PERFORMANCE TEST VALIDATION OF COMBINED CYCLE POWER PLANT

A thesis submitted in partial fulfillment of the requirements for the Degree of
Master of Technology
(Energy Systems Engineering)

By

V.KRISHNA

College of Engineering

University of Petroleum & Energy Studies

Dehradun

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To Whom It May Concern:

This is to confirm that Mr. V. Krishna, Final year student of College of Engineering, University of Petroleum & Energy Studies, Dehradun was temporarily posted to our organisation at Bangalore for 3 months as part of his Post Graduation curriculum to complete his project work.

He has successfully completed the Modelling and Performance Test Validation of a typical 366 MW Combined Cycle Power Plant under our direct supervision in the Power Generation Mechanical division. He has developed the correction curves for usage during the performance guarantee test that will be carried out after successful commissioning of a power plant.

We found him to be a hard working, intelligent and dependable engineer and a quick learner. In particular, he has learnt the usage of Thermoflow software used for design engineering, optimisation and cost estimation of Gas Turbine based thermal power plants, conventional and cogeneration power plants in a very short time.

We wish him all success in his future endeavour.

D. Psu tt

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DECLARATION

This thesis work entitled "**Performance Test Validation of Combined Cycle Power Plant**" is my own work carried out under the guidance of **Mr. D. Pushpanathan** (Head of the Department, Mechanical) and **Mr. Pankaj Sharma** (Asst. Professor, College of Engineering Studies, University of Petroleum and Energy Studies, Dehradun). This work in the same form or in any other form is not submitted by me or by anyone else for the award of any degree.

V. Krishna

Date: 28/04/2011

V. Krishna

Synopsis

The performance of a combined cycle power plant is highly whimsical and is bound to be affected by numerous parameters such as variation in atmospheric conditions, changes in grid conditions, fuel's calorific value and temperature etc besides the operation of individual equipments such as gas turbines, steam turbine, condenser etc. This project is about evaluating the performance of a CCPP by plotting suitable correction curves and validating it with a reference CCPP project. The methodology involves modeling a 366MW CCPP and evaluating its performance at various simulated test factors. The correction curves based on its various output results obtained under different test conditions is compared with the standard test curves of 366 MW CCPP reference project for consistency and validation.

Acknowledgement

First and foremost I would like to express my sincere gratitude to Mr. G. Radhakrishnan, Director, FICHTNER consulting Engineers (I) Pvt. Ltd., Bangalore for providing me a magnificent opportunity to carry out my final year M.Tech project. His courteous and considerate demeanor has become a source of inspiration to me.

I remain indebted to Mr. D. Pushpanathan, Head of the Department for permitting me to carry out my project under his direct supervision in the Mechanical engineering department of FICHTNER consulting Engineers. Also the information that he shared during my short tenure provided me with an invaluable knowledge about power plants.

I owe a lot to Mr. Yerranna Duggapu, Senior Design Engineer, Mechanical Dept who took time in guiding me throughout this project despite his busy schedule. Without his guidance I would have never been able to complete this project.

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Last but not the least I have to thank every employee of Mechanical department of FICHTNER consulting Engineers who all made my short tenure here a memorable odyssey.

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Nomenclature

P	:	Pressure at the exit, in bar
T	:	Total Temperature, in Deg C
WB	:	Wet Bulb Temperature in Deg C
h	:	Stagnation enthalpy at the exit of the compressor, KJ/Kg
Cp	:	Specific heat,KJ/KgK
k	:	Specific heat ratio
CPR	:	Compressor Pressure Ratio
Q	:	Free Air delivered, m3/hr
S	:	Entropy (kJ/kg)
M	:	Mass flow in T/hr
Subscript 1	:	Parameters at the inlet to the compressor
Subscript 2	:	Parameters at the exit of compressor or inlet of combustor
Subscript 3	:	Parameters at the exit of the combustor or inlet of GT
Subscript 4	:	Parameters at the exit of gas turbine
Subscript 0		Represents stagnation quantity
ST	:	Steam turbine
GT	:	Gas Turbine
CCPP	:	Combined Cycle Power Plant
TIT	:	Turbine inlet temperature, Deg C

TET	:	Turbine inlet temperature, Deg C
CIT	:	Compressor Inlet Temperature, Deg C
CW	:	Circulating water
FW	:	Feed Water
HP	:	High Pressure
IP	:	Intermediate Pressure
LP	:	Low Pressure
RH	:	Reheat
HPT, IPT, LPT	:	HP Turbine, IP Turbine, LP Turbine
GSC	:	Gland steam Condenser
Vs Lk	:	Valve Steam Leakage
HRSG	:	Heat Recovery Steam Generator
EC	:	Economizer
EV	:	Evaporator
SH	:	Super heater .
PP	:	Pinch point
KW	:	Power in kilo Watts
KWe	:	Electrical power output in Kilo watts
KWth	:	Thermal power output in Kilo watts
G	:	Generator

Qth	:	Total thermal power in Kilo watts
LHV	:	Low Heating Value, (KJ/Kg)
HHV	:	High Heating Value, (KJ/Kg)
HMBD	:	Heat and Mass Balance Diagram
ASME	:	American Society of Mechanical Engineers
PTC	:	Performance Testing Codes
Abs	:	Absolute
Atm	:	Atmospheric
Amb	:	Ambient
γ	:	Specific heat ratio
ή	:	Efficiency in Percentage

Chapter 1

Introduction to Combined Cycle Power Plant

1.1 Overview

A combined cycle power plant is an integration of more than one thermodynamic cycle to efficiently convert the fuel's chemical energy into electrical energy. CCPP gains prominence owing to its heat rate as low as 6400KJ/Kg and relatively ease of installation and maintenance issues. In Today's context, a CCPP mostly refers to the combination of Brayton cycle, as the Toping cycle and Rankine cycle, as the bottoming cycle.

Brayton Cycle

Gas turbines usually operate on Brayton cycle. The axial compressor draws the Atmospheric air at ambient conditions and compresses it to develop a high pressure. The pressurized air flows into the combustor and is burnt along with the fuel, at constant pressure. The consequential high temperature gases enter the turbine, where they expand through a row of fixed and moving blades namely nozzles and vanes, resulting in the rotation of the turbine shaft and thereby generating electricity. The hot flue gases exits the turbine at constant pressure.

Rankine Cycle

Steam Turbines operate on the principle of Rankine cycle. Water at high pressure, serving as the working medium, enters a boiler where it is heated at constant pressure until it becomes superheated vapor. The superheated vapor is allowed to expand in a steam turbine to vacuum, thereby rotating the ST and generating power. The expanded vapor is then condensed at constant pressure to become a saturated liquid. The saturated liquid is again pumped back to the boiler. In a CCPP, the boiler is replaced by a HRSG in which the source of heat is the exhaust gases from the turbine. This chapter details the working of the following, major components of the CCPP.

- ✓ Axial Compressor
- ✓ Combustor
- ✓ Gas Turbine
- ✓ HRSG

- ✓ Steam Turbine

1.2 Axial Compressor

Compressors are mechanical devices that raise the pressure of the incoming gas by compressing and reducing its volume. In axial compressors the inlet air flows parallel to the axis of rotation of shaft. Axial compressors are preferred in CCPP and gas turbines owing to their ability to handle high flow. The inlet air is first accelerated and subsequently diffused in order to achieve high pressure in axial flow compressors. Axial compressor compresses air utilizing multiple stages of compression and they collectively develop high compression ratios with high efficiencies. The diffusion in the stator converts the velocity increase gained in the rotor to a pressure rise. An axial flow compressor is made up of several alternating rows of rotating airfoils and stationary blades called rotors and stators respectively. Together a stator- rotor pair set is called a compressor stage and each stage raises the pressure slightly.

The construction of axial compressor is such that the rotor adds swirl to the air flow, thereby progressively increasing the total stagnation pressure carried in the flow by increasing the angular momentum (adding to the kinetic energy associated with the tangential or swirl velocity).On the other hand the stator removes the swirl from the flow and raises the static pressure (internal energy) of the air flow by converting the kinetic energy associated with the swirl. The rotors are attached to the shaft and rotate at the same speed of the shaft (and the gas turbine).The stators do not rotate .i.e.; they are fixed to the casing of the compressor and thus cannot add any net energy to the flow but diffusing the flow and only diffuses the flow.

All gas turbines have variable inlet guide vanes at the first stage of stationary row whose angle can be varied by the fuel control unit. At lower speeds of rotation the incident angle of air should be high whereas at higher speeds of rotation, the angle of incidence should be lower in. This is needed in order make the air meet the compressor at an optimum angle that ensures stall free operation of the compressor. It should be noted that the IGV as such adds no net energy to flow of inlet air but it just swirls the inlet air and lowers its Mach number relative to the rotor blades thereby augmenting the rotor's aerodynamic performance.

The efficiency of the gas turbine increases in direct proportion to the compressor pressure ratio. Also for a particular firing temperature, the GT's exhaust temperature is largely determined by the work required to drive the compressor that is, in turn, affected by the "compressor pressure ratio". Hence it is desirable to have a large pressure ratio and a lesser exhaust temperature.

In GE9FA machine, the inlet air, at standard Atmospheric conditions, progressively raises its pressure across the 17 stages axial compressor until the exhaust pressure rises up to 15.8 bar under base load reference and design conditions. This machine consumes up to 50 Percent of the power that is produced by the turbine.

1.3 Combustors

In a gas turbine, the compressor feeds the high pressure air into the combustion chamber where it is heated at constant pressure owing to combustion reaction.

A Combustor is designed in such a fashion that it

- ✓ It completely combusts the fuel gas.
- ✓ Ideally a combustor should not have any pressure drop across it. The pressure drop across the combustor reduces the pressure at the turbine inlet which in turn affects the pressure ratio and efficiency of the gas turbine.
- ✓ Very high temperature in the range 1950 Deg C can be easily reached inside the combustion chamber & hence the combustor materials should essentially withstand these higher temperatures.
- ✓ The flame originating inside the combustor should be contained the flame inside the combustion chamber in order to avoid the damaging of turbine blades can be due to the higher temperature,
- ✓ Uniform exit temperature profile in order to avoid hot spots in the exhaust gas flow. The turbine blades will experience thermal stress in case of hot spots in exit flow.
- ✓ Extensive operational range in terms of both environmental as well as ambient factors.
- ✓ The emissions are kept low.

Types of Combustors

There are three main combustion chamber types in use today

- ✓ Annular combustor chambers
- ✓ Can combustor chambers
- ✓ Can-Annular combustor chambers

Annular combustor chambers

Annular combustors have an unbroken annulus separating the outer casing and the inner lining of the combustion chamber. The secondary air or the cooling air passes through these concentric annular space provided between the liner and the inner combustion chamber housing. This air prevents the combustor materials from attaining dangerous temperature levels. Primary air mixes with the natural gas and flows through the snout. Annular combustors provide a geometrically compact design as the combustor provides a larger combustion volume.

Can combustor chambers

Can type combustors are constructed with individual combustion chambers. Air from the axial compressor flows into the separate combustion mounted around the periphery of the machine through concentric cylindrical tubes between the inner liner and the outer chamber. The combustion takes place inside the inner liner and the air flow is controlled by the holes and louvers in the outer chamber.

Can-annular combustion chamber

A can annular combustor combines the advantages of both can type and annular type combustor. A number of cans are mounted around the periphery of the machine with a common annulus (i.e. an outer shell).Primary air mixes with the fuel and is ignited by means of an igniter plug. Secondary air passing through the holes and louvers cools the combustor. The flame is carried to the rest of the liners by means of cross over tubes. The inner combustion chamber casing provides mechanical support as well as acts as heat shield by containing the flame.

1.4 Gas Turbines

In axial turbines, the inlet hot combustion gas flows parallel to the axis of rotation of the turbine shaft. The gas turbine utilizes the enthalpy present in the hot combustion gases, to rotate the turbine (& generator) and compressor, by extracting the kinetic energy present in them.

An axial flow compressor comprises of alternative rows of rotating blades and stationary nozzles called rotors and stators respectively and together a stator- rotor set is called a turbine stage. As is the case in compressors, the rotor blades are attached to the rotating turbine shaft whereas the stationary blades are fixed to the turbine casing.

Generally Turbines are classified into three types

- ✓ Impulse Turbine
- ✓ Reaction Turbine
- ✓ Impulse-Reaction turbines

In Impulse turbines, the impact generated by the kinetic energy of the hot combustion gases is used to rotate the turbine. The stationary nozzle guide vanes, increases the velocity of the medium at the cost of its pressure, and also directing the flow of the combustion gases. Hence there is no pressure drop between inlet and exit of the rotor. The impulse effect is more pronounced at the initial stages of steam turbine.

In reaction turbine, work on the basis of Newton's third law of motion. The rotation of the turbine blades is achieved by the reaction force created by convergent shaped passage between the rotor blades and stator nozzles .Hence a pressure drop is experienced the flue gases as it passes through both fixed nozzles and rotating blades. In the reaction turbine, the fixed nozzle, only alter the direction of flow of the gases. The reaction effect is more pronounced at the latter stages of steam turbine.

Most of the present day turbines are a combination of both impulse-reaction turbines where the rotational motion is created utilizing both the impact and the reaction force. The 1st stage blades are of impulse design whereas the secondary stages are of reaction type.

Metallurgy Limits TIT

The guiding parameter in the maximum power that a turbine can produce at a given pressure ratio is dependent on Turbine inlet Temperature (TIT) but metallurgical limitations prevent the turbine from reaching very high temperatures.TIT in turn is a function of the firing temperature, compression ratio, mass flow, and centrifugal stress. So these factors limit size and ultimately, efficiency. A rough rule of thumb is that 55°C increase in firing temperature gives a 10 to 13 percent power output increase and a 2 to 4 percent efficiency increase. The combustion chambers and the turbine first stage stationary nozzles and blades are therefore the most critical areas of the turbine that determine its power output and efficiency.

1.5 Heat Recovery Steam Generator

In a CCPP, as a general guideline, the gas turbine approximately represents 66% of the plant's electrical output while the remaining is generated from the steam turbine, assuming that the HRSG does not employ a duct burner.

Heat Recovery Steam Generator generates steam by utilizing the heat energy of the combusted flue gases. The condensate from the condenser of the steam turbine along with the makeup feed water enters the deaerator and follows the path of

Economizer -----Evaporator-----Super Heater-----Steam Turbine-----Condenser

The exhaust flue gases from the exit of the gas turbine enters the HRSG where its enthalpy is used to heat the steam flowing in the super heaters, the water-vapor mixture in the boiler evaporator and water in the economizer before it enters the stack.

Steam generated in the HRSG flows from the super heater and to the high pressure turbine. The cold reheat steam from the HP turbine flows gets re-superheated in the reheater and flows to IP turbine. Pinch points and approach temperatures are important HRSG design parameters. Reducing these temperatures will increase cycle efficiency. However, optimization involves fairly complicated heat transfer calculations and steam cycle heat balances to avoid operational problems.

A HRSG can be laid parallel to the ground with the flue gases flowing past the economizers, evaporators and super heaters and leading in to the duct. On the other hand HRSG's can also be constructed incorporating the economizers, evaporators and super heaters within the stack structure. This is called as vertical HRSG. Horizontal HRSG will be up to 60 meters long and 25 meters high. Thermal expansion of the ductwork is a significant design issue because of the size and the need to preserve the internal insulation.

As huge amounts of combusted flue gases flow across the HRSG, a heavy ductwork accommodating flue gases up to 600kg/s is required.

Table 1 : Basic outline about the costs, pressure levels & heat rate of the CCPP

Costs	Level	Heat Rate
Low	1 Pressure Level, non reheat	High
Low to medium	2 Pressure Levels, Non-Reheat	Med/High
Medium	3 Pressure Levels, Non-Reheat	Medium
Medium to high	3 Pressure Level, Reheat	Med/Low
High	3 Pressure Levels, Reheat	Low

Table 2 : Comparison between steam turbine sizes in HRSG

		Non Reheat Three Pressure			Reheat Three Pressure	
Steam Turbine Size	MW	≤ 40	<40	>60	61	>100
Throttle Pressure	bar	56.4	66.1	82.6	96.4	124
HP Approach temperature	Deg C	25	25	25	39	28
Reheat Pressure	bar				20.6-27.5	31
Reheat Temperature	Deg C				538	554
IP Admission Pressure	bar	7	8	11	20.6-27.5	31
IP Temperature	Deg C	11	11	11	11	11
LP Pressure	bar	1.7	1.7	1.7	2.8	2.8
IP Approach temperature	Deg C	11	11	11	11	11

Heat Transfer and Tube Finning

The differential temperature between the flue gases and water behind the evaporator is comparatively poor. Further the heat transfer is affected by the subsided heat transfer coefficient of flue gas owing to the low temperature.

Table 3 : Heat transfer coefficients of various sections of HRSG

Section of HRSG	Flue Gas	Economizer	Evaporator	Super heaters
Heat Transfer coefficient (W/m2 K)	50 (W/m2 K)	500 (W/m2 K)	2500-10000 (W/m2 K)	1000 (W/m2 K)

It follows from this that the tube wall temperatures tend to run quite close to the water side temperatures at the economizers. Even when the temperature differentials are improved, the heat transfer rates are very modest .Under such poor flue gas side heat transfer rates tubes must be of small diameter with tight spacing and be of the finned type to provide sufficient heat transfer area. The only section of HRSG that might not use finned tubes is the HP super heater where there are better heat transfer coefficients and where there might be a possibility of oxidation of the filming.

Ideally, a HRSG design should satisfy the following criteria

- ✓ The surface area of the heat exchangers should be sized to in order to be consistent with the heat transfer rate computed from the heat balance solution.
- ✓ The total gas-side pressure drop, gas mass flux, or geometry (tube length, aspect ratio, or both tube length and duct width) assumed for the entire boiler should not fall below the maximum allowable pressure drop
- ✓ A HRSG with reasonable geometric proportions and having an optimal balance between efficiency and cost is preferred.
- ✓ The water/steam-side pressure drops in the economizers, super heaters, or reheaters should be matched

- ✓ A physically-realistic design, with standard sized finned tubes and with an integer number of tubes in each row and an integer number of rows in each heat exchanger should be made.

Importance of Pinch Temperature Difference

Pinch is the local temperature difference between the saturated steam in an evaporator and the cooled flue gas leaving that evaporator. Smaller pinch temperature differences at any given evaporator result in greater cooling of the gas by that evaporator, generating more steam at that pressure level. This improved heat recovery comes at a rapidly growing cost, since each incremental increase in heat transfer is accomplished with a smaller temperature difference, requiring successively larger increments in surface area. Higher pinch temperature on the other hand results in inefficient heat transfer. Hence it becomes mandatory to select an optimal pinch temperature difference with a tradeoff between cost and efficiency.

Importance of Approach Temperature Difference

Approach temperature difference is the extent by which the final economizer exit temperature falls short of saturation temperature at the evaporator pressure. Larger values require less economizer surface, but also reduce steam production in pinch-limited heat recovery. In pinch-limited heat recovery, reducing the approach sub cooling causes more of the sensible heating of the water to be accomplished using energy of the flue gas below the pinch, liberating the hot gas above the pinch from heating sub cooled water, thereby allowing it to evaporate more water and produce more steam. Very small approach subs cooling, and the corresponding larger economizer surface, make the economizer more susceptible to steaming at off-design conditions.

In general, optimizing pinch point and approach temperatures follows the selection of the pressure level. Decreasing the pinch point and approach temperature results in higher efficiency, but higher capital cost.

1.6 Steam Turbines

A steam turbine is a mechanical device that converts thermal energy in pressurized steam into useful mechanical work. The steam turbine derives much of its better thermodynamic efficiency because of the use of multiple stages in the expansion of the steam. This results in a closer approach to the ideal reversible process.

Steam Turbine Principle

The steam energy is converted to mechanical work by expansion through the turbine. The expansion takes place through a series of fixed blades (nozzles) and moving blades and each row of fixed blades and moving blades is called a stage. The moving blades rotate on the central turbine rotor and the fixed blades are concentrically arranged within the circular turbine casing which is substantially designed to withstand the steam pressure. To maximize turbine efficiency the steam is expanded, generating work, in a number of stages. These stages are characterized by how the energy is extracted from them and are known as either impulse or reaction turbines. Most steam turbines use a mixture of the reaction and impulse designs: each stage behaves as either one or the other, but the overall turbine uses both. Typically, higher pressure sections are impulse type and lower pressure stages are reaction type.

Differences between fossil fuel fired ST and CCPP's ST machines

- ✓ The CC steam turbine admits steam at three distinct pressure levels, in order for the heat-recovery steam generator (HRSG) to extract the maximum amount of thermal energy from the gas-turbine exhaust.
- ✓ The CC steam turbine typically features axial exhaust using only two standard cylinders. For most projects, this yields a compact design, reduced manufacturing costs and a shorter erection schedule.
- ✓ As feed waters heaters are not present in CCPP, the turbine exhaust flows are much higher than a conventional utility turbine exhaust flows. Consider that the low-pressure (LP) exhaust steam flow in a CC plant can be up to 35 percent greater than the main steam flow. This effects are pronounced in the last-stage blades
- ✓ CCPP provides the advantage of combining more than one gas turbine/HRSG to a steam turbine. More than 1 gas turbine/HRSG can be connected to a steam turbine in case of need of higher power generation.
- ✓ The capital cost of the CCPP can be lowered by locking the rotor the gas and steam turbines to form a single shaft and this configuration is called as single shaft plant
- ✓ Because the steam turbine cannot match the ramp rates of a quick-starting gas turbine, the CC plant must incorporate large-and often problematic-cascading steam-bypass systems.

There are significant pressure drop across the tip of the bucket due to the reaction force generated by the steam flowing past the buckets. This causes a leakage of steam called as tip leakage losses. Also low amount of losses are suffered because of the leakages of steam through the diaphragm shaft packing and shaft end packing results in losses.

The losses in stationary flow path, such as inlets, valves, and exhausts are of considerable importance. Pressure drops in this mechanism will also inflicts losses in the extracted available and hence the derived work output.

1.7 Literature Review

Fred Starr of European Technology Development in April 2003 studied about the background to the design of HRSG systems and implications for CCGT plant cycling. His work provides important source of information pertaining to the heat transfer coefficients on both Feed water side and flue gas side and the need for extended surfaces.

L.O. Tomlinson and S. McCullough of GE Power Systems in 1996 studied on Single Shaft combined cycle power generation system. This provided important source of information pertaining to the pressure levels, maximum temperature, etc attained in GE machines

M. Khosravy-el Hossani, Q. Dorosti in April 2009 discussed about Improvement of Gas Turbine Performance Test in Combine Cycle This study details the method for minimizing the effect of ambient factors that influence the performance of the CCPP.

A Ragland,Vogt-NEM W.Stenzel in year 2000, worked on Combined Cycle Heat Recovery Optimization. The study provides the basic information about selection the pressure levels of the HRSG and also explains basic information about optimization of pressure levels in HRSG

Christian Engelbert and Batu Goker of Siemens Industry in May 2010 studied the evolution of the gas turbines and the consistently decreasing emissions of GT because of changes in combustors design

V.Ganapathy of BHEL in his article "Heat Recovery Steam Generators -Understanding the Basics" defines about the basics of the economizer and problems associated with the steaming in the economizer .This article contains vital source of information about the basics of HRSG and about the occurrence of steaming in Economizers

Paul Hurd & Frank Truckenmueller of Siemens Westinghouse Power Corporation in April, 2005 discussed the modern reaction HP & IP turbine technology. This paper introduces the features, benefits of using an opposed HP/IP Cylinder in steam turbine cycles

V.Ganapathy of BHEL in his paper "Understanding HRSG Temperature Profiles" provides vital source of information about the importance of Pinch and Approach Temperature differences in HRSG

Dave Colegrove ,Paul Mason, Klaus Retzlaff,Daniel Cornell of GE Systems in their article in 05,2001 about structured steam turbines for combined cycle market which introduced the GE H class turbines that operated with an efficiency of 60%

Advanced gas Turbines technology discusses the usage of humid air for the suppression of NOx in combustor. The paper November 2000 and also this paper provides insight into the temperature withstand capability of Turbine blades and the possibility of temperature reaching 1400 in 1st stage.

Chapter 2

Performance Parameters of CCPP

2.1 Overview

The standard conditions used by the gas turbine industry are 15 C, 1.013bar and 60% relative humidity, which are established by the International Standards Organization (ISO) and frequently referred to as ISO conditions. ISO conditions are seldom achieved in practice and most of the times; a CCPP operates at off design conditions. Also at off design conditions, variations in the operation of Brayton cycle in turn affect the performance of the bottoming cycle thereby affecting the whole CCPP. Hence an evaluation of the Brayton cycle & CCPP using variable Atmospheric conditions, variable grid conditions become necessary in order to estimates the correctness of its performance. If such an analysis is applied to the CCPP, the results can be displayed as a plot of cycle efficiency vs. specific output of the cycle.

There are numerous parameters that may affect the performance of a combined cycle power plant. Largely the parameters that affect the performance of a combined cycle power plant can be classified as

- ✓ Variable Grid Conditions
- ✓ Variable Ambient Conditions
- ✓ Variable inlet Fuel Conditions
- ✓ Plant ageing
- ✓ Pressure Drops & Environmental conditions

2.2 Variable Grid conditions

Besides faults, the most common variations in grid are related to frequency and power factor.

- ✓ Variation in Grid Frequency
- ✓ Variation in Power Factor

Variation in Grid Frequency

An alternator also called as synchronous generator, runs in parallel with numerous synchronous generators connected to the same AC grid, delivering power, at a speed determined by the following equation.

N = (120 * F)/P,

Where

- ✓ N is the speed of the generator rotor, in RPM
- ✓ F is the frequency of the grid, in Hz
- ✓ P is the number of poles of the generator, (2 for a GE9FA machine)

As numerous generators feed the grid simultaneously and several ports draw power from the grid at the same time, small variations are largely inconsequential as far as the speed of the generator is concerned unless there is a large disturbance that significantly affects the grid frequency and hence the generator speed.

As electricity cannot be stored the amount of generated power should always match the demand of the grid. As the load being independent of the power fed in the grid, all generators connected to the grid work in tandem to match the load. If the total generation, exceeds the total load demand of the grid, the grid frequency will increase, or if the amount of generation is less than the amount of the load the grid frequency will decrease.

The power (Watts) from a generator is given by the equation: P = V * I*Cos (phi),

Where,

- ✓ P is the power ,in Watts
- ✓ V is terminal voltage of generator ,in Volts and
- ✓ I is armature current ,in Amps

As two of the aforesaid factors (V, Cos Phi) being predetermined to vary within prescribed limits, the only option for the generator to produce more power is to increase the amps flowing in the generator's armature. The generator armature current can be increased only by supplying more torque from the prime mover into the generator i.e. higher the torque, more is the amps & more is the power. Hence the variation in frequency of the grid needs to be continuously monitored to vary the power produced in accordance with it.

Variation in Power Factor

Power factor of the generator is determined by the reactive load i.e., the total capacitive & inductive loads connected to the grid. The operating PF of the alternator connected to the grid can be controlled by varying the amount of excitation applied to the generator field (usually the rotor). ON the other hand the amount of excitation is related to the generator terminal voltage. If the amount of excitation being applied is exactly equal to the amount required to make the generator terminal voltage exactly equal to the grid voltage (at the generator terminals), the power factor will be 1.0 (or, unity). If the excitation is less than required, the power factor will be less than 1.0, and usually is considered negative, and the power factor will be leading. If the excitation is more than required, the power factor will be less than 1.0, and usually considered positive, and the power factor will be lagging. (Note: This is from a generator perspective.)

When you have a unit connected to a large grid and it is not being operated in power factor control mode, and the excitation being applied to the generator rotor is stable, the real power output (watts, KW, MW) is stable, and the power factor changes, that's because the grid voltage is changing. (Grid voltage changes throughout the day, all day long, on most grids.) To maintain a certain power factor, one has to adjust the generator excitation to keep the generator terminal voltage in the desired relationship to system voltage (as determined by the power factor). Also, usually when generators are loaded or unloaded, if the excitation is held constant the terminal voltage will change and that will affect the power factor (called armature reaction).

2.3 Variable Ambient Conditions

- ✓ Variation in Ambient air temperature
- ✓ Variation in Ambient air humidity
- ✓ Site Elevation

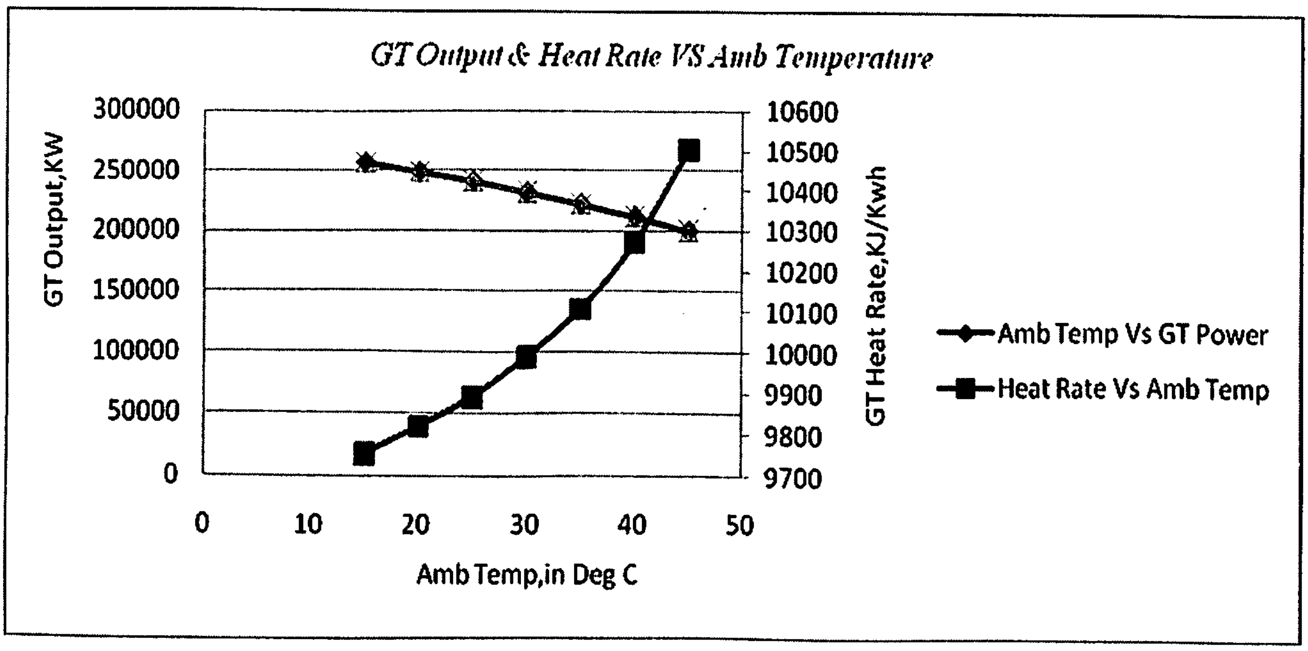
Variation in Ambient Air temperature

The gas turbine performance, considered to be constant air flow machine, is affected by anything the affects the mass flow rate of air or the density of the air. The mass flow rate of air is affected by changes in the ambient weather conditions. An increase in ambient temperature (assuming no changes in Relative humidity) decreases the density of the air and therefore the mass flow rate of the air and vice versa. Every turbine model has its own temperature-effect curve, as it depends on the cycle parameters and component efficiencies as well as air mass flow.

(Source: Simulated through GT PRO Software)

Table 4: Performance of GE9351FA at various ambient Temperatures

Ambient temperature	C	15	20	25	30	35	40	45
Compressor inlet mass flow	t/h	2331.4	2277.5	2223.7	2162.4	2097.3	2029.1	1956.7
GT fuel flow	t/h	50.01	48.73	47.52	46.2	44.85	43.44	41.91
Turbine inlet temperature	C	1327.2	1326.1	1325.2	1324.2	1322.8	1321.5	1319.1
Turbine exhaust mass flow	t/h	2381.5	2326.2	2271.2	2208.6	2142.2	2072.5	1998.7
Turbine exhaust temperature	C	603.9	607.9	612.2	617.6	623.3	630.1	637.2
GT gross power	kW	256662	248388	240435	231504	222118	211690	199644
GT gross LHV eff	%	36.92	36.66	36.4	36.04	35.63	35.05	34.27
GT gross heat rate	kJ/kWh	9752	9819	9891	9988	10105	10271	10505



Graph 1: Performance of GE9351FA at various Ambient Temperatures

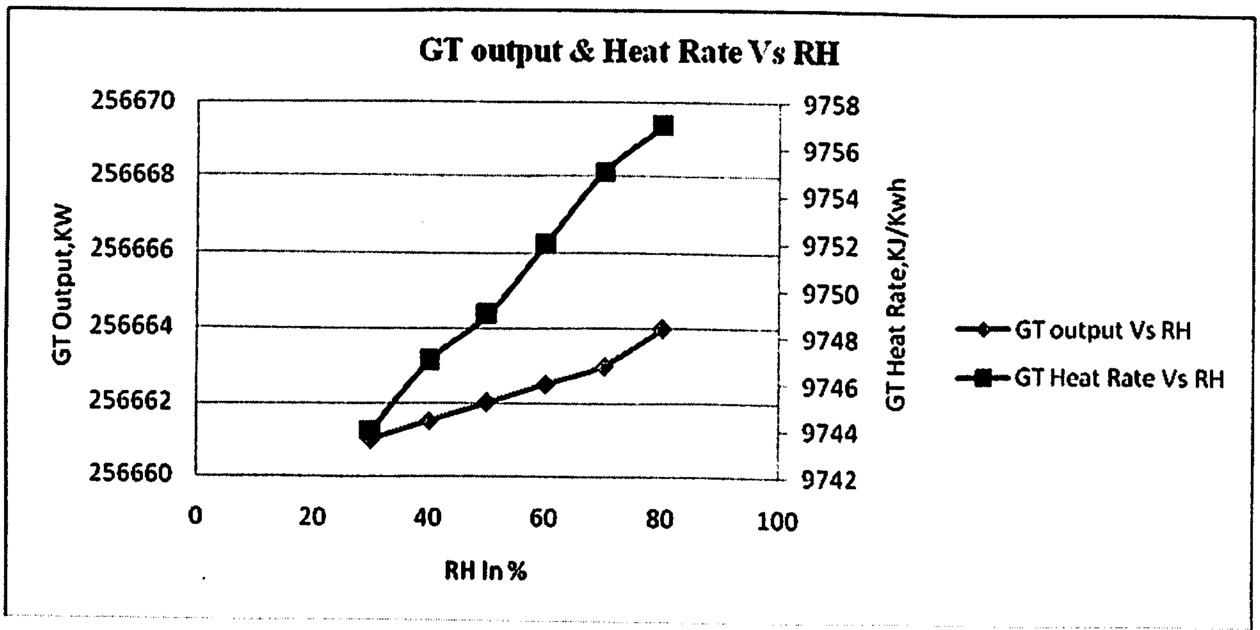
Variation in Ambient Humidity

The density of air decreases with increase in the humidity and hence the power output from the machine and the heat rate. But the effects of humidity are more pronounced in machines of higher rating.

(Source: Simulated through GT PRO Software)

Table 5 : Performance of GE9351FA at various RH at 15^{0}c

Relative Humidity	%	30	40	50	60	70	80
Ambient Temp	Deg C	15	15	15	15	15	15
GT gross power	kW	256661	256662	256662	256662	256663	256663
GT gross LHV efficiency	%	36.95	36.94	36.93	36.92	36.9	36.89
GT gross heat rate	kJ/Kwh	9744	9747	9749	9752	9755	9757
Compressor inlet mass flow	t/h	2335.9	2334.4	2332.9	2331.4	2330	2328.5
Turbine inlet temperature	C	1328.2	1327.9	1327.5	1327.2	1326.8	1326.5
Turbine exhaust mass flow	t/h	2385.9	2384.4	2382.9	2381.5	2380	2378.5
Turbine exhaust temperature	C	603.6	603.7	603.8	603.9	603.9	604
GT fuel flow	t/h	49.97	49.98	50	50.01	50.03	50.04



Graph 2: Performance of GE9351FA at various RH at 15^{0}c

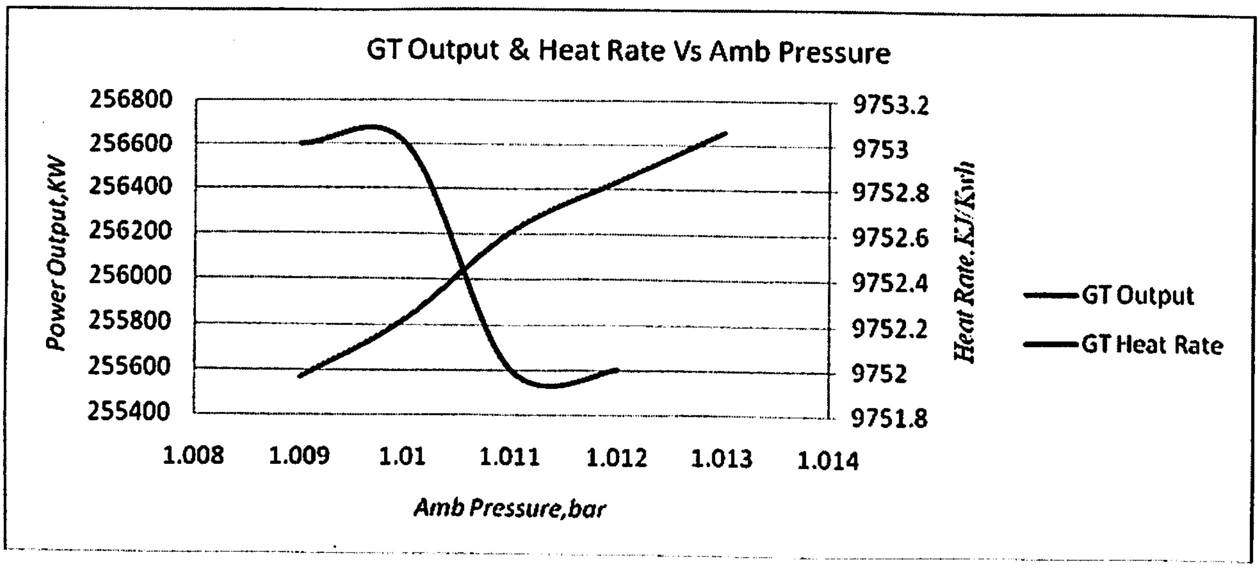
Atmospheric pressure and Site Elevation

The density of air decreases with the increase of the elevation of the site. Hence as the site elevation increases, the Atmospheric pressure decreases and thereby inlet air flow reduces resulting in the decrease of the power output.

(Source: Simulated through GT PRO Software)

Table 6 : Performance of GE9351FA at various atmospheric pressures at 15^0c

Ambient Pressure	Bar	1.013	1.012	1.011	1.01	1.009
Ambient temperature	C	15	15	15	15	15
Compressor inlet mass flow	t/h	2331.4	2329.4	2327.4	2323.3	2321.3
GT gross power	kW	256662	256437	256211	255825	255566
GT gross LHV eff	%	36.92	36.91	36.91	36.91	36.91
GT gross heat rate	kJ/kWh	9752	9752	9752	9753	9753
Turbine inlet temperature	C	1327.2	1327.2	1327.2	1327.2	1327.2
Turbine exhaust mass flow	t/h	2381.5	2379.4	2377.3	2373.1	2371.1
Turbine exhaust temperature	C	603.9	603.9	603.9	603.9	603.9
GT fuel flow	t/h	50.01	49.97	49.93	49.84	49.8



Graph 3 : Performance of GE9351FA at various atmospheric pressures at 15^0C

From these reading it can be inferred the inlet flow varies at almost a constant rate of 2Tons/hr for every 0.001 bar

2.4 Variation in quality of Fuel

The power output from the gas turbine is given by the product of mass flow rate of the combustion gases, specific heat of the combusted gas (Cp), and difference in temperature across the turbine. With other factors remaining the same the power output from the gas turbine can be increased by raising the heat energy content (Specific heat) of the flue gases. Higher the water vapor content produced by the higher hydrogen/carbon ration of the methane more is the specific heat. Hence a fuel with higher C: H ratio is preferred and this may compensate the lower mass flow rate. Further the output of the GT increases (& also the LHV) when the fuel is completely made up of carbon and methane with the absence of inert gases and no oxygen atoms. As said earlier, the effects of specific heat become more pronounced than that mass flow.

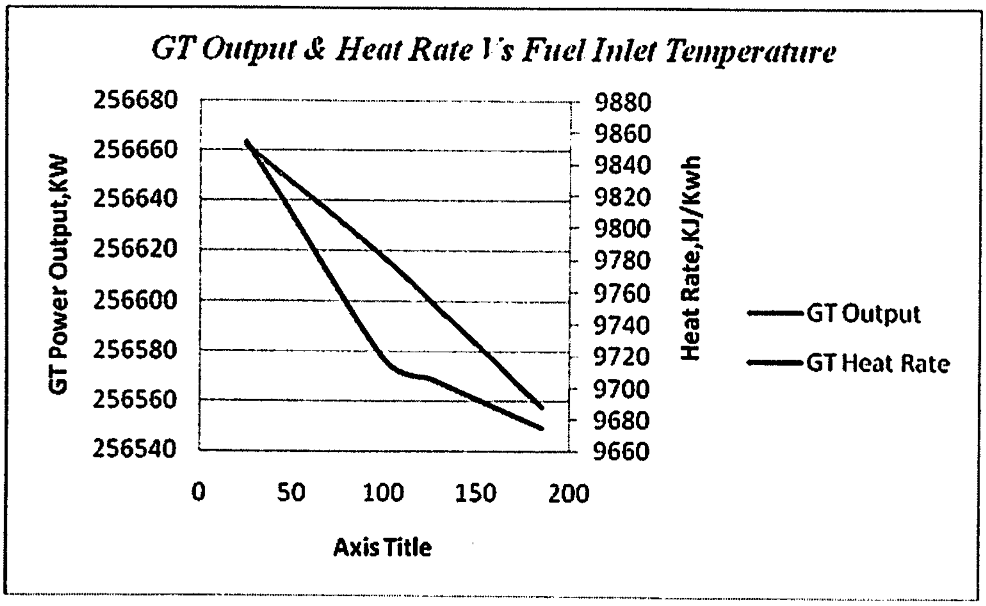
Fuel Heating

A heated fuel tends to carry a higher calorific value and hence some GT's use fuel heating in order to reduce the quantity of fuel required. It should be noted that by heating the fuel, for attaining the same firing temperature we are using lesser quantity of fuel than is actually required owing to its higher calorific value but at the same time the total mass flow decreases and this may produce lesser power output from the gas turbine but with an improvement in the heat rate of the machine. Either external source or IP feed water from HRSG can be used to heat the fuel. Experimental results by GE prove that the CCPP efficiency increases up to 0.6% because of fuel heating by performance heater.

(Source: Simulated through GT PRO Software)

Table 7 : Performance of GE9351FA at various fuel Inlet Temperature

Fuel Inlet Temperature	Deg C	25	100	125	150	185
GT gross power	kW	256662	256617	246600	256583	256558
GT gross LHV efficiency	%	36.92	37.04	37.09	37.14	37.21
GT gross heat rate	kJ/Kwh	9854	9718	9706	9693	9675
Compressor inlet mass flow	t/h	2331.4	2331.4	2331.4	2331.4	2331.4
Turbine exhaust mass flow	t/h	2381.5	2381.2	2381.1	2381.1	2381.1
GT fuel flow @26.42 bar	t/h	50.01	49.83	49.76	49.7	49.6
Cal of Fuel	KJ/Kg	50047	50220	50283	50348	50442



Graph 4 : Performance of GE9351FA at various fuel Inlet Temperature

2.5 Plant ageing

The performance degradation of gas turbine & CCPP is primarily due to continuous usage over the ages and all machines suffer degradation and performance losses with respect to time.

The performance degradation losses can be classified as

- ✓ Recoverable losses
- ✓ Non Recoverable losses

Those performance degradation losses like fouling of the blades of compressor that can be partially rectified by water washing or fully by mechanical cleaning are called recoverable losses.

Those performance degradation losses like increase in compressor and turbine clearances over the years, degradation of the surface finish of the turbine etc., that reduce the component efficiencies and cannot be rectified by cleaning and other procedures fall under the category of non recoverable loss. Replacement of parts is the only possible solution in these cases.

Quantification of ageing losses is cumbersome since obtaining field data is difficult and further this loss is exacerbated by the addition of performance affecting factors related to ambience.

As per Industrial standards, the performance degradation for the turbo machinery should usually lie between 2 to 6 % in the first 24,000 hours of operation, when measured during performance test corrected to guaranteed conditions. This assumption is valid if the degenerated components are not rectified or replaced. In case of replacement, the performance degradation should lie between 1 to 1.5 Percent.

2.6 Pressure drop

The pressure of the air at the inlet to the compressor drops as a result of air filters, evaporative coolers, silencers etc. It is the measure as the pressure difference between the ambient air pressure and the pressure of air at the inlet to compressor. The performance of the CCPP deteriorates as pressure drops at the inlet increases because the density of the air reaching the compressor drops with absolute pressure at the compressor inlet.

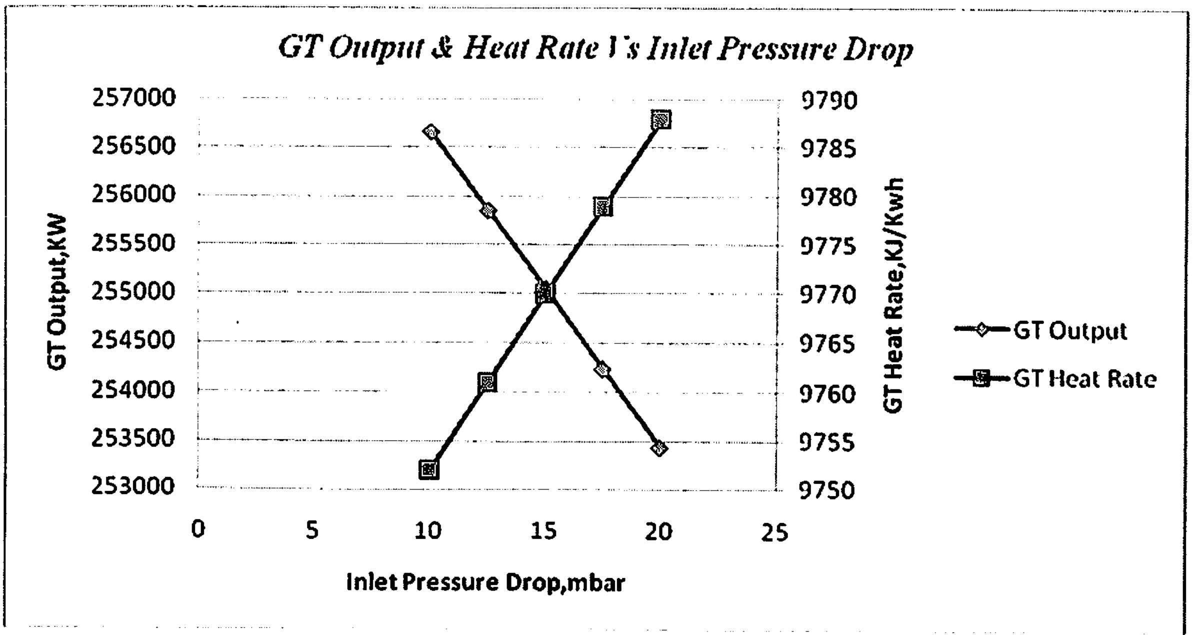
The pressure of the flue gases at the exhaust of the turbine drops as a result of HRSG, silencers etc. It is the measure as the pressure difference between the ambient air pressure and the pressure of the flue gases at the exhaust of the turbine.

Lower air density = less mass flow = less power output & lower efficiency.

(Source: Simulated through GT PRO Software)

Table 8 : Performance of GE9351FA at different inlet pressure drop at 15^{0}C

Inlet filter pressure loss	millibar	10	12.5	15	17.5	20
GT gross power	kW	256662	255855	255048	254240	253433
GT gross LHV eff	%	36.92	36.88	36.85	36.81	36.78
GT gross heat rate	kJ/kWh	9752	9761	9770	9779	9788
Compressor inlet mass flow	t/h	2331.4	2325.6	2319.8	2314	2308.2
Compressor inlet temperature	C	15	15	15	15	15
Turbine inlet temperature	C	1327.2	1327.2	1327.2	1327.3	1327.3
Turbine exhaust mass flow	t/h	2381.5	2375.5	2369.6	2363.7	2357.8
Turbine exhaust temperature	C	603.9	604.3	604.7	605.2	605.6
GT fuel flow	t/h	50.01	49.9	49.79	49.68	49.56

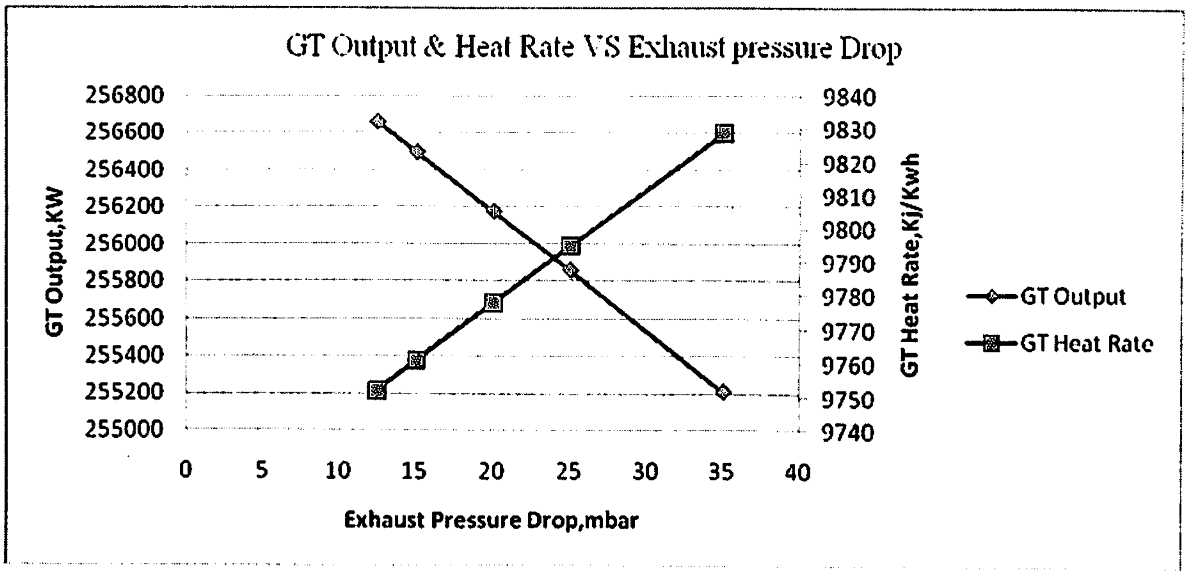


Graph 5 : Performance of GE9351FA at different inlet pressure drop at 15^{0}C

(Source: Simulated through GT PRO Software)

Table 9 : Performance of GE9351FA at different exhaust pressure drop at 15^{0}C

Total exhaust pressure loss	millibar	12.45	15	20	25	35
GT gross power	kW	256662	256499	256178	255857	255212
GT gross LHV eff	%	36.92	36.88	36.82	36.75	36.63
GT gross heat rate	kJ/kWh	9752	9761	9778	9795	9829
Compressor inlet mass flow	t/h	2331.4	2331.4	2331.4	2331.4	2331.4
Turbine inlet temperature	C	1327.2	1327.2	1327.3	1327.3	1327.4
Turbine exhaust mass flow	t/h	2381.5	2381.5	2381.5	2381.5	2381.6
Turbine exhaust temperature	C	603.9	604.3	605.1	606	607.6



Graph 6 : Performance of GE9351FA at different exhaust pressure drop at 15^0C

It can be seen that as the exhaust pressure drop increases the GT produces less power but without any impact in the inlet air flow

Chapter 3

Modeling of the CCPP

3.1 Overview of GT PRO & GT Master

Developed by Thermo flow Inc, GT PRO is a CCPP tool that is widely used to generate CCPP heat balance diagram and also estimate the physical components required to implement it based on optimal sizes and economical considerations. Thermo flow inc is one the leading developers of power plant HMBD software. To match with the up to date industrial practices, all thermo flow software are updated periodically. The current version of GT Pro being used is 21.

GT PRO can thoroughly auto generate the modeling of an open cycle and combined cycle power plants. Besides auto generation, GT PRO allows the user to enter all the pertinent data like ambient conditions, pressure, temperature & mass flow of steam turbines, selection of power augmenting devices like performance heater, evaporative coolers, fuel compressor etc in the gas turbines. The user can select the types of gas turbine, a wide-ranging list of gas turbines & one of the different condensing methods. It has a comprehensive list of 3200 variables to allow the user make a choice of convenience. Besides these facilities the user can define miscellaneous parameters like efficiency of the pumps, sizing of the pipes etc.

Once designed, the CCPP is expected to operate at various ambient conditions, grid conditions, and environmental factors that are generally called as off design conditions. GT Master is a tool that helps to evaluate the performance of the plant under these off design conditions. The user can simulate close to 2500 inputs that can define the performance of the plant at these off design conditions.

The basic difference between GT Pro and GT Master is that the GT Pro helps to create HMBD at standard- user defined conditions whereas GT Master helps to evaluate the performance of the modeled plant at off design conditions and also with differing loads.

3.2 Important Data pertaining to modeling of CCPP

Ambient temperature	:	30°C
Ambient Wet bulb temperature	:	23.8°C
Site Elevation	:	35.5 m
Atmospheric pressure	:	1.009 bar
Relative humidity	:	60 %

Fuel

Calorific vale of Fuel	:	Methane, 47242 KJ/Kg at 25 Deg C (LHV Basis)
Fuel Temperature after heating	:	185°C
Fuel Line Pressure	:	27.58 bar
HR of Fuel at combustor inlet	:	47609 KJ/Kg at 185 Deg C (LHV Basis)
Fuel heating Source	:	FW from IP Economizer

Gas Turbine

User Defined gas Turbine

Gross power (KW)	:	236800 KW
Heat Rate	:	9920 KJ/Kwh
GT Efficiency	:	36.29 %
Exhaust Flow	:	2256.6 T/hr
Turbine Exhaust-temperature	:	615.3°C
RPM	:	3000
Gas Turbine exhausts Pressure	:	1.04 bar
Inlet Pressure Drop	:	9.943 millibar
Exhaust Pressure Drop	:	31.11 millibar

Evaporative cooler effectiveness	:	85 Percent

HRSG with three pressure levels and re-heating

HP Steam	:	96.6 bar &565 Deg C, 271.8 t/h
IP Steam	:	23.26 bar &566 Deg C, 302.2 t/h
LP Steam	:	5 bars &295 Deg C, 31.8 t/h
Condenser Pressure	:	0.103 bar

Pinch Temperature

HP	:	7.1 Deg C
IP	:	15.4 Deg C
LP	:	15.1 Deg C

Approach Temperature

HP	:	4.5 Deg C
IP	:	11 Deg C
HRSG Draft Loss	:	30 millibar

Steam Turbine

ST Output	:	122257KW
HP Steam	:	96.62bar&565 Deg C, 260.3 t/h
IP Steam	:	22.84bar &565 Deg C, (259.3 CRH+43.55 IPSH) t/h
LP Steam	:	3.611 bar&295 Deg C, (302.8IP exit + 31.82 LPSH) t/h
Quality of steam at Turbine Exhaust	:	97% Dry
Sealing Steam Regulator Pressure	:	1.01 bar
Gland Steam Condenser Pressure	:	0.87 bar
Generator power factor	:	0.85

Cooling System Type

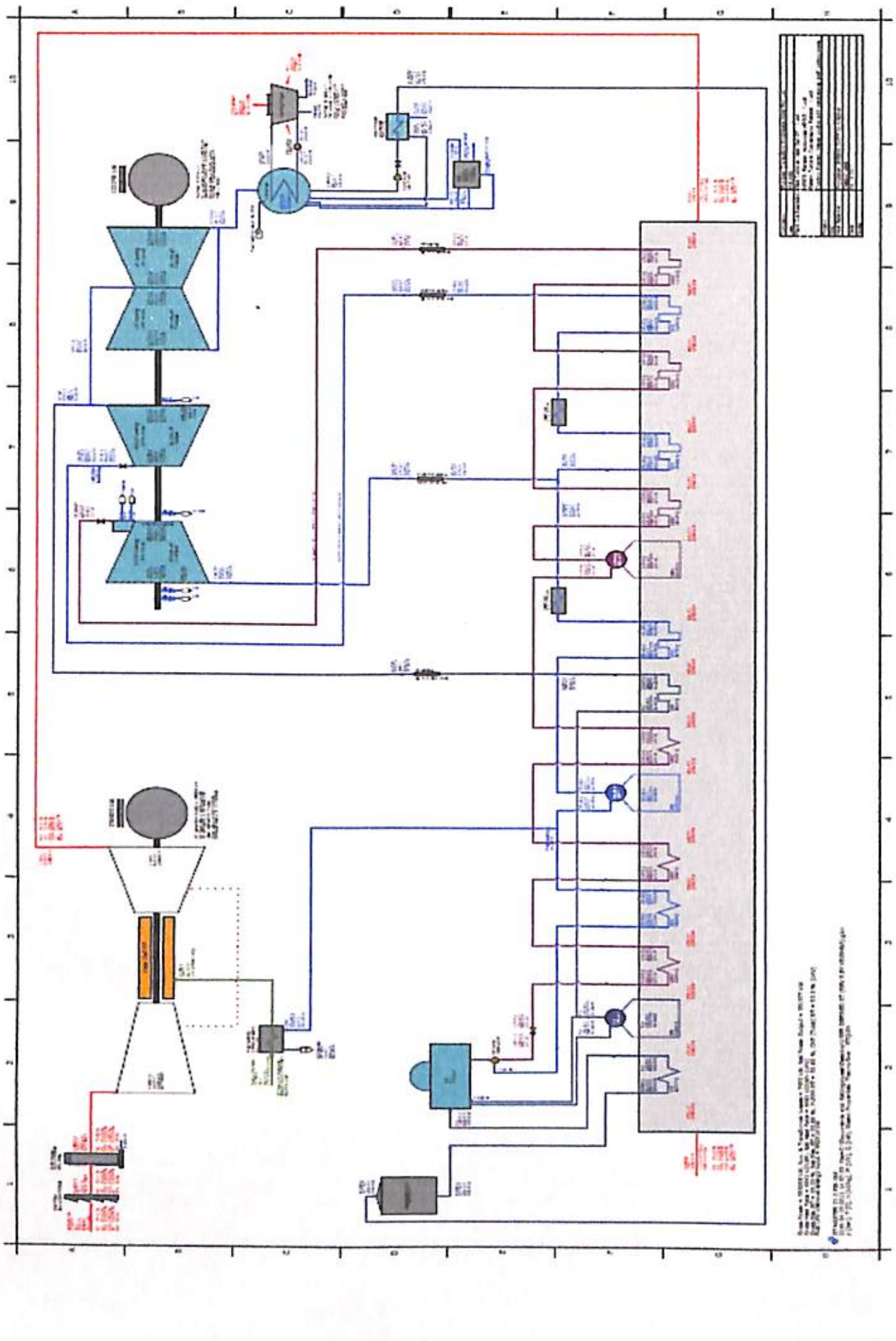
Water cooling with mechanical draft cooling tower

Water	:	Fresh Water
CW in Temperature	:	33^0C
CW Range	:	9^0C

3.3 HMBD Model

(Source: Simulated through GT PRO Software)

(Also Refer Appendix)



3.4 Text Output Summary

Refer Appendix 1

3.5 Graphics Output Summary

Refer Appendix 2

Chapter 4

Performance Testing

4.1 Overview

ASME Performance Test Codes (PTC) provides rules and procedures for planning, preparing, executing, and reporting performance tests. A performance test is an engineering evaluation; its results indicate how well the equipment performs its functions. Today, nearly 50 PTCs are available; they cover individual components (steam generators, turbines, pumps, compressors), systems (flue gas desulfurization, fuel cells), and complete plants (combined cycle and cogeneration plants). In addition to equipment codes, supplements on instruments and apparatus cover measurement systems (temperature, pressure, flow) and techniques (uncertainty analysis) common to several codes. For more than a century, ASME PTC tests have provided results with the highest level of accuracy, based on current engineering knowledge and practices, and taking into account the costs of the tests and the value of the information obtained. All ASME codes are developed using input from a range of parties, who may be interested in the code and/or in the associated equipment or process. Codes have the force of a legal document when cited in contracts, as they frequently are, for determining the method by which equipment performs as guaranteed. PTCs are used by equipment owners, equipment suppliers, and test engineers.

ASME PTCs protect users from poorly performing products and enable suppliers to compete fairly by offering reliable products. Purchase specifications are greatly strengthened by citing the results of PTC tests. When buying new equipment, purchasers may specify that the equipment guarantee will be based on the results of a specific ASME PTC test. Design engineers consult PTC documents to ensure that proper instrument connections will be available. Test engineers install the required instrumentation and use the code's procedures and calculation methods to conduct tests on the new equipment. Representatives of all parties to the test ensure that the test methods are in compliance with the code. Finally, the test results are compared to the performance criteria.

4.2 Some PTC Codes in Practice

PTC 4- 1998	:	Fired Steam Generators
PTC 4.4-1981	:	Gas Turbine Heat Recovery Steam Generators
PTC 6-2004	:	Steam Turbines
PTC 6.2 – 2004	:	Steam Turbines in Combined Cycles
PTC6-S	:	Procedures for Routine Performance Test of Steam Turbines
PTC19.1 – 2005	:	Measurement Uncertainty
PTC46 – 1996	:	Overall Plant Performance

4.3 Scope of ASME PTC 46

Performance testing codes are used to gauge the performance of a plant in its typical working state, with all components in a clean and fully-functional condition. This Code describes clear methods and standards for combined- cycle power plants. This code is not applicable to open cycle turbines where PTC 22 should be used. The scope of this Code begins only when a heat-recovery steam generator is included within the test boundary connected at the exhaust of gas based power generating unit. Following criteria should be satisfied to test a particular power plant

- ✓ A resource must be available to measure, all of the heat inputs entering the test boundary and all of the electrical power and secondary outputs leaving the test boundary;
- ✓ A resource must be available to measure all of the parameters to correct the results from the test to the base reference condition;
- ✓ The test result uncertainties are expected to be less than or equal to the standard uncertainties
- ✓ The working fluid for vapor cycles must be steam

Deviations from the methods recommended in this Code are acceptable only if it can be demonstrated they provide equal or lower uncertainty.

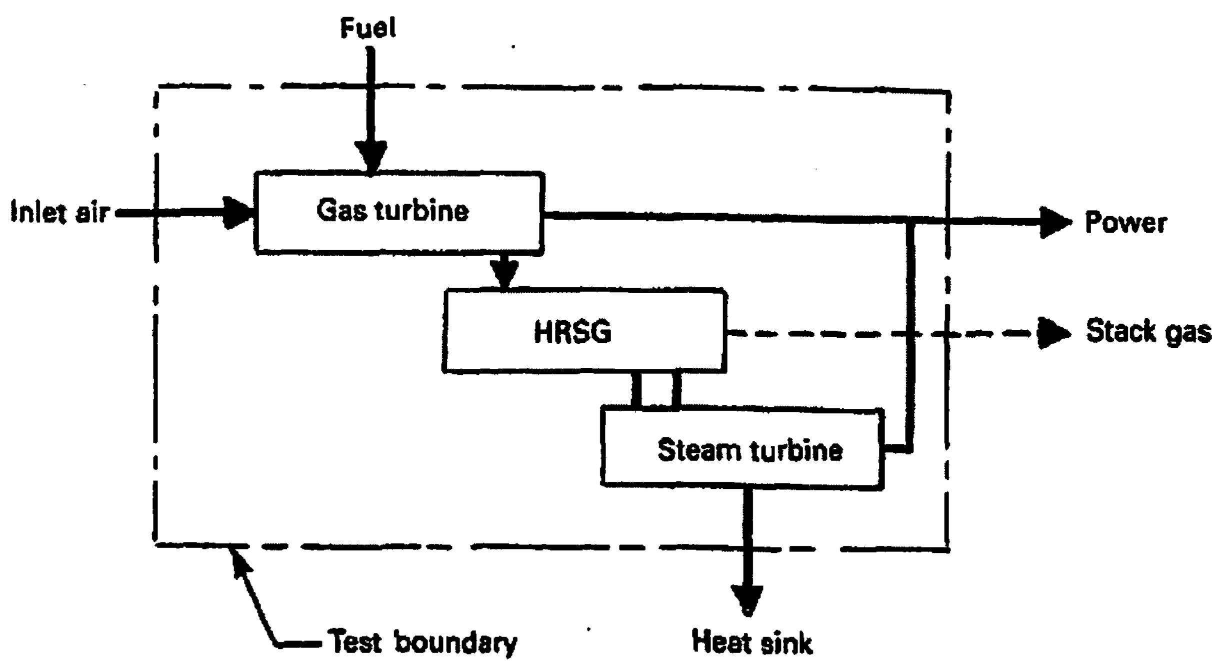
4.4 Expected Test Run Uncertainty

Type of Plant	:	Combined Cycle
Description	:	Combined gas turbine and steam turbine cycles with or without supplemental firing to a steam generator
Corrected Heat Rate	:	1.5%
Corrected Net Power	:	1.0%

4.5 Test Boundary and Required Measurements

The test boundary delineates the energy streams that must me determined to evaluate the corrected results. The test boundary accounts the streams that must be measured to determine the performance. All input and output energy streams required for test calculations must be determined with reference to the point at which they cross the boundary.

A typical combined cycle test boundary as defined in ASME PTC 46



4.6 Test Plan

A detailed test plan must be prepared prior to conducting a Code test. It will document agreements on all issues affecting the conduct of the test and provide detailed procedures for performing the test. The test plan should be approved, prior to the testing, by authorized signatures of all parties to the test. It must reflect any contract requirements that pertain to the test objectives and performance guarantees and provide any needed clarifications of contract issues. In addition to documenting all prior agreements, the test plan should include the schedule of test activities, responsibilities of the parties to the test, test procedures, and report of results.

Schedule of Test Activities

A test schedule should be prepared which should include the sequence of events and anticipated time of test, notification of the parties to the test, test plan preparations, test preparation and conduct, and preparation of the report of results.

4.7 Test Preparations

All parties to the test shall be given timely notification, as defined by prior agreement, to allow them the necessary time to respond and to prepare personnel, equipment, or documentation. Updated information should be provided as it becomes known. A test log must be maintained during the test to record any occurrences affecting the test, the time of the occurrence, and the observed resultant effect. This log will be part of the permanent record of the test. Personnel and instrumentation involved in the test should be considered. For example, provision of safe access to test point locations, availability of suitable utilities and safe work areas for personnel as well as potential damage to instrumentation or calibration shifting because of extreme ambient conditions such as temperature or vibration. Documentation must be developed or be made available for calculated or adjusted data to provide independent verification of algorithms, constants, scaling, calibration corrections, offsets, base points, and conversions.

4.8 Conduct of Test

This Subsection provides guidelines on the actual conduct of the performance test and addresses the following areas:

- ✓ Starting and stopping tests and test runs
- ✓ Methods of operation prior to and during tests
- ✓ Adjustments prior to and during tests
- ✓ Duration and number of tests and number of read constancy of test conditions

Starting and Stopping Tests and Test Runs

The test coordinator is responsible for ensuring that all data collection begins at the agreed-upon start of the test, and that all parties to the test are informed of the starting time.

Starting Criteria

Prior to starting each performance test, the following conditions must be satisfied:

Operation, configuration, and disposition for testing have been reached in accordance with the agreed upon test requirements, including:

- ✓ Equipment operation and method of control unit configuration, including required valve line-up, availability of consistent fuel and fuel supplements within the allowable limits of the fuel analysis for the test (by analysis as soon as practicable preceding the test) ,Plant operation within the bounds of the performance correction curves, algorithms or programs
- ✓ Equipment operation within allowable limits for a series of test runs, completion of internal adjustments required for repeatability

Stabilization

Prior to starting test, the plant must be operated for a sufficient period of time at test load to demonstrate and verify stability

Data Collection

Data acquisition system(s) functioning, and test personnel in place and ready to collect samples or record data.

Stopping Criteria

Tests are normally stopped when the test coordinator is satisfied that requirements for a complete test run have been satisfied. The test coordinator should verify that methods of operation during test. The test coordinator may extend or terminate the test if the requirements are not met. Data logging should be checked to ensure completeness and quality. After all test runs are completed, secure equipment operating for purposes of test only (such as vent steam). Return operation control to normal dispatch functions, if appropriate.

Methods of Operation Prior to and During Tests

All equipment necessary for normal and sustained operation at the test conditions must be operated during the test or accounted for in the corrections. Intermittent operation of equipment within the test boundary should be accounted for in a manner agreeable to all parties. Typical but non exhaustive examples of operating equipment for consideration include fuel handling equipment, soot blowers, ash handling systems, gas turbine compressor inlet chillers or evaporative coolers, gas compressors, water treatment and blow down. Any environmental control system must be operating and within normal ranges, including percent solids, gas flow, inlet and outlet emission concentrations, pH, and solid and liquid concentrations

Stabilization

Agreement must be reached on the necessary stable conditions before starting the test. The length of operating time necessary to achieve the required steady state will depend on previous operations

Plant Output

A test may be conducted at any load condition, as required to satisfy the goals of the test. For those tests which require a specified corrected or measured load, the test run electrical output should be set so that the estimated test result of net electrical power is within one percent of the applicable design value. For those tests which require a specified disposition of the plant, the test electrical output will be dependent on the performance of the plant itself and will not be controlled. At no time should the actual test conditions exceed any equipment ratings provided by the manufacturer

Plant Thermal Energy

Cogeneration plant thermal energy export shall be set at levels specified or as mutually agreed by parties to the test. If automatic control of export energy does not provide sufficient stability and proximity to design conditions, manual control or venting of export energy may be required

Duration of Runs, Number of Test Runs, and Number of Readings & Duration of Runs

The duration of a test run shall be of sufficient length that the data reflects the average efficiency and/or performance of the plant; this includes consideration for deviations in the measurable parameters due to controls, fuel, and typical plant operating characteristics.

The test coordinator and the parties to the test may determine that a longer test period is required. The recommended times are generally based upon continuous data acquisition. Depending upon the personnel available and the method of data acquisition, it may be necessary to increase the length of a test in order to obtain a sufficient number of samples of the measured parameters to attain the required test uncertainty.

Number of Test Runs

A run is a complete set of observations with the unit at stable operating conditions. A test is a single run or the average of a series of runs. While not requiring multiple runs, the advantages of multiple runs should be recognized. Conducting more than one run will: Provide a valid method of rejecting bad test runs

- ✓ Allow the parties to the test to examine the validity of the results
- ✓ Verify the repeatability of the results. Results may not be repeatable due to variations in either test methodology (test variations) or the actual performance of the equipment being tested (process variations)
- ✓ After completing the first test run that meets the criteria for an acceptable test run (which may be the preliminary test run), the data should be consolidated and preliminary results calculated and exam

4.9 Calculation and Reporting Of Results

The data taken during the test should be reviewed and rejected in part or in whole if not in compliance with the requirements for the constancy of test conditions. Each code test shall include pretest and post-test uncertainty analyses and the results of these analyses shall fall within code requirements for the type of plant being tested

Causes for Rejection of Readings

Upon completion of test or during the test itself, the test data shall be reviewed to determine if data from certain time periods should be rejected prior to the calculation of test results. A test log should be kept. Any plant upsets which cause test data to violate the requirements shall be rejected. A minimum of 10 minutes following the recovery of these criteria shall also be rejected to allow for re-stabilization. Should serious inconsistencies which affect the results be detected during a test run or during the calculation of the results, the run shall be invalidated completely, or it may be invalidated only in part if the affected part is at the beginning or at the end of the run. A run that has been invalidated shall be repeated, if necessary, to attain the test objectives. The decision to reject a run shall be the responsibility of the designated representatives of the parties to the test. During the test, should any control system set points be modified that effects stability of operation beyond code allowable limits. Test data shall be considered for rejection from the calculations of test results. The period rejected shall start immediately prior to the change and end no less than 10 minutes following the recovery of the criteria .An outlier analysis of spurious data should also be performed in accordance with PTC 19.1 on all primary measurements after the test has ended. This analysis will highlight any other time periods which should be rejected prior to calculating the test results.

4.10 Uncertainty

Test uncertainty and test tolerance are not interchangeable terms. This Code does not address test tolerance, which is a contractual term. Procedures relating to test uncertainty are based on concepts and methods described in PTC 19.1

Measurement Uncertainty

PTC 19.1 specifies procedures for evaluating measurement uncertainties from both random and fixed errors, and the effects of these errors on the uncertainty of a test result.

This Code addresses test uncertainty in the following four sections

- ✓ Section 1 defines expected test uncertainties.
- ✓ Section 3 defines the requirements for pretest and post-test uncertainty analyses, and how they are used in the test.
- ✓ Section 4 describes the bias uncertainty required for each test measurement.
- ✓ Section 5 and Appendix F provide applicable guidance for determining pretest and post-test uncertainty analysis results

4.11 Test Procedures

The test plan should include test procedures that provide details for the conduct of the test. The following are included in the test procedures:

The objective of test and the operating methodology

- ✓ Acceptance criteria for the completion of the test & the Base reference conditions for which the plant is designed
- ✓ Test boundaries should be clearly defined in order to measure the inputs and outputs and also the location of the measuring instruments.
- ✓ The intent of any contract or specification as to operating conditions, performance guarantees and environmental compliance.
- ✓ Complete pretest uncertainty analysis, with bias uncertainties established for each measurement
- ✓ Specific type, location, and calibration requirements for all instrumentation and measurement systems and frequency of data acquisition including the method of plant operation
- ✓ Measurement requirements for applicable emissions, including measurement location, instrumentation, and frequency and method of recording
- ✓ Identification of testing laboratories to be used for fuel, sorbent, and ash analyses
- ✓ Required operating disposition or accounting for all internal thermal energy and auxiliary power consumers having a material effect on test results
- ✓ Required standard of equipment cleanliness and inspection procedures
- ✓ Procedures to account for performance degradation
- ✓ Valve line-up requirements
- ✓ Preliminary testing requirements
- ✓ Required steadiness criteria and methods of maintaining operating conditions within these limits

- ✓ Allowable variations from base reference conditions and methods of setting and maintaining operating conditions within these limits
- ✓ Number of test runs and durations of each run test starts and stop requirements
- ✓ Data acceptance and rejection criteria
- ✓ Allowable range of fuel conditions, including correction curves or algorithms
- ✓ Specifying test run data reduction and calculation and correction of test results to base reference condition
- ✓ The method for combining test runs to calculate the final test results
- ✓ Test report format, contents, inclusions, and index constituents and heating value

Chapter 5

Correction Curves for 366 MW CCPP

5.1 Overview

Performance curves are the graphs that depict the operational characteristics of the machine with respect any of its performance affecting factors. A machine is bound to be affected by numerous factors and the impact of the effect of these individual factors can be visualized in the performance characteristics curve.

Correction curves on the other hand facilitate the evaluation of impact of these factors on the overall performance of the machine at site reference conditions. Site reference conditions are established specifically for a project site and used as a datum to evaluate different plants offered and also to establish the site design rating of the plant. A correction curve helps estimate the justness of performance of the machine under off-design conditions. A multiplicative or divisive factor called the correction factor will be utilized to estimate the correctness of the off design performance with respect to the designed performance. A CCPP performance is evaluated in terms of its heat rate and power output and hence correction curves are plotted for power output and heat rate vs. different factors that affect the output within the boundary of the CCPP.

The correction curves for a 366MW CCPP are plotted in the following chapter.

1) Correction Factor/curve for Power output / heat rate vs. Ambient Temperature
2) Correction Factor/curve for Power output / heat rate vs. Frequency
3) Correction Factor/curve for Power output / heat rate vs. Atmospheric pressure
4) Correction Factor/curve for Power output / heat rate vs. calorific value of fuel.

5.2 GT Master simulated Correction Factors and Correction Curves

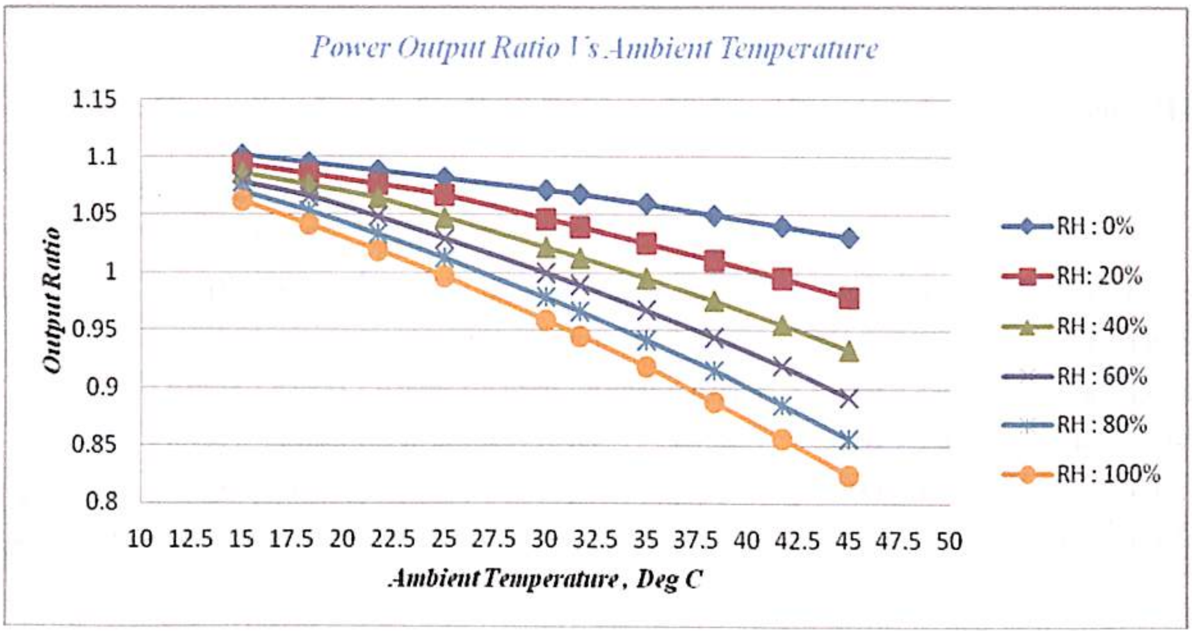
(Source: Simulated through GT MASTER Software)

Table 10 : Power output vs. Ambient Temperature at Various RH

	Ambient Temperature, Deg C									
	15	18.3	21.7	25	30	31.7	35	38.33	41.7	45
RH in %	395708	393244	390690	388206	384605	383351	380208	376783	373385	370177
	392709	389503	386283	382894	375655	373146	367985	362719	357318	351507
	389694	385977	381891	376004	366838	363747	357377	350390	343024	335298
	386856	382522	376162	369470	359058	355325	347458	339069	330379	320651
	384019	378026	370817	363467	351510	347228	338223	328914	318198	307514
	381081	373692	365826	357776	344425	339574	329909	318814	307472	295825

Table 11 : Correction Factor for Power output vs. Ambient Temperature at Various RH

	Ambient Temperature, Deg C									
	15	18.3	21.7	25	30	31.7	35	38.33	41.7	45
RH in %	1.102	1.0952	1.088	1.0811	1.0711	1.0676	1.0589	1.0493	1.0399	1.0309
	1.0937	1.0847	1.0758	1.0663	1.0462	1.0392	1.0248	1.0101	0.9951	0.9789
	1.0853	1.0749	1.0635	1.0471	1.0216	1.013	0.9953	0.9758	0.9553	0.9338
	1.0774	1.0653	1.0476	1.0289	1	0.9896	0.9676	0.9443	0.9201	0.893
	1.0695	1.0528	1.0327	1.0122	0.9789	0.967	0.9419	0.916	0.8862	0.8564
	1.0613	1.0407	1.0188	0.9964	0.9592	0.9457	0.9188	0.8879	0.8563	0.8238



Graph 7 : Correction curve for Power output vs. Ambient Temperature at Various RH

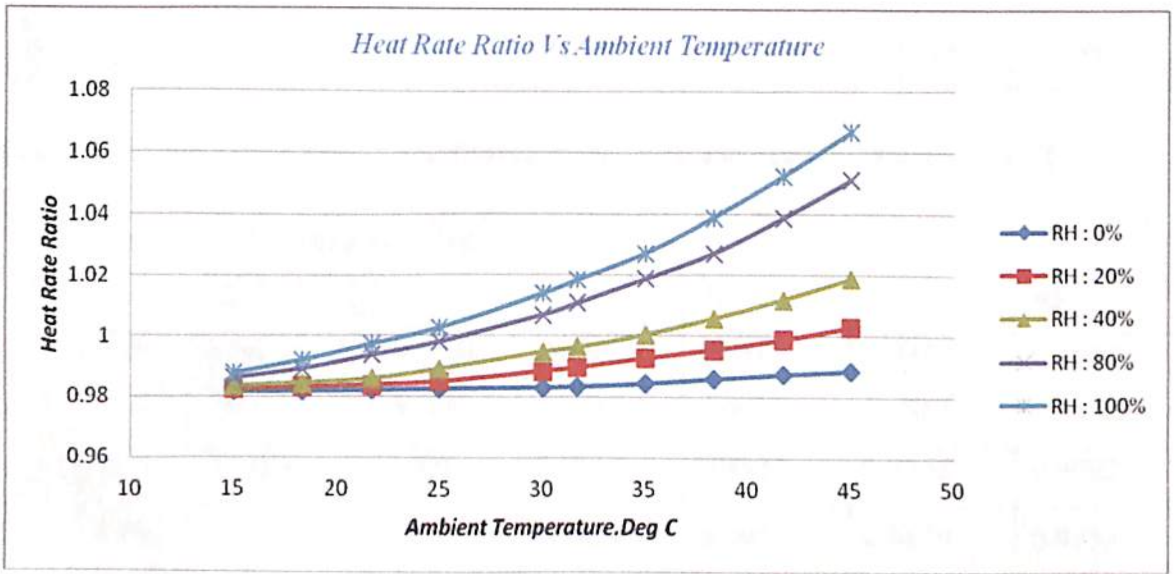
(Source: Simulated through GT MASTER Software)

Table 12 : Heat Rate vs. Ambient Temperature at Various RH

	Ambient Temp,Deg C										
		15	18.3	21.7	25	30	31.7	35	38.33	41.7	45
RH in %	0.01	6421	6422	6424	6427	6430	6432	6439	6449	6459	6467
	20	6426	6431	6435	6442	6465	6473	6493	6511	6532	6559
	40	6434	6440	6449	6470	6507	6517	6543	6578	6618	6663
	60	6441	6451	6474	6501	**6542**	6560	6602	6650	6703	6767
	80	6451	6470	6500	6528	6586	6611	6663	6719	6795	6876
	100	6461	6489	6524	6558	6631	6661	6719	6796	6884	6977

Table 13 : Correction Factor for Heat Rate vs. Ambient Temperature at Various RH

	Ambient Temp,Deg C										
		15	18.3	21.7	25	30	31.7	35	38.33	41.7	45
RH in %	0.01	0.9815	0.9816	0.9819	0.9824	0.9828	0.9831	0.9842	0.9857	0.9873	0.9885
	20	0.9822	0.9830	0.9836	0.9847	0.9882	0.9894	0.9925	0.9952	0.9984	1.0025
	40	0.9834	0.9844	0.9857	0.9889	0.9946	0.9961	1.0001	1.0055	1.0116	1.0184
	60	0.9845	0.9860	0.9896	0.9937	1	1.0027	1.0091	1.0165	1.0246	1.0343
	80	0.9860	0.9889	0.9935	0.9978	1.0067	1.0105	1.0184	1.0270	1.0386	1.0510
	100	0.9876	0.9918	0.9972	1.0024	1.0136	1.01819	1.0270	1.0388	1.0522	1.0664



Graph 8 : Correction curve for Heat Rate vs. Ambient Temperature at Various RH

(Source: Simulated through GT MASTER Software)

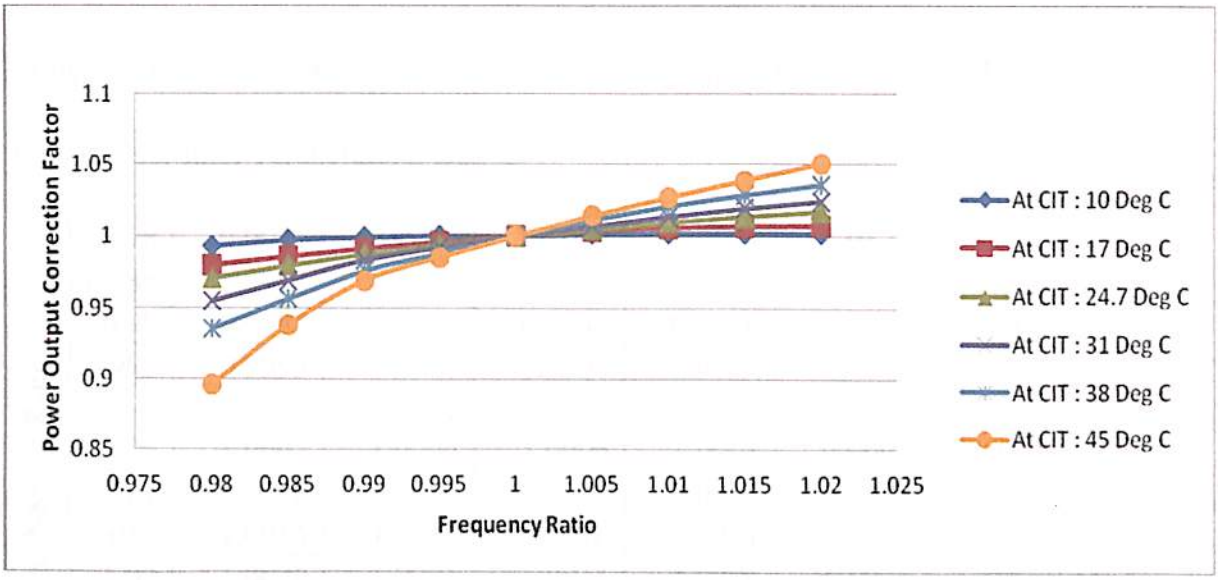
Table 14 : Power Output vs. Frequency at various CIT

Shaft-Speed Ratio	Compressor Inlet Temperature, Deg C					
	10	17	24.7	31	38	45
0.98	363005	355589	348701	339029	327546	309388
0.985	364552	357749	351724	344192	334880	323794
0.99	365223	359682	354500	349099	341700	334543
0.995	365650	361426	356806	352277	345919	340102
1	365595	362993	359015	355080	350156	345162
1.005	366192	364297	360863	357589	354084	349976
1.01	366340	365067	362539	359930	357473	354444
1.015	366394	365411	363975	362000	360213	358524
1.02	366320	365584	365247	363767	362714	362685

Table 15 : Correction Factor for Power Output vs. Frequency at various CIT

Shaft-Speed Ratio	Compressor Inlet Temperature, Deg C					
	10	17	24.7	31	38	45
0.98	0.9929	0.97960	0.9712	0.9547	0.9354	0.8963
0.985	0.9971	0.98555	0.9796	0.9693	0.9563	0.9380
0.99	0.9989	0.9908	0.9874	0.9831	0.9758	0.9692
0.995	1.0001	0.9956	0.9938	0.9921	0.9879	0.9853
1	1	1	1	1	1	1
1.005	1.0016	1.0035	1.0051	1.0070	1.0112	1.0139
1.01	1.0020	1.0057	1.0098	1.0136	1.0208	1.0268
1.015	1.0021	1.0066	1.0138	1.0194	1.0287	1.0387
1.02	1.0019	1.0071	1.0173	1.0244	1.0358	1.0507

(Source: Simulated through GT MASTER Software)



Graph 9 : Correction Curve for Power Output vs. Frequency at various CIT

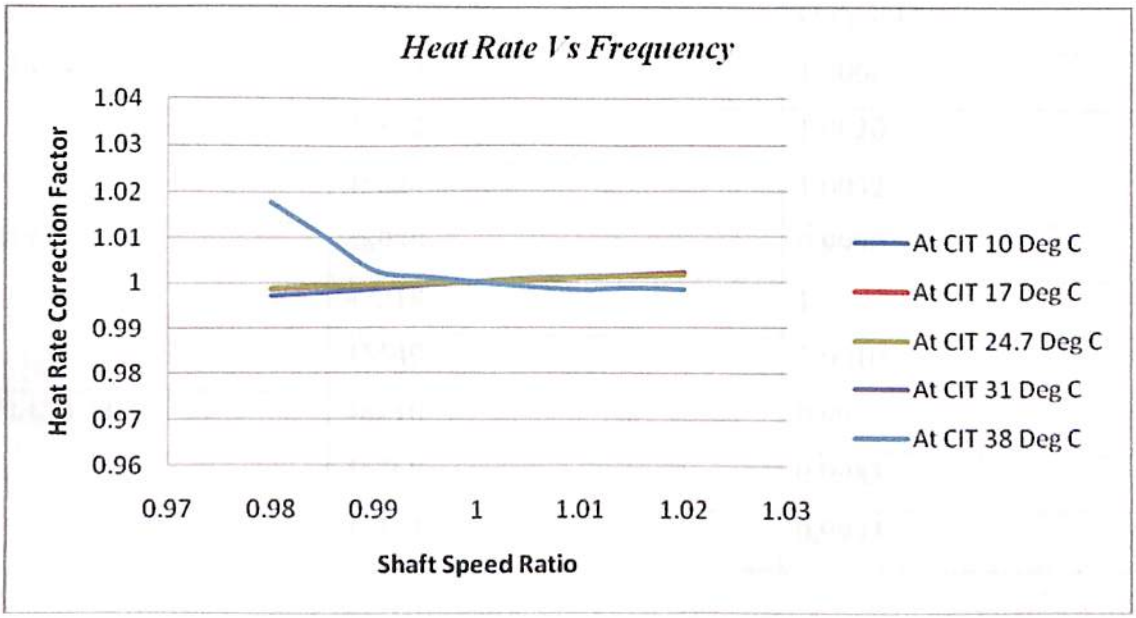
(Source: Simulated through GT MASTER Software)

Table 16 : Heat Rate vs. Frequency at Various RH

Compressor Inlet Temperature, Deg C							
		10	17	24.7	31	38	45
Shaft-Speed Ratio	0.98	6400	6463	6536	6665	6829	6988
	0.985	6404	6466	6539	6638	6780	6888
	0.99	6409	6468	6541	6614	6727	6822
	0.995	6414	6470	6542	6614	6717	6820
	1	6418	6471	6543	6616	6709	6806
	1.005	6423	6474	6547	6617	6703	6792
	1.01	6425	6478	6549	6617	6699	6780
	1.015	6427	6481	6550	6617	6701	6774
	1.02	6431	6485	6552	6620	6699	6764

Table 17 : Correction Factor for Heat Rate vs. Frequency at Various RH

Compressor Inlet Temperature, Deg C							
Shaft-Speed Ratio		10	17	24.7	31	38	45
	0.98	0.9971	0.9987	0.9989	1.0074	1.0178	1.0267
	0.985	0.9978	0.9992	0.9993	1.0033	1.0105	1.0120
	0.99	0.9985	0.9995	0.9996	0.9996	1.0026	1.0023
	0.995	0.9993	0.9998	0.9998	0.9996	1.0011	1.0020
	1	1	1	1	1	1	1
	1.005	1.0007	1.0004	1.0006	1.0001	0.9991	0.9979
	1.01	1.0010	1.0010	1.0009	1.0001	0.9985	0.9961
	1.015	1.0014	1.0015	1.0010	1.0001	0.9988	0.9952
	1.02	1.0020	1.0021	1.0013	1.0006	0.9985	0.9938

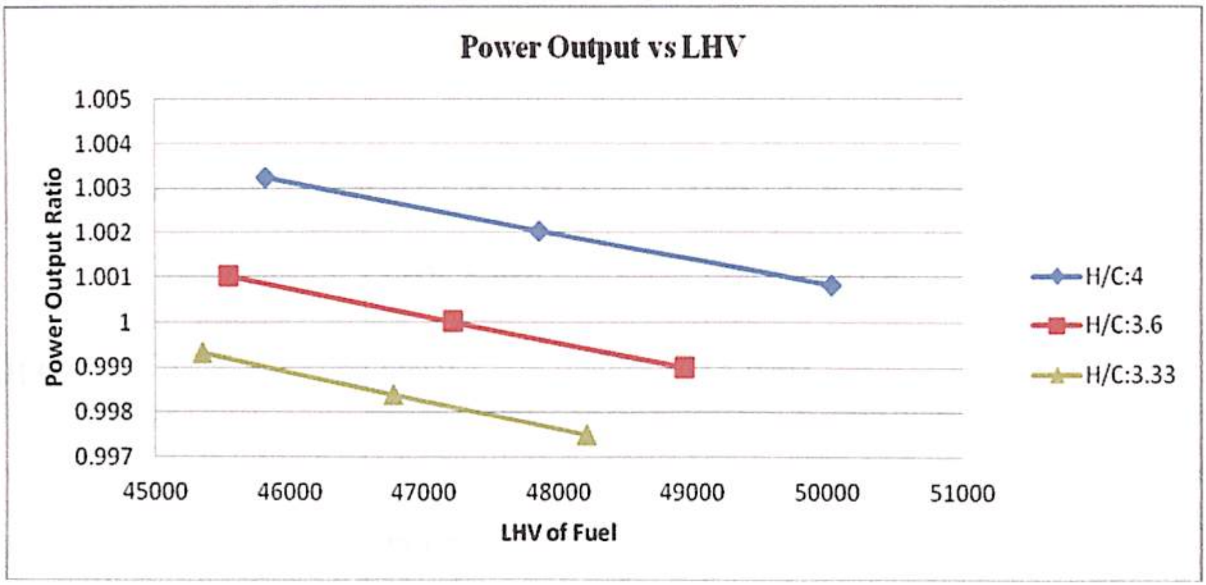


Graph 10 : Correction curve for Heat Rate vs. Frequency at Various RH

(Source: Simulated through GT MASTER Software)

Table 18 : Correction Factor Power Output vs. calorific Value of Fuel (LHV Basis)

	LHV, KJ/kg	Power Output
H/C:4	50035	359354
	47859	359789
	45824	360216
H/C:3.6	48938	358704
	47218	359058
	45549	359429
H/C:3.33	48210	358162
	46768	358482
	45354	358815
	LHV, KJ/kg	Output CF
H/C:4	50035	1.0008
	47859	1.0020
	45824	1.0032
H/C:3.6	48938	0.9990
	47218	1
	45549	1.0010
H/C:3.33	48210	0.9975
	46768	0.9983
	45354	0.9993



Graph 11 : Correction curve for Power Output vs. calorific Value of Fuel (LHV Basis)

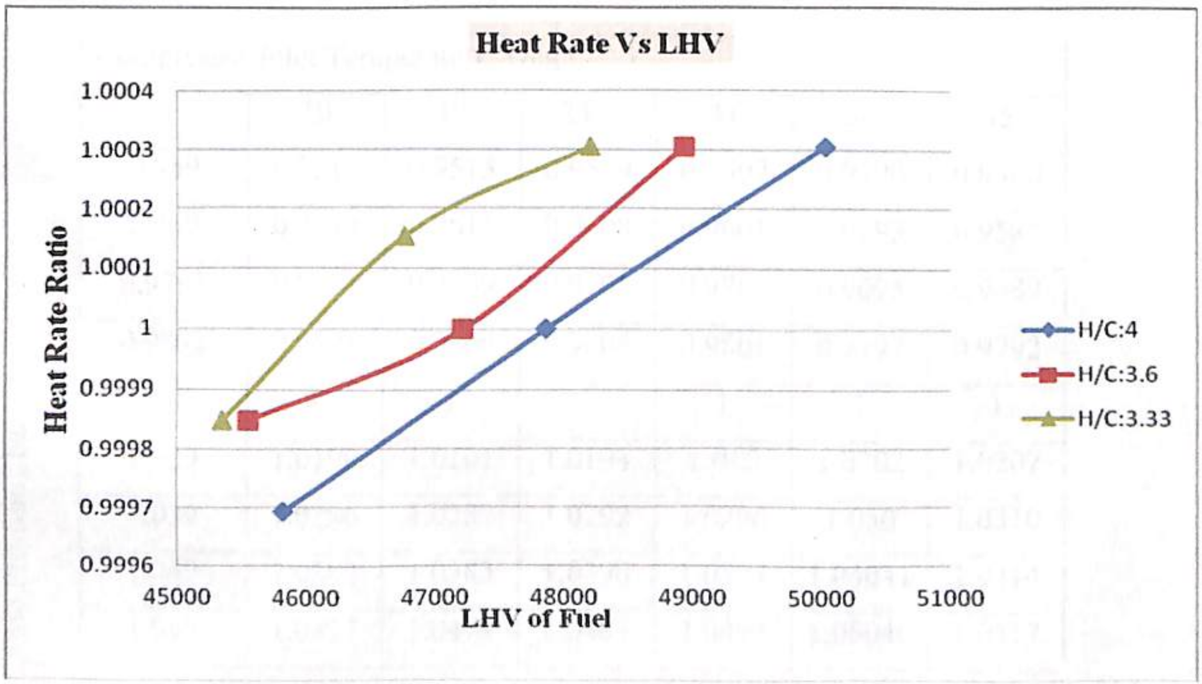
(Source: Simulated through GT MASTER Software)

Table 19 : Heat Rate vs. calorific Value of Fuel (LHV Basis)

	LHV, KJ/kg	Heat Rate ,KJ/Kwh
H/C:4	50035	6544
	47859	6542
	45824	6540
H/C:3.6	48938	6544
	47218	6542
	45549	6541
H/C:3.33	48210	6544
	46768	6543
	45354	6541

Table 20 : Correction Factor for Heat Rate vs. calorific Value of Fuel (LHV Basis)

	LHV, KJ/kg	Heat Rate CF
H/C:4	50035	1.0003
	47859	1
	45824	0.9996
H/C:3.6	48938	1.0003
	47218	1
	45549	0.9998
H/C:3.33	48210	1.0003
	46768	1.0001
	45354	0.9998



Graph 12 : Correction curve for Heat Rate vs. calorific Value of Fuel (LHV Basis)

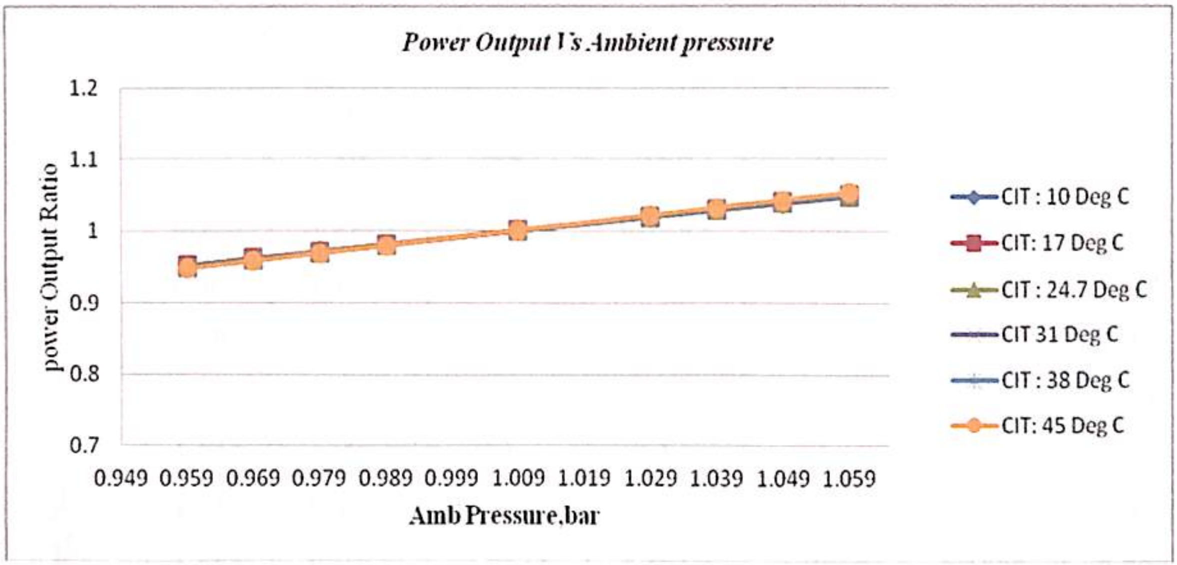
(Source: Simulated through GT MASTER Software)

Table 21 : Power Output vs. Atmospheric Pressure at various CIT

	Compressor Inlet Temperature,Deg C						
Amb Pressure, bar		10	17	24.7	31	38	45
	0.959	355335	349284	341354	333675	324323	314617
	0.969	358962	352900	344901	337172	327817	318089
	0.9791	362582	356476	348407	340692	331298	321550
	0.9892	366190	360044	351944	344170	334793	324977
	1.009	373357	367146	358967	351152	341717	331850
	1.029	380474	374177	365963	358075	348630	338726
	1.039	384049	377696	369473	361565	352078	342159
	1.049	387621	381211	372976	365005	355528	345593
	1.059	391184	384731	376477	368475	358963	349011

Table 22 : Correction factor for power Output vs. Atmospheric Pressure at various CIT

	Compressor Inlet Temperature, Deg C						
Amb Pressure , bar		10	17	24.7	31	38	45
	0.959	0.9517	0.9513	0.9509	0.9502	0.9490	0.9480
	0.969	0.9614	0.9611	0.9608	0.9601	0.9593	0.9585
	0.9791	0.9711	0.9709	0.9705	0.9702	0.9695	0.9689
	0.9892	0.9808	0.9806	0.9804	0.9801	0.9797	0.9792
	1.009	1	1	1	1	1	1
	1.029	1.0190	1.0191	1.0194	1.019	1.0202	1.0207
	1.039	1.0286	1.0287	1.0292	1.0296	1.030	1.0310
	1.049	1.0382	1.0383	1.0390	1.0394	1.04041	1.0414
	1.059	1.0477	1.0478	1.0487	1.0493	1.05046	1.0517



Graph 13 : Correction Curve for power Output vs. Atmospheric Pressure at various CIT

(Source: Simulated through GT MASTER Software)

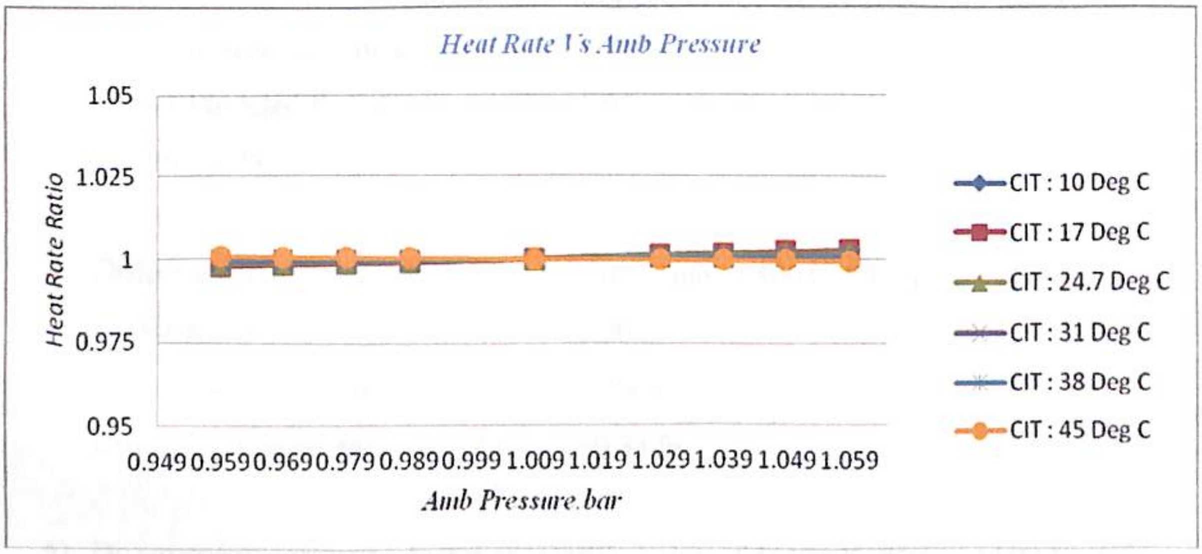
Table 23 : Heat Rate vs. Atmospheric Pressure at various CIT

	Compressor Inlet Temperature, Deg C						
Amb Pressure, bar		10	17	24.7	31	38	45
	0.959	6277	6385	6534	6683	6874	7083
	0.969	6280	6387	6535	6684	6874	7081
	0.979	6282	6390	6538	6685	6874	7080
	0.989	6285	6392	6539	6687	6874	7080
	1.009	6292	6398	6544	6690	6874	7079
	1.029	6299	6405	6549	6693	6876	7078
	1.039	6302	6408	6551	6695	6876	7077
	1.049	6305	6412	6553	6697	6877	7077
	1.059	6308	6415	6555	6698	6877	7076

Table 24 : Correction factor for Heat Rate vs. Atmospheric Pressure at various CIT

	Compressor Inlet Temperature, Deg C						
Amb Pressure, bar		10	17	24.7	31	38	45
	0.959	0.9976	0.9979	0.9984	0.9989	1	1.0005
	0.969	0.9980	0.9982	0.9986	0.9991	1	1.0002
	0.9791	0.9984	0.9987	0.9990	0.9992	1	1.0001
	0.9892	0.9988	0.9990	0.9992	0.9995	1	1.0001
	1.009	1	1	1	1	1	1
	1.029	1.0011	1.0010	1.0007	1.0004	1.0002	0.9998
	1.039	1.0015	1.0015	1.0010	1.0007	1.0002	0.9997
	1.049	1.0020	1.0021	1.0013	1.0010	1.0004	0.9997
	1.059	1.0025	1.0026	1.0016	1.0011	1.0004	0.9995

(Source: Simulated through GT MASTER Software)



Graph 14 : Correction Curve for heat rate vs. Atmospheric Pressure at various CIT

5.3 Comparison of correction factors of Modeled output and actual output

Difference in correction factor at 15 Deg C ,0.01%RH between model & Reference

CF for Reference Project	:	1.1007
CF for Modeled Project	:	1.1020
Difference Percentage	:	-0.125 %

1) Difference in correction factor at 45 Deg C ,80% RH between model & Reference

CF for Reference Project	:	0.8553
CF for Modeled Project	:	0.8238
Difference Percentage	:	3.63%

2) Difference in correction factor at shaft Speed of 2940RPM & CIT 17 Deg C

CF for Reference Project	:	0.9904
CF for Modeled Project	:	0.9929
Difference Percentage	:	-0.25 %

3) Difference in correction factor at shaft Speed of 3060RPM & CIT 45 Deg C

CF for Reference Project	:	1.0476
CF for Modeled Project	:	1.0507
Difference Percentage	:	-0.30 %

4) Difference in correction factor at calorific value of 50035 KJ/Kg on LHV Basis

CF for Reference Project	:	1.0042
CF for Modeled Project	:	1.0008
Difference Percentage	:	+0.33 %

5) Difference in correction factor at calorific value of 45354 KJ/Kg on LHV Basis

CF for Reference Project	:	1.0013
CF for Modeled Project	:	0.9993
Difference Percentage	:	+0.19 %

6) Difference in correction factor at ambient Pressure of 0.959 bar & CIT of 10 Deg C

   CF for Reference Project : 0.9509

   CF for Modeled Project : 0.9517

   Difference Percentage : -0.08 %

7) Difference in correction factor at ambient Pressure of 1.059 bar & CIT of 45 Deg C

   CF for Reference Project : 1.0464

   CF for Modeled Project : 1.0517

   Difference Percentage : -0.5 %

5.4 Comparison of correction factors of Modeled Heat Rate and actual Heat Rate

1) Difference in correction factor at ambient temperature 15 Deg C ,0.01%RH between model & Reference

   CF for Reference Project : 0.9810

   CF for Modeled Project : 0.9815

   Difference Percentage : -0.04%

2) Difference in correction factor at ambient temperature of 45 Deg C ,80% RH between model & Reference

   CF for Reference Project : 1.0523

   CF for Modeled Project : 1.0664

   Difference Percentage : -1.34%

3) Difference in correction factor at shaft Speed of 2940RPM& CIT 10 Deg C

   CF for Reference Project : 0.9986

   CF for Modeled Project : 0.9971

   Difference Percentage : 0.15%

4) Difference in correction factor at shaft Speed of 3060RPM & CIT 45 Deg C

CF for Reference Project	:	0.9968
CF for Modeled Project	:	0.9938
Difference Percentage	:	0.30%

5) Difference in correction factor at calorific value of 50035 KJ/Kg on LHV Basis

CF for Reference Project	:	0.9967
CF for Modeled Project	:	1.0003
Difference Percentage	:	-0.35%

6) Difference in correction factor at calorific value of 45354 KJ/Kg on LHV Basis

CF for Reference Project	:	1.0013
CF for Modeled Project	:	0.9998
Difference Percentage	:	0.15%

7) Difference in correction factor at ambient Pressure of 0.959 bar & CIT of 10 Deg C

CF for Reference Project	:	0.9984
CF for Modeled Project	:	0.9976
Difference Percentage	:	0.08%

8) Difference in correction factor at ambient Pressure of 1.059 bar & CIT of 45 Deg C

CF for Reference Project	:	1.0046
CF for Modeled Project	:	0.9995
Difference Percentage	:	0.50%

Chapter 6

Conclusion

A 366MW combined cycle power plant is successfully modeled and its performance correction curves at various test conditions are plotted. The results obtained are satisfactory when compared with an existing reference project. The results are to be compared with the actual test results of a new project that is to be commissioned.

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Further Reading

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Gas Turbine Engineering Handbook 2nd Edition: Meherwan P Boyce

ASME Performance Testing Codes 46

Power Plant Engineering by Black & Veatch

Steam Its Generation and Use by Babcock & Wilcox

Appendix

Appendix 1	:	Text Output of 366MW CCPP
Appendix 2	:	Graphics Output of 366MW CCPP
Appendix 3	:	Manual Calculations of CCPP

System Summary Report

GT MASTER 21.0 FIB-069
2146 04-15-2011 14:57:53 file=C:\Documents and Settings\user\Desktop\USER DEFINED GT (REV 5,BY KRISHNA).gtm
Simulated 366MW CCPP by Krishna
Design case
Plant Configuration: GT, HRSG, and condensing reheat ST
Steam Property Formulation: Thermoflow - STQUIK

SYSTEM SUMMARY

	Power Output kW		LHV Heat Rate kJ/kWh		Elect. Eff. LHV%	
	@ gen. term.	net	@ gen. term.	net	@ gen. term.	net
Gas Turbine(s)	236800		9920		36.29	
Steam Turbine(s)	122258					
Plant Total	359058	351577	6542	6681	55.03	53.88

PLANT EFFICIENCIES

PURPA efficiency %	CHP (Total) efficiency %	Power gen. eff. on chargeable energy, %	Canadian Class 43 Heat Rate, kJ/kWh
53.89	53.90	53.89	7215

GT fuel HHV/LHV ratio =	1.103	
DB fuel HHV/LHV ratio =	1.103	
Total plant fuel HHV heat input / LHV heat input =	1.103	
Fuel HHV chemical energy input (77F/25C) =	719947	kW
Fuel LHV chemical energy input (77F/25C) =	652516	kW
Total energy input (chemical LHV + ext. addn.) =	652516	kW
Energy chargeable to power (93.0% LHV alt. boiler) =	652360	kW

GAS TURBINE PERFORMANCE - User Def GT

	Gross power output, kW	Gross LHV efficiency, %	Gross LHV Heat Rate kJ/kWh	Exh. flow t/h	Exh. temp. C
per unit	236800	36.29	9920	2257	615
Total	236800			2257	

Number of gas turbine unit(s) =	1	
Gas turbine load [%] =	100	%
Fuel chemical HHV (77F/25C) per gas turbine =	719947	kW
Fuel chemical LHV (77F/25C) per gas turbine =	652516	kW

STEAM CYCLE PERFORMANCE

HRSG eff. %	Gross power output kW	Internal gross elect. eff., %	Overall elect. eff., %	Net process heat output kW
86.65	122258	34.37	29.78	145

Number of steam turbine unit(s) =	1	
Fuel chemical HHV (77F/25C) to duct burners =	0	kW
Fuel chemical LHV (77F/25C) to duct burners =	0	kW
DB fuel chemical LHV + HRSG inlet sens. heat =	410539	kW
Net process heat output as % of total output =	0.0412	%

System Summary Report

ESTIMATED PLANT AUXILIARIES (kW)		
GT fuel compressor(s)*	0	kW
GT supercharging fan(s)*	0	kW
GT electric chiller(s)*	0	kW
GT chiller/heater water pump(s)	0	kW
HRSG feedpump(s)*	1688.9	kW
Condensate pump(s)*	240.5	kW
HRSG forced circulation pump(s)	0	kW
LTE recirculation pump(s)	0	kW
Cooling water pump(s)	1490	kW
Air cooled condenser fans	0	kW
Cooling tower fans	1138.2	kW
HVAC	65	kW
Lights	130	kW
Aux. from PEACE running motor/load list (PEACE = 924.8 kW X Multiplier = 0)	0	kW
Miscellaneous gas turbine auxiliaries	473.6	kW
Miscellaneous steam cycle auxiliaries	279.2	kW
Miscellaneous plant auxiliaries	179.5	kW
Constant plant auxiliary load	0	kW
Gasification plant, ASU*	0	kW
Gasification plant, fuel preparation	0	kW
Gasification plant, AGR*	0	kW
Gasification plant, other/misc	0	kW
Desalination plant auxiliaries	0	kW
Program estimated overall plant auxiliaries	5685	kW
Actual (user input) overall plant auxiliaries	5685	kW
Transformer losses	1795.3	kW
Total auxiliaries & transformer losses	**7480**	**kW**
* Heat balance related auxiliaries		

System Summary Report

PLANT HEAT BALANCE				
Energy In	**765426**	**kW**		
Ambient air sensible	18744	kW		
Ambient air latent	24234	kW		
Fuel enthalpy @ supply	722337	kW		
External gas addition to combustor	0	kW		
Steam and water	110.8	kW		
Makeup and process return	0	kW		
Energy Out	**765576**	**kW**		
Net power output	351577	kW		
Stack gas sensible	72574	kW		
Stack gas latent	96702	kW		
GT mechanical loss	1449.7	kW		
GT gear box loss	0	kW		
GT generator loss	3383	kW		
GT miscellaneous losses	1972.8	kW		
GT ancillary heat rejected	0	kW		
GT process air bleed	0	kW		
Fuel compressor mech/elec loss	0	kW		
Supercharging fan mech/elec loss	0	kW		
Condenser	223946	kW		
Process steam	145	kW		
Process water	0	kW		
Blowdown	0	kW		
Heat radiated from steam cycle	5255	kW		
ST/generator mech/elec/gear loss	3020	kW		
Non-heat balance related auxiliaries	3756	kW		
Transformer loss	1795.3	kW		
Energy In - Energy Out	**-149.6**	**kW**	**-0.0195**	**%**
Zero enthalpy: dry gases & liquid water @ 32 F (273.15 K)				
Gas Turbine and Steam Cycle: Energy In - Energy Out = -149.6 kW				

System Summary Table

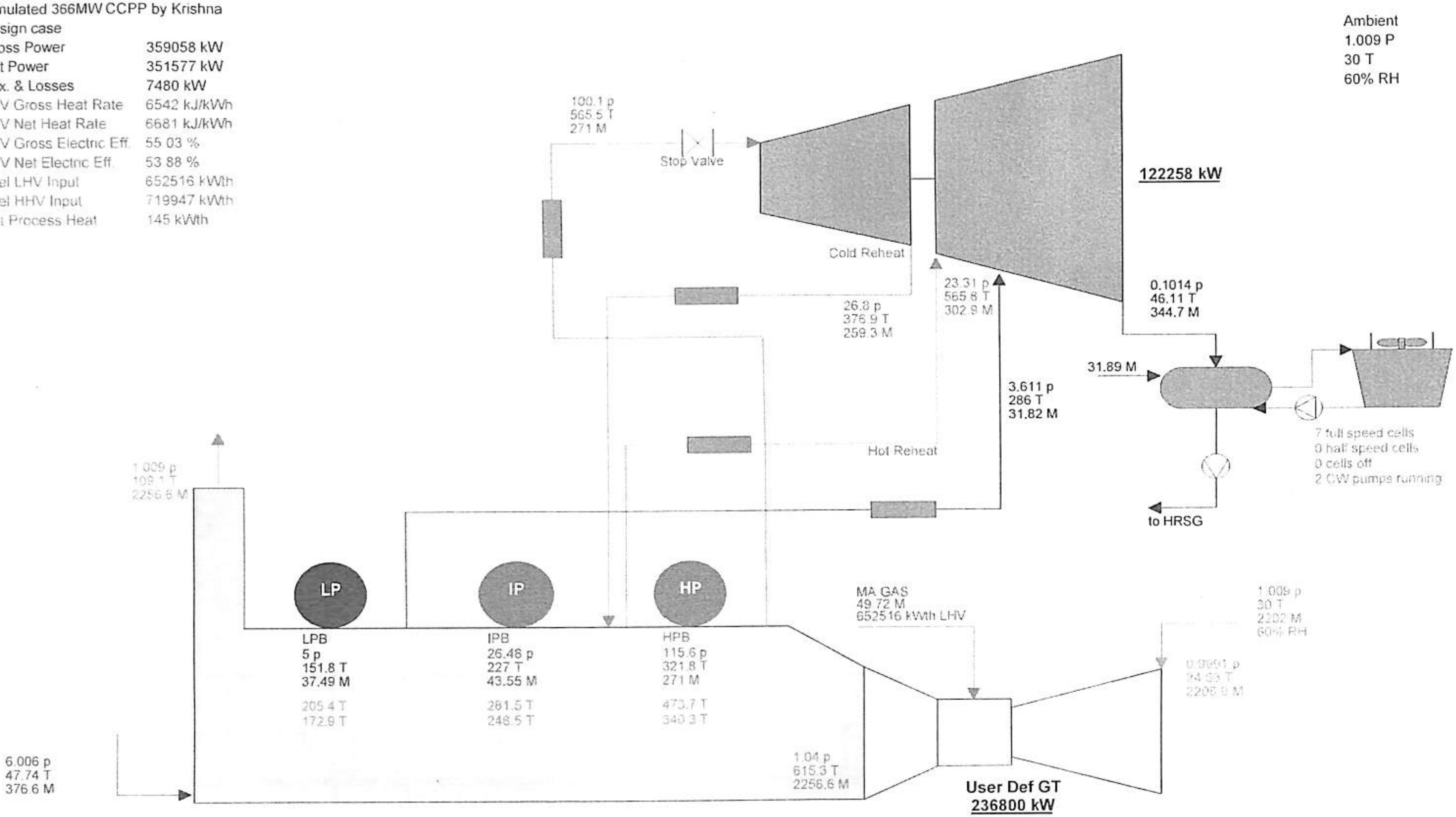
Plant Summary		
1. System Summary		
Plant total power output @ generator terminal	359058	kW
Total auxiliaries & transformer losses	7480	kW
Plant net power output	351577	kW
Plant LHV heat rate @ generator terminal	6542	kJ/kWh
Plant HHV heat rate @ generator terminal	7218	kJ/kWh
Plant net LHV heat rate	6681	kJ/kWh
Plant net HHV heat rate	7372	kJ/kWh
Plant LHV electric eff. @ generator terminal	55.03	%
Plant HHV electric eff. @ generator terminal	49.87	%
Plant net LHV electric efficiency	53.88	%
Plant net HHV electric efficiency	48.83	%
2. Plant Efficiencies		
PURPA efficiency, LHV	53.89	%
PURPA efficiency, HHV	48.84	%
CHP (Total) efficiency, LHV	53.9	%
CHP (Total) efficiency, HHV	48.85	%
Power generation eff. on chargeable energy, LHV	53.89	%
Power generation eff. on chargeable energy, HHV	48.85	%
Canadian Class 43 heat rate	7215	kJ/kWh
Plant fuel LHV chemical energy input (77F/25C)	652516	kW
Plant fuel HHV chemical energy input (77F/25C)	719947	kW
Total energy input (chemical LHV + ext. addn.)	652516	kW
Energy chargeable to power, LHV	652360	kW
Energy chargeable to power, HHV	719775	kW
GT fuel chemical HHV/LHV ratio	1.103	
DB fuel chemical HHV/LHV ratio	1.103	
Plant fuel HHV heat input /LHV heat input	1.103	
3. Gas Turbine Performance (per unit)	**User Def GT**	**1 unit(s)**
Gross power output	236800	kW
Gross LHV efficiency	36.29	%
Gross HHV efficiency	32.89	%
Gross LHV heat rate	9920	kJ/kWh
Gross HHV heat rate	10945	kJ/kWh
Exhaust mass flow	2256.6	t/h
Exhaust temperature	615.3	C
Fuel chemical LHV input (77F/25C)	652516	kW
Fuel chemical HHV input (77F/25C)	719947	kW
4. Steam Cycle Performance (LHV)		
HRSG efficiency	86.65	%
Steam turbine gross power	122258	kW
Internal gross efficiency	34.37	%
Overall efficiency	29.78	%
Net process heat output	145	kW
Fuel chemical LHV (77F/25C) to duct burners	0	kW
Fuel chemical HHV (77F/25C) to duct burners	0	kW
DB fuel chemical LHV + HRSG inlet sens. heat	410539	kW
Net process heat output / total output	0.0412	%
5. Plant Auxiliaries		
GT fuel compressor(s)	0	kW
GT supercharging fan(s)	0	kW
GT electric chiller(s)	0	kW
GT chiller/heater water pump(s)	0	kW

System Summary Table

Plant Summary		
HRSG feedpump(s)	1688.9	kW
Condensate pump(s)	240.5	kW
HRSG forced circulation pump(s)	0	kW
LTE recirculation pump(s)	0	kW
Cooling water pump(s)	1490	kW
Air cooled condenser fans	0	kW
Cooling tower fans	1138.2	kW
HVAC	65	kW
Lights	130	kW
Aux. from PEACE running motor/load list	0	kW
Miscellaneous gas turbine auxiliaries	473.6	kW
Miscellaneous steam cycle auxiliaries	279.2	kW
Miscellaneous plant auxiliaries	179.5	kW
Constant plant auxiliary load	0	kW
Gasification plant, ASU	0	kW
Power to AGR	0	kW
Gasification plant, air boost compressor	0	kW
Gasification plant, fuel preparation	0	kW
Gasification plant, syngas recirculation compressor	0	kW
Gasification plant, Other/misc	0	kW
Desalination plant auxiliaries	0	kW
Program estimated overall plant auxiliaries	5685	kW
Actual (user input) overall plant auxiliaries	5685	kW
Transformer losses	1795.3	kW
Total auxiliaries & transformer losses	7480	kW

GT MASTER 21.0 FIB-069
Simulated 366MW CCPP by Krishna
Design case

Gross Power	359058 kW
Net Power	351577 kW
Aux. & Losses	7480 kW
LHV Gross Heat Rate	6542 kJ/kWh
LHV Net Heat Rate	6681 kJ/kWh
LHV Gross Electric Eff.	55.03 %
LHV Net Electric Eff.	53.88 %
Fuel LHV Input	652516 kWth
Fuel HHV Input	719947 kWth
Net Process Heat	145 kWth



p [bar] T [C] M [t/h], Steam Properties: Thermoflow - STQUIK
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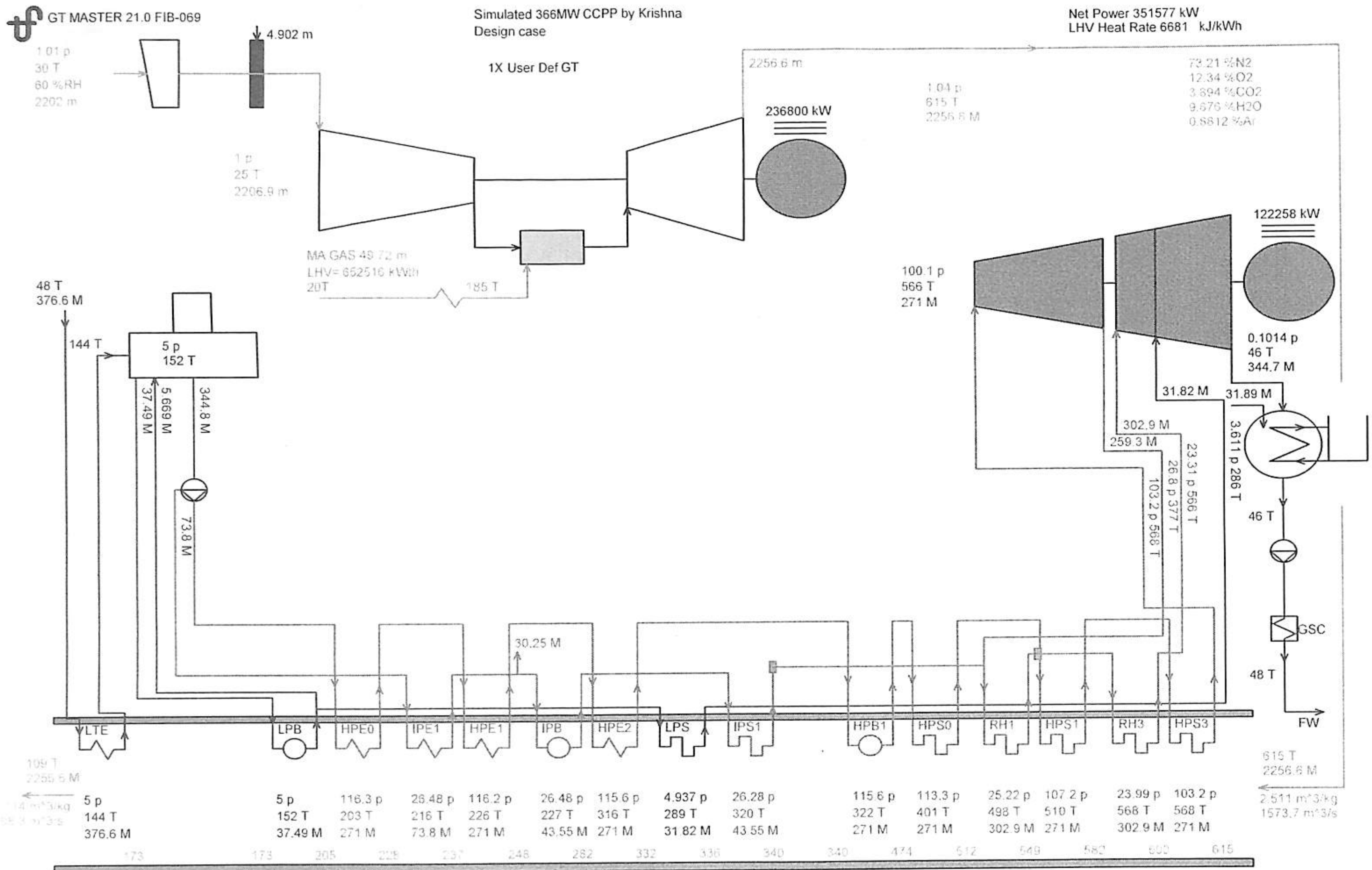
GT MASTER 21.0 FIB-069



p[bar], T[C], M[t/h], Steam Properties: Thermoflow - STQUIK

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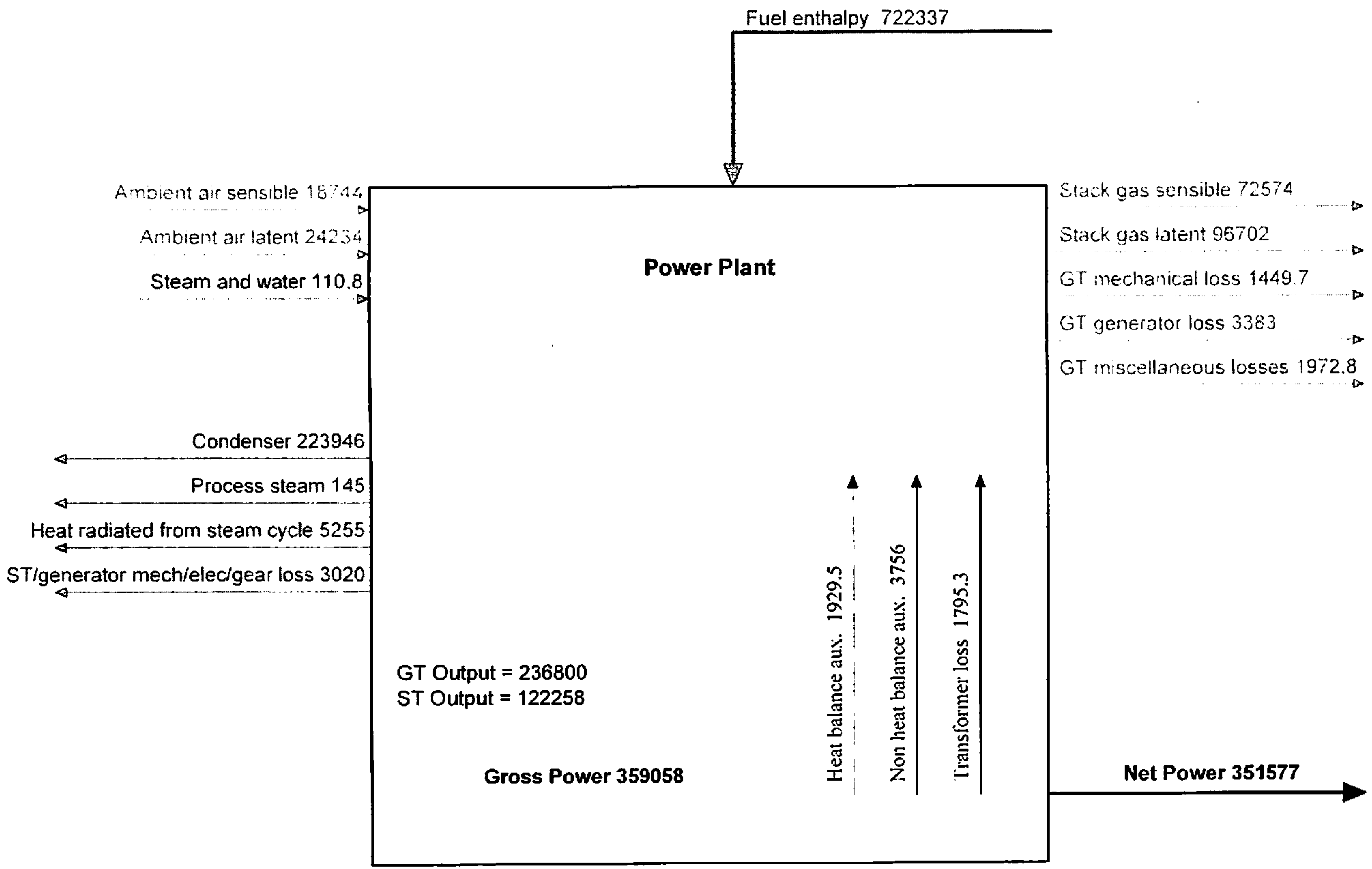
Fuel chemical LHV input = 652516 kW
Fuel chemical HHV input = 719947 kW

Power Plant Energy Flow Schematic [kW]



Zero enthalpy: dry gases & liquid water @ 32 F (273.15 K)

GT MASTER 21.0 FIB-069
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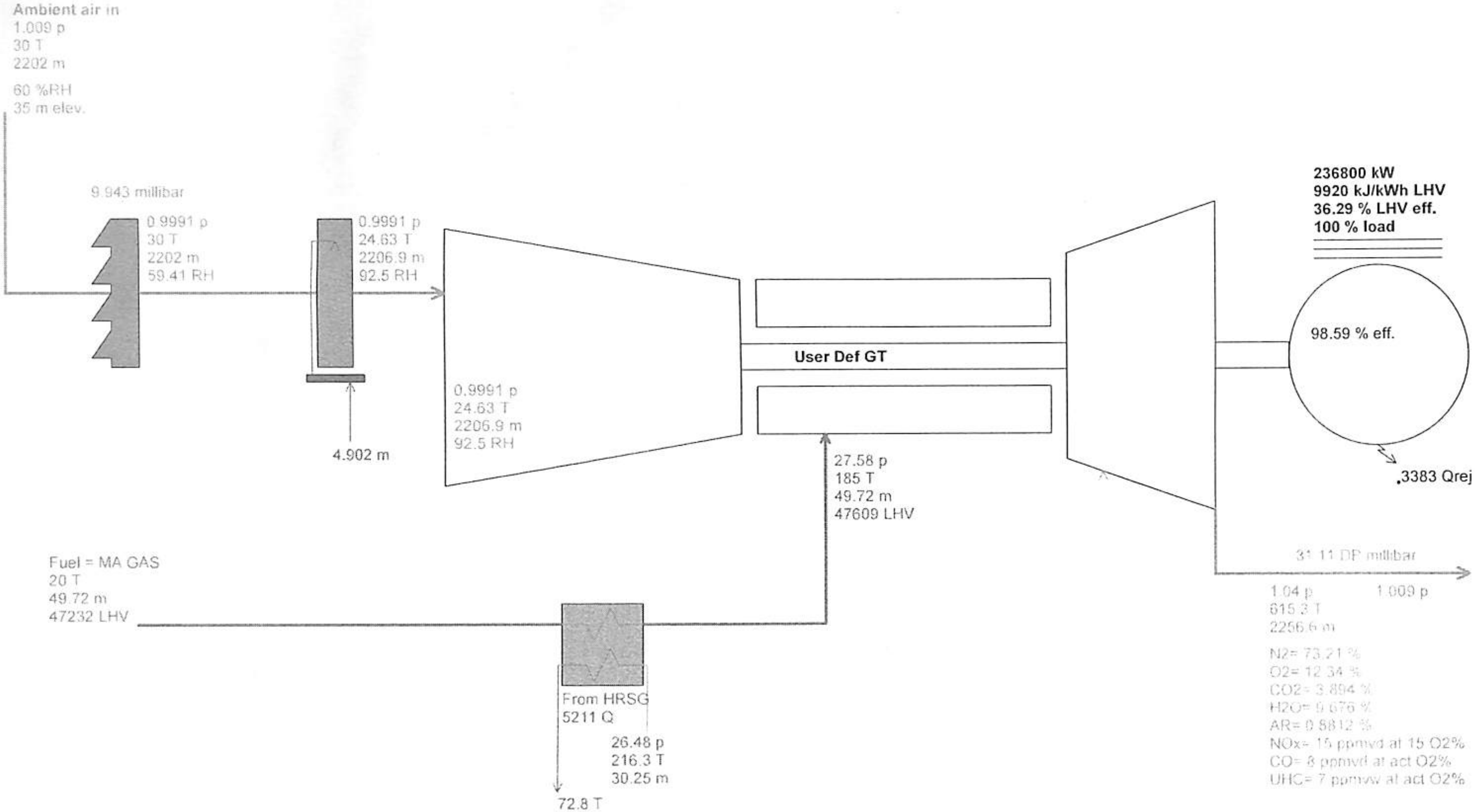
GT generator power = 236800 kW
GT Heat Rate @ gen term = 9920 kJ/kWh
GT efficiency @ gen term = 32.89% HHV = 36.29% LHV



p[bar], T[C], M[t/h], Q[kW], Steam Properties: Thermoflow - STQUIK

GT MASTER 21.0 FIB-069
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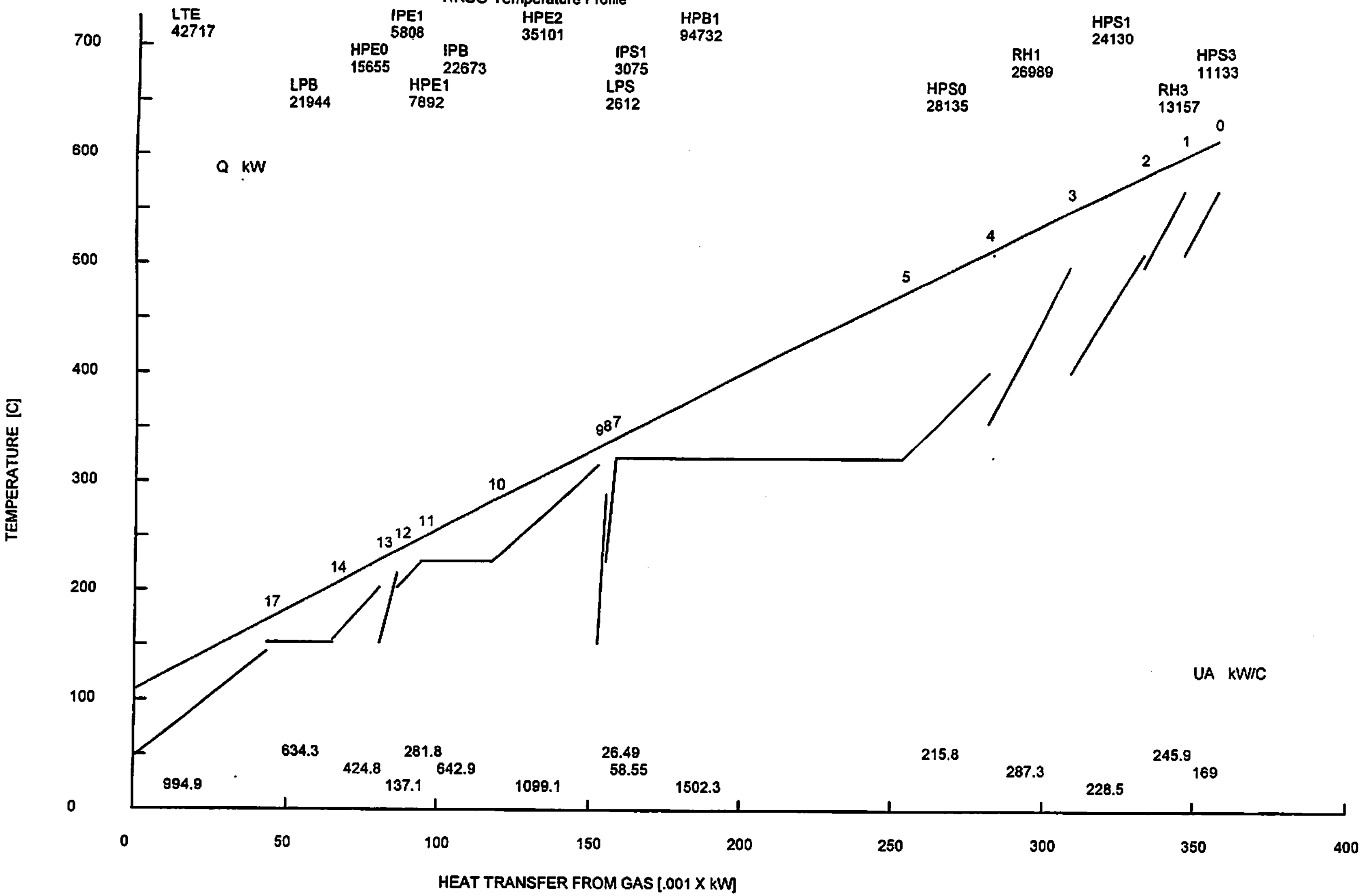
GT MASTER 21.0 FIB-069

Simulated 366MW CCPP by Krishna
Design case
HRSG Temperature Profile

Net Power 351577 kW
LHV Heat Rate 6681 kJ/kWh



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