise in compressor ratio, cause plant efficiency will reduced which and hence increase in heat rate.

6.3.3 Effect of Gas Turbine Inlet Temperature on Combined Cycle Power Plant Performance

The impact of turbine inlet temperature (TIT) effect the both Steam Turbine & Gas Turbine performance which will ultimately effect performance of 220 MW CCPP. At a design point of pressure ratio 14.7 & ambient temperature of 298°K, increase in TIT cause increase in overall efficiency & Power output of ST & GT cycle.

Due to increase in temperature, the heat (enthalpy) available at the gas turbine inlet will increase, thus power output of the GT will increase with same work done by compressor unit. But raise in temperature, the heat available at the heat transfer area of HRSG will increase cause increase in enthalpy at the HP turbine inlet and work output will increase. It is shown in figure 27 below raise in design temperature of 1322°C to 1522°C cause in increase in power output from 220 MW to 297 MW and plant efficiency will increase from 52.36% to 55.76%.

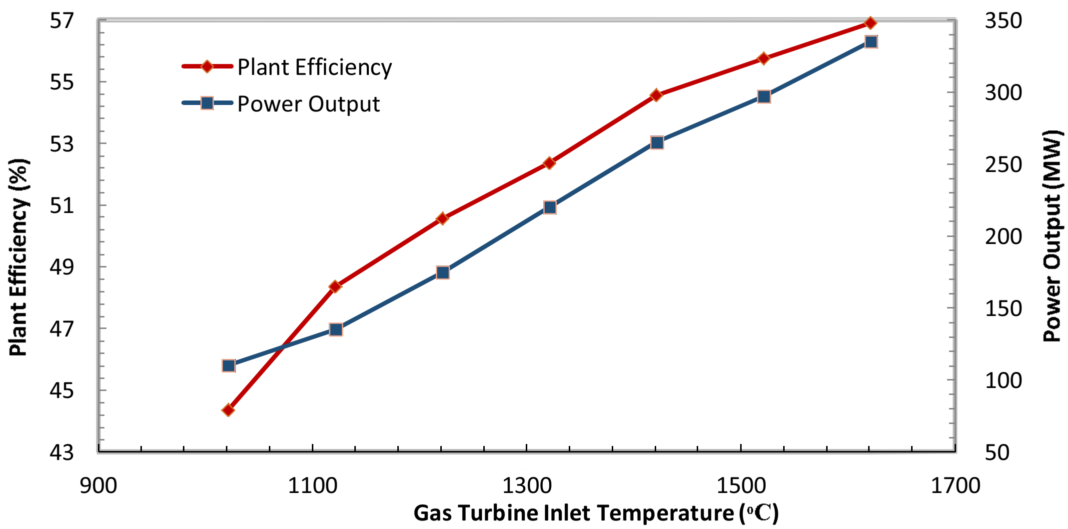


Fig. 27 - Effect of Gas Turbine IT on Power output & overall efficiency of 220 MW CCPP

As we know, due to increase in TIT the power output of steam turbine will increase even compression ratio will increase because increase. As we know due to more heat energy available at HRSG for the steam generation in the bottoming cycle will reduce for ST unit power. As shown in below curve when gas turbine inlet temperature will change from 1322°C to 1522°C, the steam flow rate will reduce from 63.8 kg/s to 44.3 kg/s. Due to raise in Gas turbine inlet temperature, the heat rate of the plant will also increase from 6875 kJ/kWh to 7455 kJ/kWh.

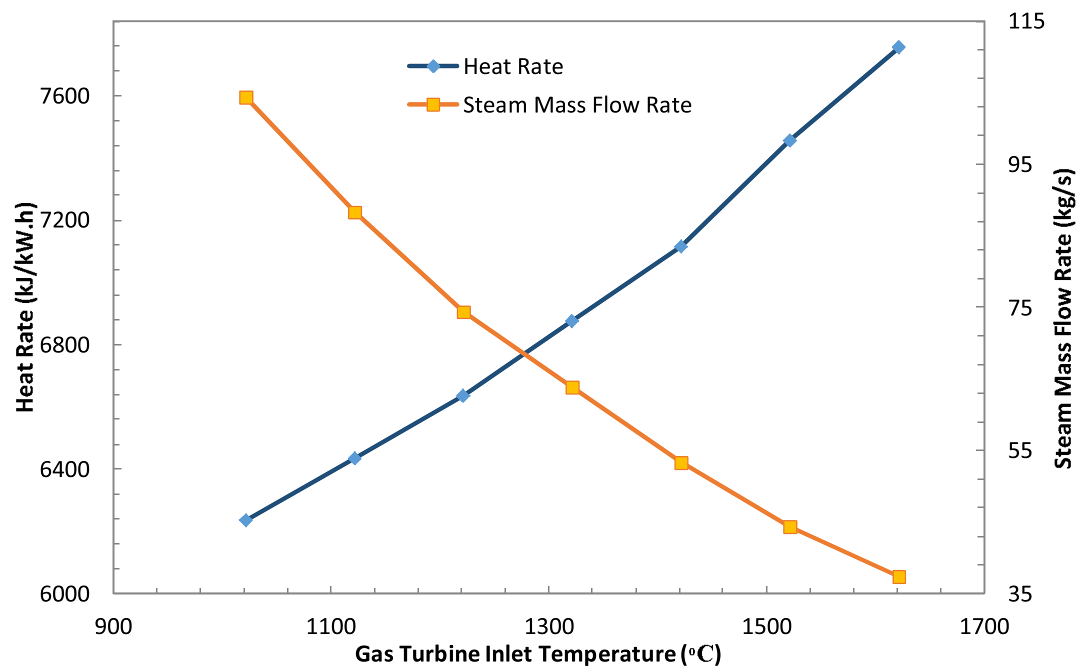


Fig. 28 - Effect of Gas Turbine Inlet Temp on heat rate & steam rate of 220 MW CCPP

6.3.4 Effect of Steam Inlet Pressure of Rankine Cycle with Different Steam Temperature on Combined Cycle Power Plant

As we know that with increase in pressure at corresponding temperature cause raise in heating value (enthalpy) of the steam, hence the net power output of the cycle will increase, because heat input is directly proportional the power output. Therefore the performance of the Rankine cycle will improve.

A plot has been shown below for raise in power output with increase in pressure with corresponding temperature.

As it is shown below, cycle is design for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. The pressure ratio 14.7 & Turbine inlet temperature 1322°C will remain fixed in the cycle. Three cases of Steam temperature at inlet of HP turbine has been considered of 500°C, 650°C and 800°C.It is been depicted that with increase in pressure from 20 bar to 230 bar & with steam turbine cycle from 500 to 800°C the specific power output will continuously increase from 902 kJ/kg to 977 kJ/kg .

As discussed belowe, with the increase in pressure with corresponding increase Power output of the cycle will increase. Thus when the specific power output of the cycle increases will leads to improvement in the Plant efficiency of cycle.

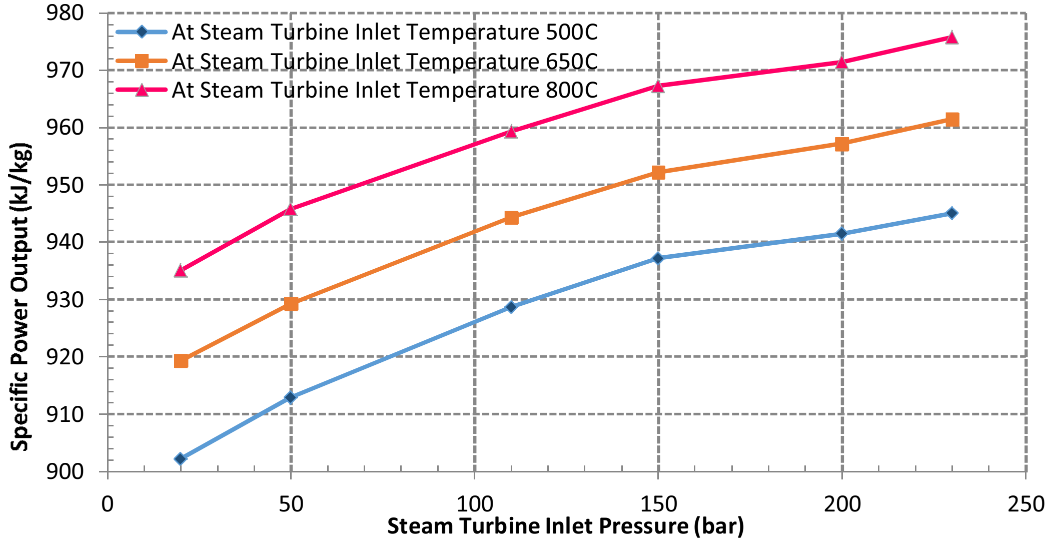


Fig. 29 - Effect of HP Steam turbine pressure on Specific Power output of 220 MW Combined Cycle Power Plant at Compression ratio 14.7 & Gas Turbine Inlet temperature 1322°C

As it is shown below, cycle is design for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. It is been depicted that with increase in pressure from 20 bar to 230 bar & with steam turbine cycle temperature range from 500 to 800°C the plant efficiency will continuously increase from 49.5% to 54.6%.

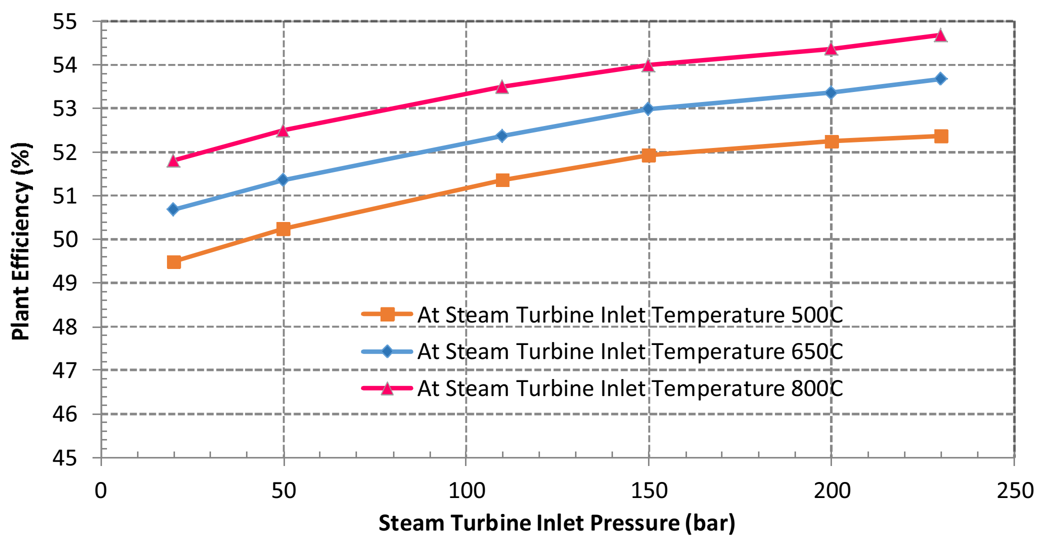


Fig. 30 - Effect of HP Steam turbine pressure on Plant efficiency of 220 MW Combined Cycle Power Plant at Compression ratio 14.7 & Gas Turbine Inlet temperature 1322°C

6.3.5 Effect of Compression Ratio with Different Gas Turbine Inlet Temperature on Combined Cycle Power Plant

As we know that selection of compression ratio for the range of power output is very optimistic. Always increase in compression ratio with set of design parameters may lead to even decrease in power output of the Cycle. As shown below for set of design condition with Inlet turbine temperature 1100°C, if further increase in compression ratio from 11.7 to 17.7 leads to decrease in specific power output from 383.3 kJ/kg to 345 kJ/kg. However on other hand, plot show with turbine inlet temperature of 1300°C & 1500°C leads to increase in work out from 675 kJ/kg to 1028 kJ/kg.

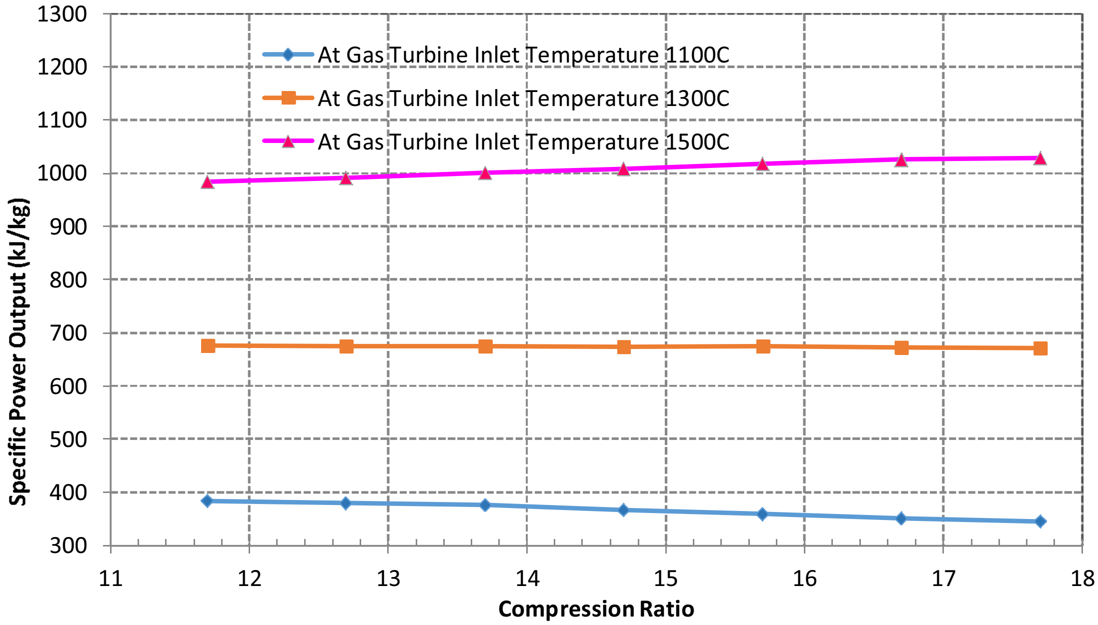


Fig. 31- Effect of Compression ratio on Specific Power output of 220 MW Combined Cycle Power Plant at Turbine Inlet pressure 110 bar & temperature 565°C

As shown below, with the same reference condition of Steam pressure 110 bar and Steam temperature 565°C, with the increase in compression ratio from 11.7 to 17.7 with corresponding turbine inlet temperature of 1100°C, 1300°C & 1500°C, the efficiency will increase from 44% to 56.5%.

The Gas turbine inlet temperature has strong influence over the Plant efficiency. As the temperature increase, the impact of compression ratio keep sudden raise in Plant efficiency.

For all the three cases, the result is as follows:

- Case 1 HP turbine condition 110 bar, 565°C, TIT 1100°C, CR 14.7 Plant Ƞ 48%
- Case 2 HP turbine condition 110 bar, 565°C, TIT 1300°C, CR 14.7 Plant Ƞ 52.36%
- Case 3 HP turbine condition 110 bar, 565°C, TIT 1500°C, CR 14.7 Plant Ƞ 53.78%

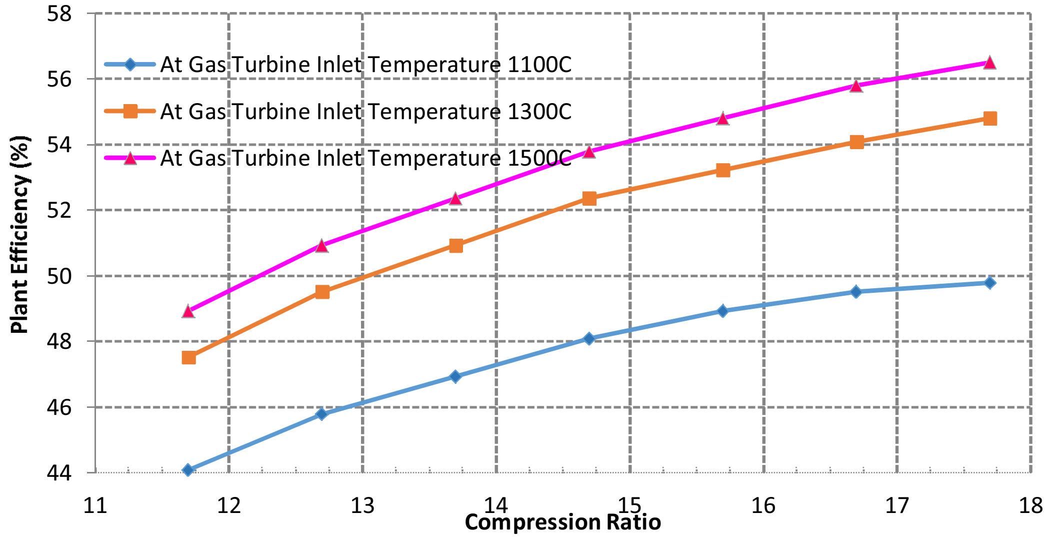


Fig. 32 - Effect of Compression ratio on Plant efficiency of 220 MW Combined Cycle Power Plant at Turbine Inlet pressure 110 bar & temperature 565°C

6.3.6 Effect of Steam Inlet Temperature of Rankine Cycle with Different HP Steam Pressure on Combined Cycle Power Plant

As we know that with increase in temperature with corresponding pressure cause raise in heating value of the steam, it is found that the net-work out of the cycle increase due to improvement in performance of Rankine cycle. A plot has been shown below for raise in specific power output with increase in temperature with corresponding pressure.

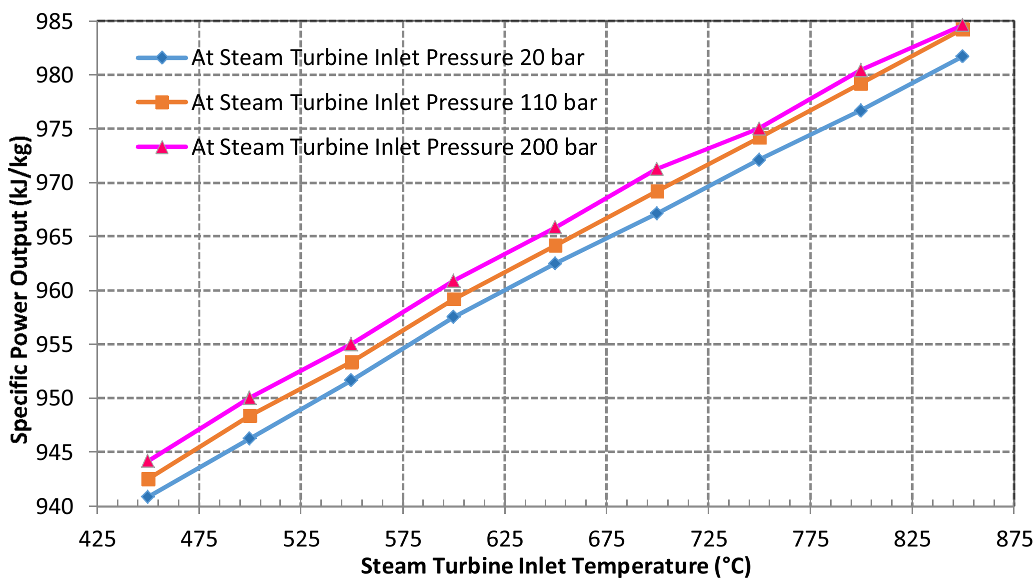


Fig. 33 - Effect of HP Steam Turbine Temperature on Specific Power output of 220 MW Combined Cycle Power Plant at Compression ratio 14.7 & Gas Turbine Inlet temp 1322°C

As it is shown above, cycle is design for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. It is been depicted that with increase in temperature from 450°C to 850°C & with steam turbine cycle pressure for 20 bar to 200 bar, specific power output will continuously increase from 941 kJ/kg to 984 kJ/kg .

As discussed above, with the increase in temperature with the corresponding pressure leads to increase in specific power output of the cycle. Thus when the specific power output of the cycle increases, due to which the Plant efficiency of cycle will increases.

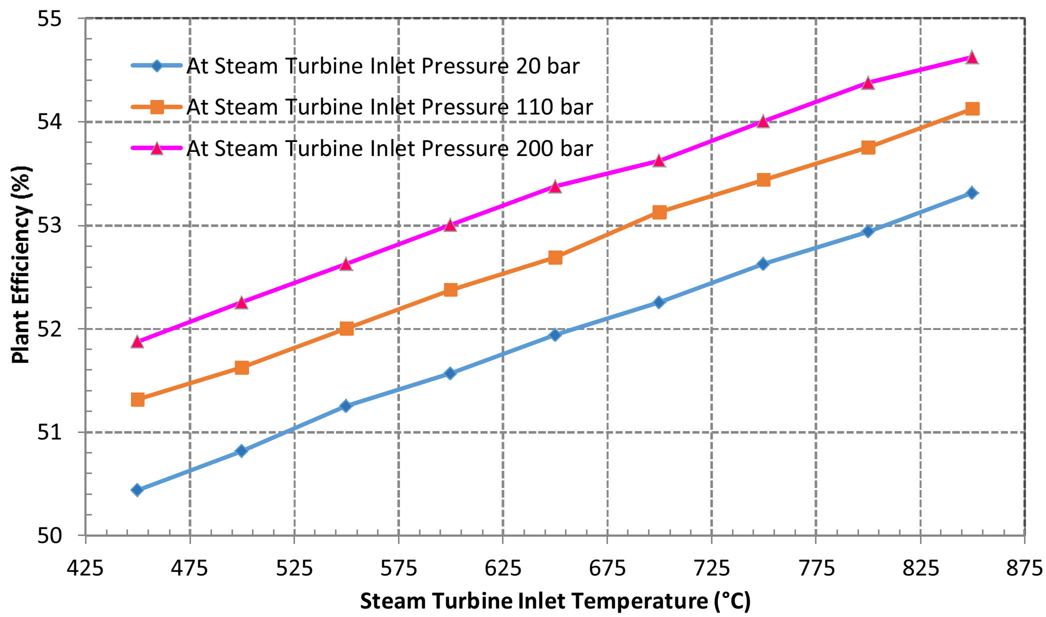


Fig. 34 - Effect of HP Steam turbine temperature on Plant efficiency of 220 MW Combined Cycle Power Plant at Compression ratio 14.7 & Gas Turbine Inlet temp 1322°C

As it is shown above, with design point for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. It is been depicted that with increase in steam temperature from 450°C to 850°C & with steam turbine cycle pressure for 20 bar to 200 bar, work out will continuously increase from 50.4% to 54.6%.

6.3.7 Effect of Gas Turbine Inlet Temperature with Different Compression Ratio on Combined Cycle Power Plant

As we know that with increase in gas temperature at the GT inlet cause increase in heating value of the Gas with corresponding compression ratio, the heat energy available at the gas turbine inlet will increase, thus power output of the Brayton cycle will increase & Net power output of the combined Cycle will increase.

As it is shown below, cycle is design for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. It is been depicted that with increase in gas turbine inlet temperature from 1100°C to 1600°C & with compression ratio range from 10 to 15, the specific power output will continuously increase from 314.2 kJ/kg to 714 kJ/kg .

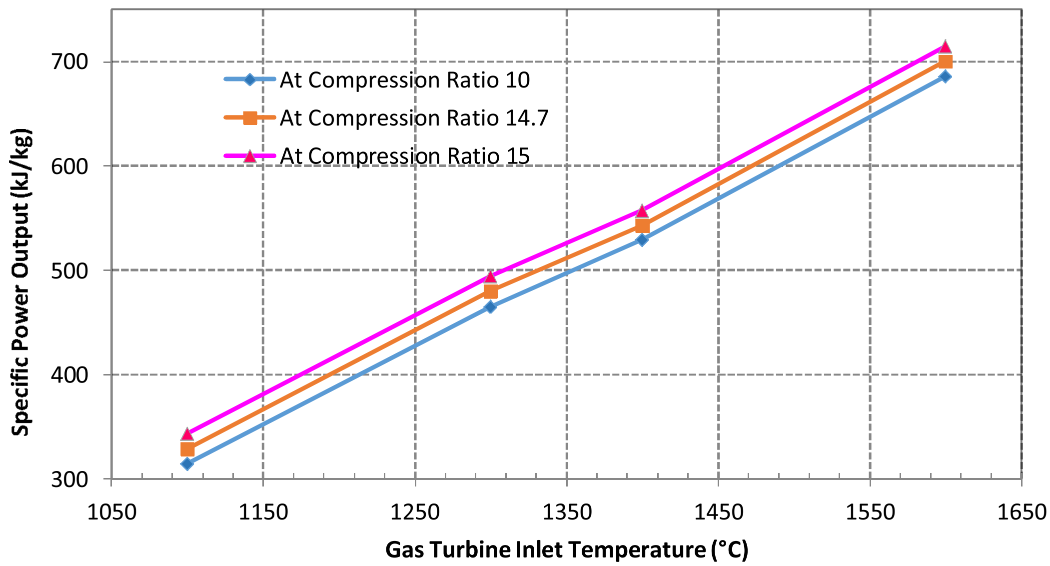


Fig. 35 - Effect of Gas Turbine Inlet Temperature on Specific Power output of 220 MW Combined Cycle Power Plant at Steam turbine inlet pressure 110 bar & temp 565°C

As we know that with increase in gas temperature at the GT inlet cause increase in heating value of the Gas with corresponding compression ratio, the heat energy available at the gas

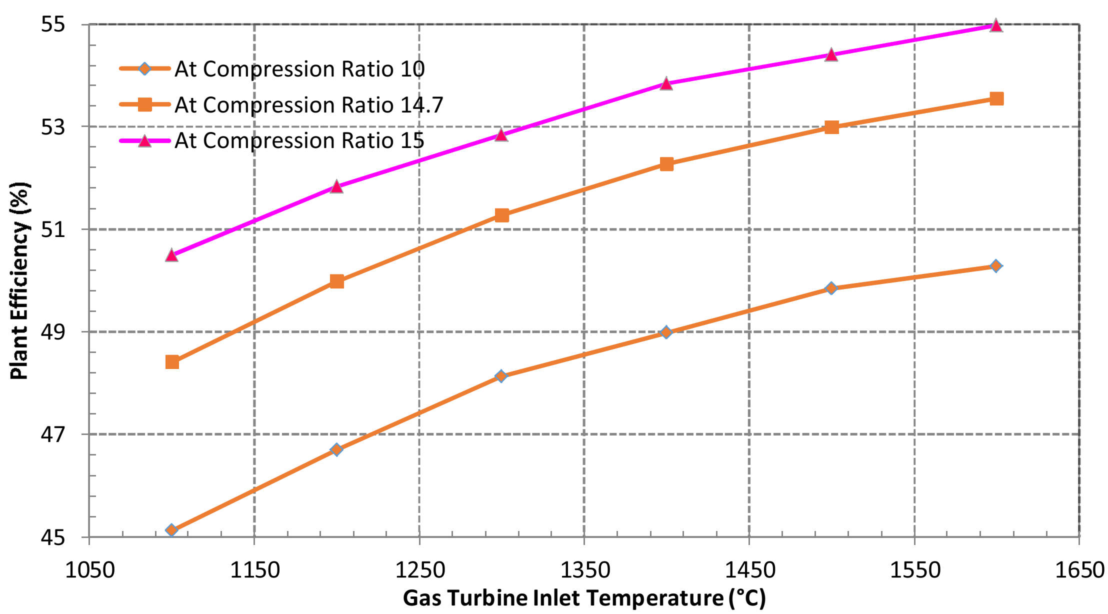


Fig. 36 - Effect of Gas Turbine Inlet Temp on Plant efficiency of 220 MW Combined Cycle Power Plant at Steam turbine inlet pressure 110 bar & temp 565°C

inlet will increase, thus with increase in power output of the Brayton cycle the overall efficiency will also increase. As it is shown below, cycle is design for Compression ratio 14.7, Gas inlet temperature 1322°C, Rankine cycle with HP steam pressure 110 bar & temperature 565°C. It is been depicted that with increase in temperature from 1100°C to 1600°C & with compression ratio range from 10 to 15, the efficiency will improve from 45.1% to 54.9%.

6.3.8 Effect of Steam Reheat Pressure Ratio on Combined Cycle Power Plant Performance

As determined from figure 37 we know that fraction of HRSG HP has been plotted as Steam reheat pressure. As it has been depicted that the Power output is increased with Steam reheat pressure ratio. But due steam reheat pressure ratio is limited as 0.5 due to limitation of dryness fraction (0.8).

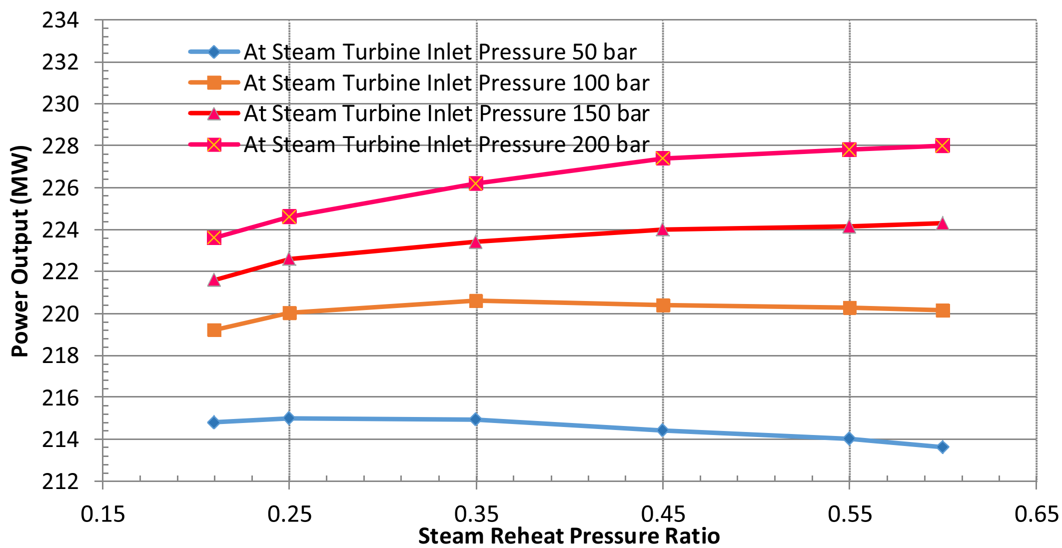


Fig. 37 - Effect of Steam Reheat Pressure Ratio on Power output of 220 MW CCPP

The impact of Steam reheat pressure ratio on the Plant efficiency of 220 MW CCPP plotted refer fig. 38. It has been depicted that plant efficiency of the CCPP will increase at higher pressure with increase in Steam Reheat pressure ratio. It has been noticed that with low HP steam pressure, the Plant efficiency will reduce with increase in Steam reheat pressure ratio

For all the three cases, the result is as follows:

- Case 1 HP turbine inlet 50 bar, reheat pressure ratio 25%, Plant Ŋ will be 51.76%.
- Case 2 HP turbine inlet 100 bar, reheat pressure ratio 25%, Plant Ŋ will be 52.36%.
- Case 3 HP turbine inlet 150 bar, reheat pressure ratio 25%, Plant Ŋ will be 52.72%.

- Case 4 HP turbine inlet 200 bar, reheat pressure ratio 25%, Plant Ƞ will be 53.01%.

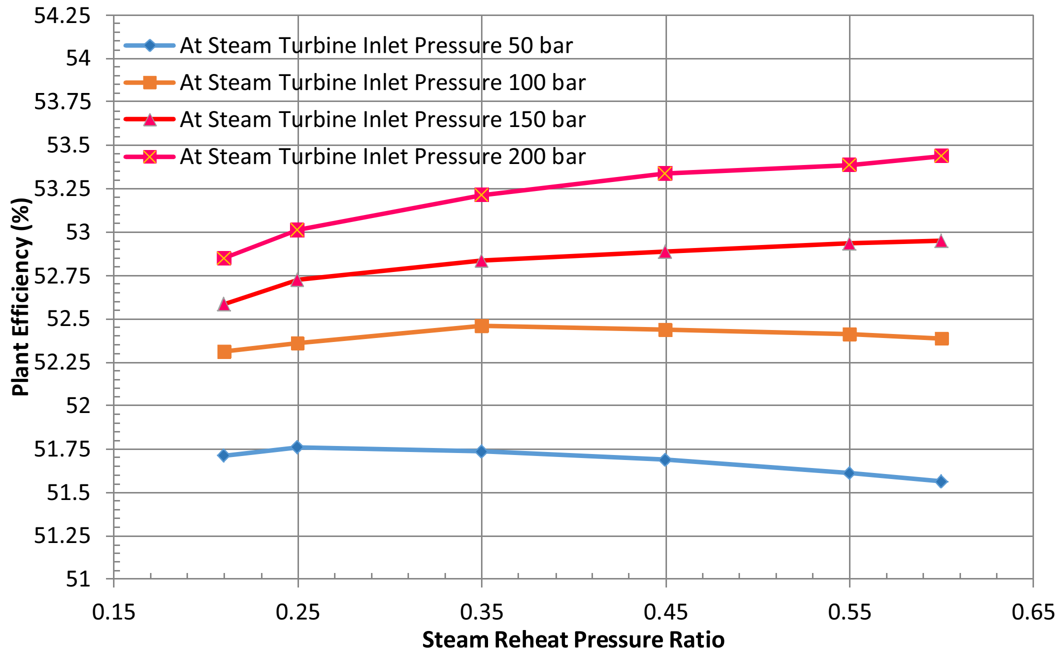


Fig. 38 - Effect of Steam Reheat Pressure Ratio on Plant efficiency of 220 MW CCPP

6.3.9 Effect of Deaerator Temperature Ratio on Combined Cycle Power Plant Performance

As determined from figure 39 we know that fraction of HRSG HP has been plotted as deaerator temperature ratio. The deaerator temperature ratio is defined as below:

$$Deaerator\ Temp\ ratio\ (T_{deae}) = \frac{T_{deae,sat} - T_{cond,sat}}{T_{HP,sat} - T_{cond,sat}}$$

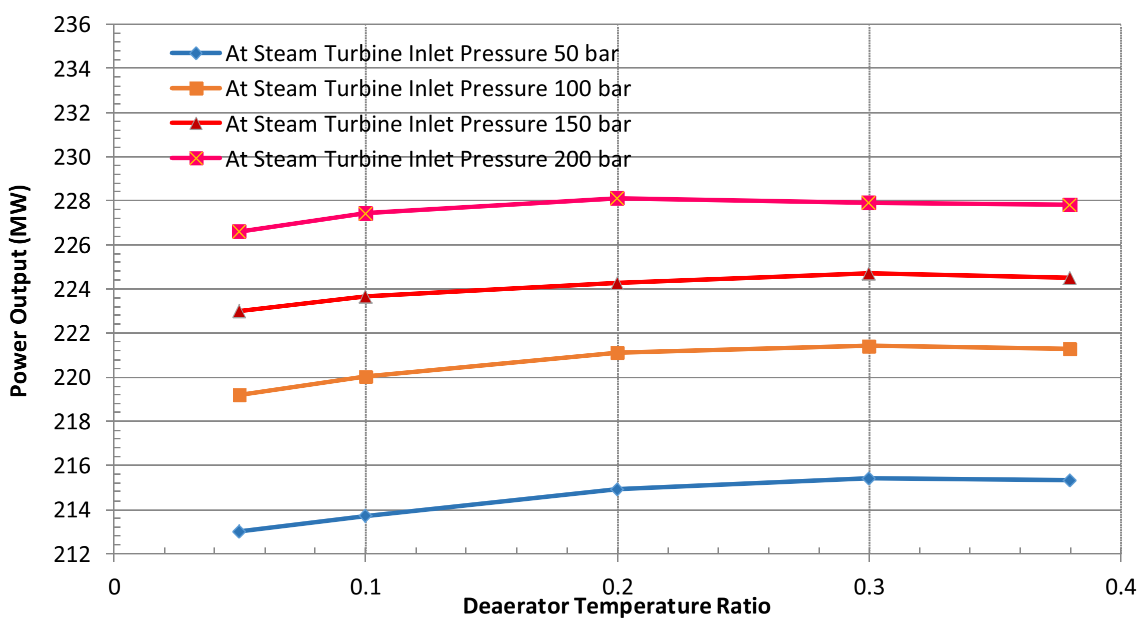


Fig. 39 - Effect of Deaerator temperature ratio on Power output of 220 MW CCPP

It has been depicted that the Power output is increased with increase in deaerator temperature ratio at different sets of HRSG High Pressure. Due to variation of HRSG HP of 50 bar to 200 bar gives best results when deaerator temperature ratio found in between 0.25 to 0.3.

Similarly, the impact of Deaerator temperature ratio on the Plant efficiency of 220 MW CCPP plotted below. It has been depicted that plant efficiency of the CCPP will increase at higher value of HRSG high pressure with increase in deaerator temperature ratio.

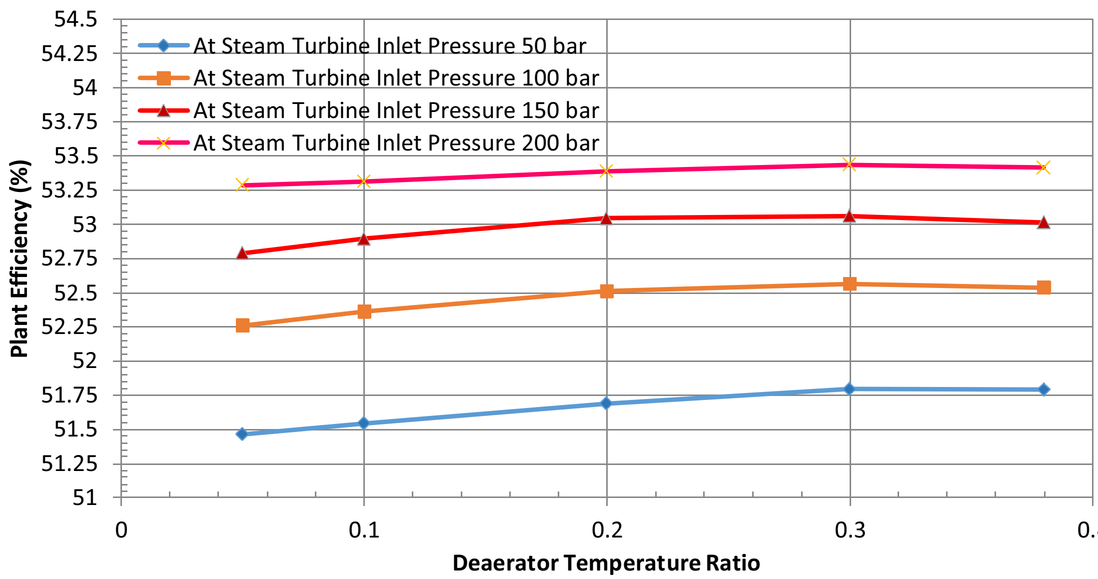


Fig. 40 - Effect of Deaerator temperature ratio on Plant efficiency of 220 MW CCPP

6.3.10 Effect of Pinch Point on Combined Cycle Power Plant Performance

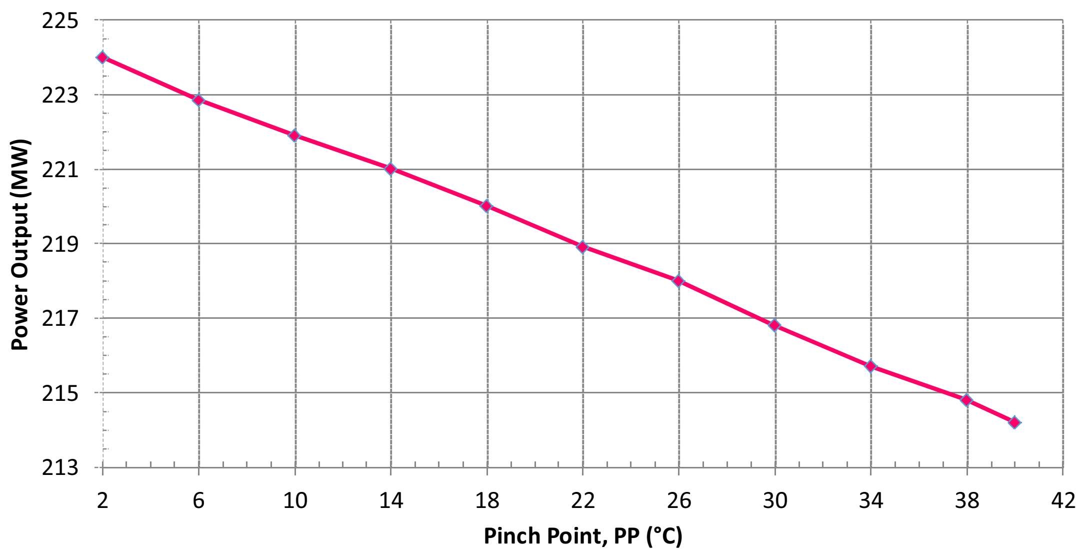


Fig. 41 - Effect of Pinch Point on Power output of 220 MW CCPP

From above Figure 41, the impact of Pinch point on the performance of CCPP can be assessed. It has been depicted that the gross power output of combined cycle will decreases with increase in pinch point.

Thus reduction in pinch point increases the more requirement of heat transfer area of HRSG. It is easier to mention that maximum steam turbine power output & efficiency will be maximum at null value of pinch point and infinite times of heat transfer area.

6.3.11 Effect of HRSG High Drum Pressure on Combined Cycle Power Plant Performance

It has been determined from below plotted figure 42 that the HRSG drum high pressure plays a major role for the gross power output of the Rankine cycle. The power output of the Rankine cycle (gross power output) will increase with increase in HRSG drum high pressure.

As we know our cycle has been design point of HP pressure is 110 bar with 220 MW, when pressure increase in HP drum upto 200 bar, the power output will increase by 224.3 MW. However it has been observed that even increase in HP drum pressure by two times, there marginal increase in gross power by 2% only.

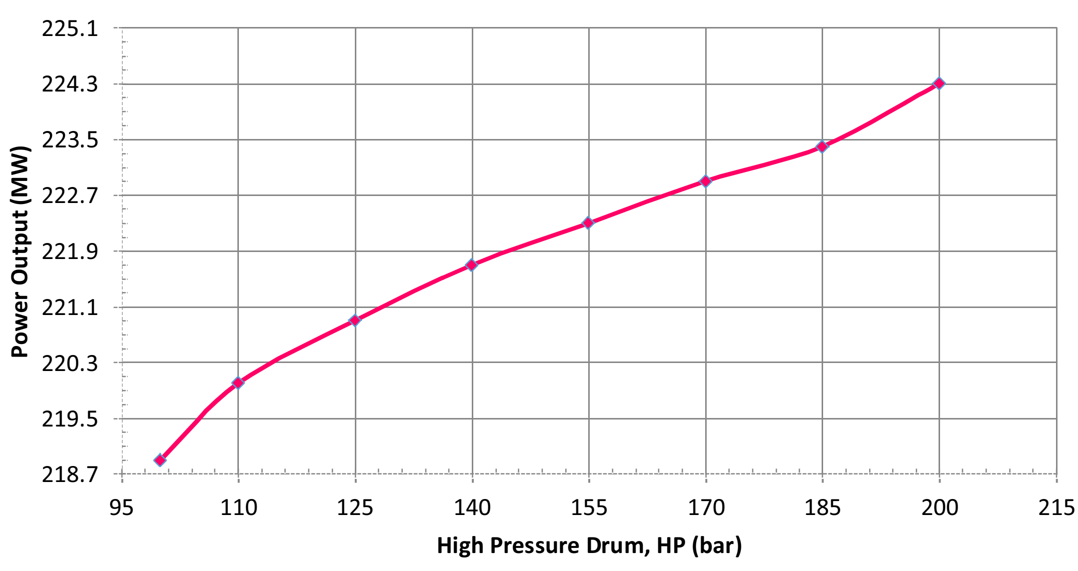


Fig. 42 - Effect of HRSG High Pressure on Power output of 220 MW CCPP

6.3.12 Effect of HRSG Intermediate Drum Pressure on Combined Cycle Power Plant Performance

It has been determined from below plotted figure 43 that the HRSG drum intermediate pressure plays a major role for the gross power output of the Rankine cycle. The power output of the Rankine cycle (gross power output) will decrease with increase in HRSG drum intermediate pressure. As we know our cycle has been design point of IP pressure is 24.1 bar with 220 MW, when pressure increase in IP pressure upto 38.3 bar, the power output will decrease by 218.18 MW.

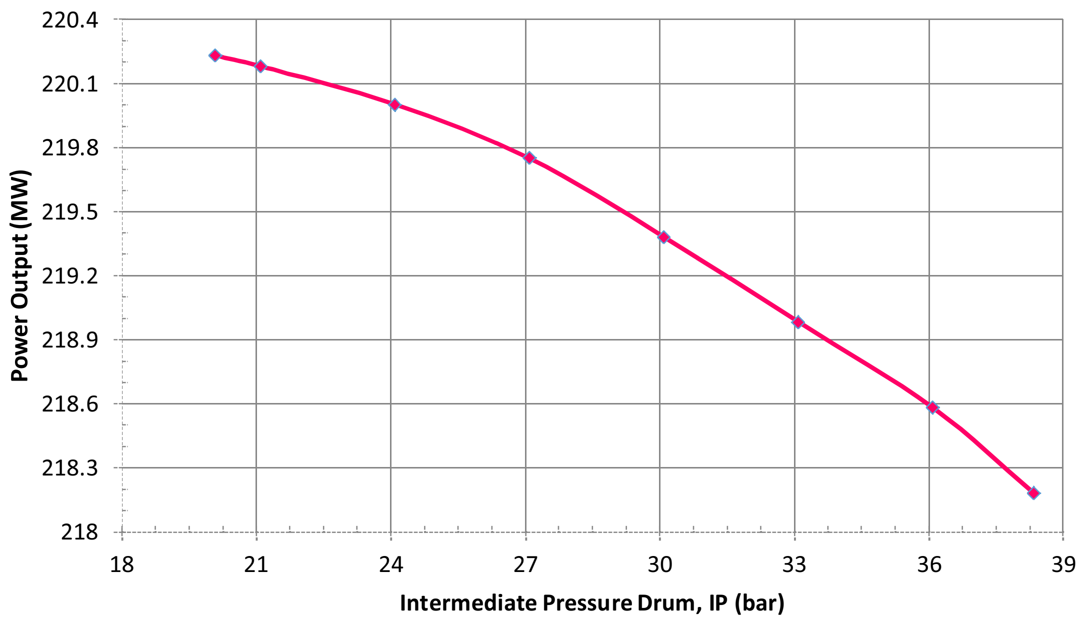


Fig. 43 - Effect of HRSG Intermediate Pressure on Power output of 220 MW CCPP

6.3.13 Effect of HRSG Low Drum Pressure on Combined Cycle Power Plant Performance

It has been determined from below plotted figure 44 that the dependence of combined cycle power output on HRSG drum low pressure in Rankine cycle.

The power output of the Rankine cycle (gross power output) will increase with increase in HRSG drum low pressure, however after attaining an optimum value, the power output will start decrease with increase in HRSG low pressure. We know our cycle has been design point of LP pressure is 8.4 bar with 220 MW.

The variation of LP drum pressure on Power output is summarised as:

- With LP drum pressure of 6 bar the Power output will be 217.63 MW
- With LP drum pressure of 7 bar the Power output will be 219.26 MW
- With LP drum pressure of 8 bar the Power output will be 219.85 MW
- With LP drum pressure of 9 bar the Power output will be 220.01 MW
- With LP drum pressure of 9.5 bar the Power output will be 219.93 MW

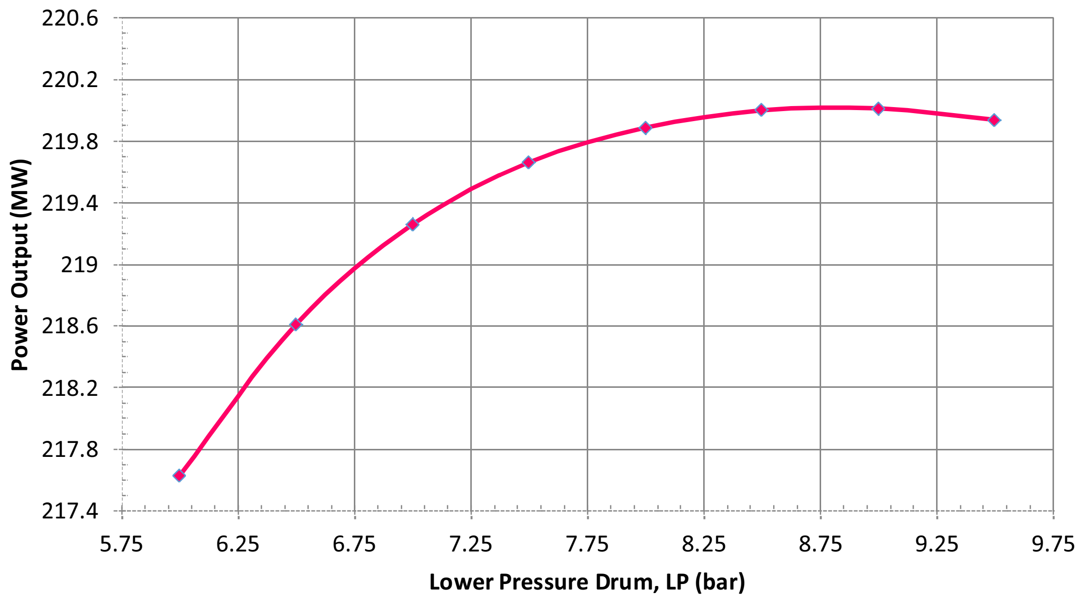


Fig. 44 - Effect of HRSG Low Pressure on Power output of 220 MW CCPP

CHAPTER – 7
CONCLUSION & RECOMMENDATION

7. CONCLUSION & RECOMMENDATION

The simulated thermodynamic modelling of triple pressure reheat type combined cycle power plant with supplementary fired has been developed and the influence of variation of various parameters on its performance is summarized below:

The impact of compressor ratio, Gas turbine inlet temperature and ambient temperature, HP steam parameters of Rankine cycle, deaerator temperature ratio, reheat pressure ratio, Pinch point, is focused for evaluate and estimation of thermodynamic designed combined cycle.

- It has been observed that with increase in temperature of ambient the performance of combined cycle was reduced in terms of power generation, plant heat rate, overall plant efficiency etc.
- For the increasing value of compression ratio, the plant efficiency of the cycle will increase but in parallel the power output will reduced. An optimum value of compression ratio need to be selected for efficient cycle because after a certain limit, further increase in compressor ratio cause reduction in performance of combined cycle.
- Turbine inlet temperature plays an important role in performance of combined cycle. Turbine inlet temperature keeps a linear correlation with the power output, plant efficiency, heat rate etc with the topping cycle & bottoming cycle. Increase in turbine inlet temperature cause increase in performance of combined cycle.
- The Steam parameters of Steam Turbine cycle also plays equal role to enhance the performance of Rankine cycle & overall combined cycle. With increase the steam property cause increase its enthalpy cause increase in performance of combined cycle.
- It has been observed that the performance of combined cycle will improved due to increase of steam reheat pressure ratio with different HP steam pressure. However as noticed an optimum value to be selected for efficient design of thermodynamic cycle.
- Also as described the deaerator temperature ratio of Rankine cycle also improved the performance of combined cycle. Upto an optimum value of deaerator temperature ratio, the performance got improved, after an optimum value further increase in deaerator temperature ratio, the performance starts reducing.
- The impact of pinch point on the power output of combined cycle as increase in value of pinch point cause reduction in power output of the combined cycle. It establish an

inversely proportion relation i.e. Null value of pinch point give maximum power output.

- The increased in HRSG high drum pressure cause increase in combined cycle power output. However after a certain range of effective value, the further increased in value of HRSG high pressure not increased power output with cost effective manner.
- An opposite to HRSG high pressure drum, the power output of combined cycle will reduced due to increase in HRSG intermediate drum pressure.
- Also variation in HRSG low pressure also cause increase in power output with increase in drum lower pressure. However after an optimal value the gross power output will reduced after further increased in value of HRSG drum lower pressure.

CHAPTER – 8
LIMITATION & SCOPE FOR FUTURE WORK

8. LIMITATIONS & SCOPE FOR FUTURE WORK

In this project an effort is being providing to design a thermodynamic modeling of Triple pressure reheat type combined cycle power plant having facility of supplementary firing. Every trait has been added up to keep in mind to improve the performance of the cycle which could be economically feasible & viable.

As we know by definition critical pressure is a point in Rankine when working fluid is directly converted into steam i.e. there is no transition phase to convert water into steam. So no evaporator is envisaged into the steam generator unit. The critical point of water occurring at 221.1 bar, 374°C.

The introduction of supercritical technology to meet the commercial targets is the challenges of the present scenarios in cost-effective manners. When we operate the Rankine cycle above critical point, cause more heat will introduce into the working fluids, thus energy available with steam will increase due to which thermal energy that will further used to meet the requirement of higher power output, overall plant efficiency and better heat rate per capita of infrastructure investment.

As we know the working fluid going to the steam turbine train is well below the critical point.

With being implementation of supercritical technology in the combined cycles of the present will termed as Super Critical Combined Cycles. The implementation of super critical technology in combined cycles has several advantages over conventional sub critical combined cycles.

These advantages of adopting the supercritical technology have the following advantages:

- Include simpler heat recovery units
- Higher fuel efficiency
- Reduced air pollutant emissions
- Less auxiliary power consumption
- Less water consumption
- Capability to generate more incremental power by fringe fuel gas in the heat recovery unit efficiently and capability to vary power output quickly to accommodate to changing power demand.

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