# Thermodynamic Simulation of A Combined Cycle Power Plant At Part-Load Operation

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ABSTRACT

A 355 MW combined-cycle power plant has been modeled and simulated at part loads. Results of a sensitivity analysis of the effect of high pressure (HP) on the combined-cycle performance are presented. The best combination of process parameters of steam leaving the steam generator that will give optimum performance of the combined-cycle power plant were determined at part load operation. Results for the optimum values of thermal efficiency and power output, together with values of the decision variables, are presented. At part loads, the plant demands lower values of HP pressure to get a suitable dry steam at the steam turbine exit. The combined cycle gives higher power output with high efficiency at full load comparatively at partial loads.

***Keywords***: Sensitivity, parametric, optimization, part load.

***Nomenclature***

| | |
|---|---|
| m | mass, kg/kg mol fuel |
| LHV | lower heating value, kJ/kg mol |
| P | pressure, bar |
| r | compressor pressure ratio |
| T | temperature, K |

| | |
|---|---|
| w | specific work, kJ/kg mol fuel |
| W | work, kJ |
| $\eta$ | efficiency |
| $\gamma$ | ratio of specific heats |

***Suffix***

| | |
|---|---|
| c | compressor |
| cc | combined cycle |
| ex | exhaust |
| f | fuel |
| gc | gas cycle |
| gt | gas turbine |
| HP | high pressure |
| IP | intermediate pressure |
| LP | low pressure |
| p | pump |
| sat | saturation |
| sc | steam cycle |
| st | steam turbine |
| 1 | first law |

## INTRODUCTION

The goal of the thermodynamic optimization of power plant processes is to determine the most favorable combination of process parameters with regard to efficiency and output [1]. The uncertainties in fuel supply and a variation in energy demand run the power plant at different loads. The characteristics of the plant with part loads have been studied in this work. Thermodynamic optimization of energy systems can be based on either energy or exergy. Recently, researchers are paying attention to the heat recovery steam generator (HRSG) to improve combined cycle performance. Pasha and Sanjeev [2] presented a discussion about the parameters that influence the type of circulation and the selection for HRSG. Exergy based optimization is most relevant in process improvements useful in the design of new systems and in the retrofit of old systems. Ongiro et al. [3] developed a numerical method to predict the performance of the HRSG for the design and operation constraints. Ganapathy et al. [4] described the features of the HRSG used in the Cheng cycle system. In this system, a large quantity of steam is injected into a gas turbine to increase electrical power output. The performance of an existing system, especially at part load, can be improved by proper choice of the best process parameters obtained by optimization based on energy analysis. It is, therefore, very important for existing power plants to be operated optimally by the choice of the best process conditions.

When combined-cycle power plants operate at part load, it is necessary to know the best combination of mass flow and steam conditions at entry to the steam turbine, which will give both maximum output and efficiency. Subrahmanyam [5] discussed the various factors affecting the HRSG design for achieving highest combined-cycle efficiency with cheaper, economical and competitive designs and with the highest requirements to meet the shortest deliveries. The steam turbine in a combined-cycle installation operates at sliding pressure with or without sliding temperature. That is, the steam pressure and temperature is not fixed, as in a conventional steam turbine plant, and can be controlled with the flow rate. In practice, the control is affected at the steam generator side. Therefore, for optimum performance of the steam turbine at a given load, it is then necessary to have the best combination of pressure, temperature and mass flow rate of the steam. Noelle and Heyen [6] designed a once-through HRSG that is ideally matched to very high temperature and pressure, well into the supercritical range. Ragland and Stenzel [7] compared four plant designs using natural gas, with a view of cost benefits achieved through HRSG optimization. Casarosa et al. [8] determined the operating parameters by thermodynamic and thermoeconomic analysis, minimizing a suitable objective function by analytical or numerical mathematical methods. Preceding this optimization procedure is the mass and energy balances of individual components and the entire system. This can be carried out with a suitable energy simulation program. To effectively simulate a power plant, it is required that the associated models in the simulation program be able to describe the behavior of individual components of the system under study. In cases where the system components cannot be adequately represented by a single model, a combination of two or more models can represent an approximate behavior of such a system component. A computer simulation program was used in this study.

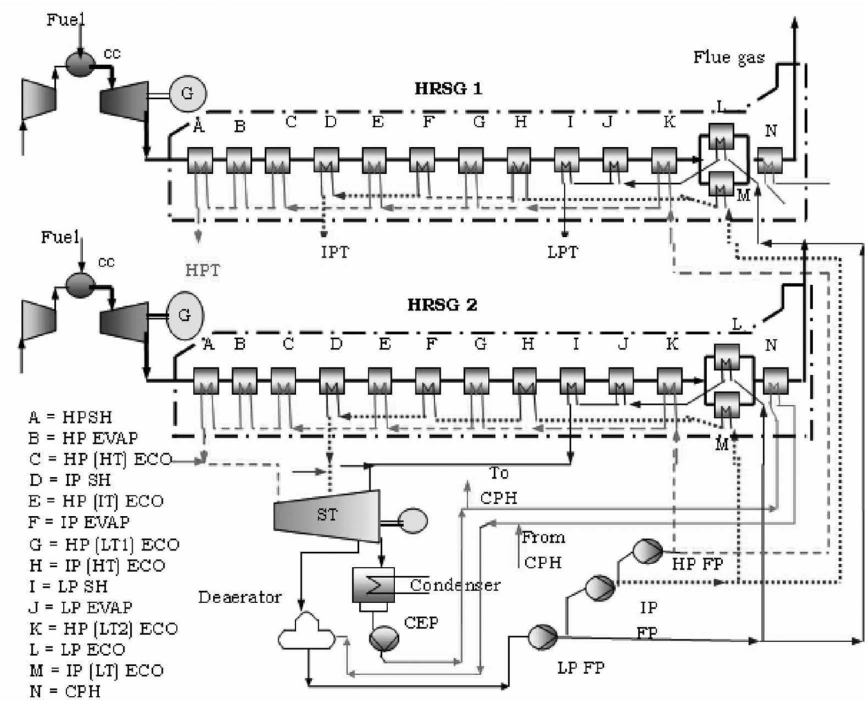
The objective function is taken as HP steam pressure at turbine entry at part load, and this is to be optimized. This leads to maximization of the output and efficiency at part load operation of the combined-cycle power plant.

The constraints associated with this optimization are mainly technological [9] and include: equality constraint placed on the pinch points in HP, IP and LP heaters in the steam generator; satisfaction of mass and energy balances of individual components; and overall system and value corresponding to the value at full load.

## PLANT SIMULATION

A schematic of the combined-cycle plant is shown in Figure 1. The system includes a gas turbine, triple pressure heat recovery steam generator and a steam turbine. The power plant was modeled and simulated with a simulator adapted to completely model all power plant components.

Simulation of the modeled plant has been carried out with a minimum of plant input data to obtain mass and energy balances of individual components and the overall plant. Air at a pressure of 1.01325 bar and a temperature of 25°C enters the compressor, which compresses the air to a pressure of 12.0 bar. The mixture of compressed air and fuel (natural gas) is burned in the burner, and the resulting gaseous products are expelled from the burner at a temperature of 1200°C. The burner was modeled in such a way that the air supply to combustion can be calculated. The pressure loss in the burner has been neglected in this simulation. The products of combustion expand through the turbine to a pressure of 1.04 bar. At full load operation, the gross power developed is 608 MW, of which approximately 41.5 percent is used to drive the compressor and pumps, thereby producing a net power output of 355 MW. The thermal efficiency of the gas turbine plant is 34 percent and to improve the overall fuel utilization, the exhaust gas from the turbine is passed through a heat recovery steam generator to raise steam for expansion in a steam turbine. A combined-cycle system, the energy recovered from a single gas turbine exhaust, is usually insufficient to power a steam turbine because the efficiency of a steam turbine is steeply declined when its size (capacity) becomes too small. To collect enough recovered energy, the combined-cycle system is designed with two gas turbines plus one steam turbine. The heating devices in the HRSG are arranged to get the minimum temperature difference between the flue




CEP: condensate extraction pump
CPH: condensate preheater
ECO: economizer
EVAP: evaporator
FP: feed pump

HRSG: heat recovery steam generator
HT: high temperature
IT: intermediate temperature
LT: low temperature
SH: super heater

**Figure 1. A schematic of combined cycle power plant with triple pressure HRSG**

gas and the water/steam. The enthalpy rise between feed water inlet and steam outlet must equal the enthalpy drop of the exhaust gases in the HRSG. The pressure drop on the gas side of the heat recovery steam generator has been neglected in the present simulation.

The steam turbine is represented in the model by two turbine models, the first to represent the expansion up to the point where steam is bled and the other represents the remaining expansion process. The combination of the two turbine models and the splitter model, which divides the flow passing through it, actually simulates the behavior of the steam turbine with one bleeding point. The power from the steam turbine is 143 MW, and the resulting thermal efficiency is 22.5 percent.

For the gaseous streams, the enthalpy is computed in the simulation program using the enthalpy of formation of the constituents of the streams at 298.15 K and 1.013 bar and of which an arbitrary value of zero is assigned to the enthalpy of the stable elements at this reference datum. Entropy is computed using the enthalpy and Gibbs energy of formation of the components. For liquid water and steam, the reference state is same as used in steam tables.

A procedure for solving the model as presented by Sorgel [10] involves making several guesses of the pressure and temperature at part load and solving iteratively until suitable values are obtained. This manual iteration procedure is replaced in this work by a computer approach involving optimization of the heat recovery from exhaust.

Isentropic efficiencies of the compressor and gas turbine are determined from the pressure ratio ($r_c$), specific heat ratio ($\gamma$) and polytropic state efficiency ($\eta_\infty$) [11].

**Table 1. Assumptions made for thermodynamic evaluation of HRSG in combined cycle.**

| | |
|---|---|
| Atmospheric condition | 25°C and 1.01325 bar |
| Compressor pressure ratio ($r_c$) and maximum temperature | 12 and 1200°C |
| Inlet pressure for HP steam turbine | 200 bar |
| Terminal temperature difference (TTD) -temperature difference between flue gas and reheated/superheated steam | 25 |
| Condenser pressure | 0.05 bar |
| Pinch Point (PP) - minimum temperature differences between the flue gas and the steam in the evaporators of HRSG | 25 |
| Degree of superheat (DSH) in superheater | 50 |
| Polytropic stage efficiency for compressor and gas turbine | 85% |
| Isentropic efficiency of steam turbine | 90% |
| Pressure drop in HRSG, deaerator and condenser is neglected<br>Heat loss in HRSG, turbines, condenser, and deaerator is neglected | |

Isentropic efficiency of compressor,

$$\eta_c = \frac{r_c^{(\frac{(\gamma-1)}{\gamma})} - 1}{r_c^{(\frac{(\gamma-1)}{\gamma\eta_{\infty c}})} - 1} \quad \text{(Eq 1)}$$

Isentropic efficiency of turbine,

$$\eta_t = \frac{1 - (\frac{1}{r_t})^{(\frac{\eta_{\infty c}(\gamma-1)}{\gamma})}}{1 - (\frac{1}{r_t})^{(\frac{(\gamma-1)}{\gamma})}} \quad \text{(Eq 2)}$$

Where $\eta_{\infty c}$ and $\eta_{\infty t}$ is the polytropic efficiencies of compressor and gas turbine, respectively.

The combustion equation in the gas turbine combustion chamber is

$$CH_4 + x\,(O_2 + 3.76\,N_2) \rightarrow CO_2 + 2\,H_2O + (x - 2)\,O_2 + 3.76\,x\,N_2 \quad \text{(Eq 3)}$$

Intermediate pressure (IP) and low pressure (LP) are evaluated from the saturation temperatures. The steam flow rate and the local exhaust temperature in the heating devices (HP, IP, LP and deaerator) are obtained from the heat balance equations.

The saturation temperature of the IP evaporator,

$$T_{IP\ sat} = T_{IP\ ex\ in} - TTD_{IP} - DSH_{IP} \quad \text{(Eq 4)}$$

The saturation temperature of the LP evaporator,

$$T_{LP\ sat} = T_{LP\ ex\ in} - TTD_{LP} - DSH_{LP} \quad \text{(Eq 5)}$$

At HP evaporator the outlet temperature of the flue gas,

$$T_{HP\ ex\ out} = T_{HP\ sat} + PP_{HP} \quad \text{(Eq 6)}$$

At IP evaporator the outlet temperature of the flue gas,

$$T_{IP\ ex\ out} = T_{IP\ sat} + PP_{IP} \quad \text{(Eq 7)}$$

At LP evaporator the outlet temperature of the flue gas,

$$T_{LP\ ex\ out} = T_{LP\ sat} + PP_{LP} \quad \text{(Eq 8)}$$

From the saturation temperature of the evaporators, IP and LP pressures are calculated. The steam flow rates in the heating devices (HP, IP, LP and deaerator) are determined from the energy balance equations.

The enthalpy and temperature of the steam after mixing at the inlet of the IP and LP steam turbine are obtained from the mass and energy balance of the adiabatic mixing process. The flue gas temperatures at different heating zones are calculated with the energy balance equations. The outputs and the inputs of the gas turbine, steam turbines and compressor are related to the unitary mass flow of fuel.

Net steam cycle output, $w_{netsc} = w_{st} - w_p$ (Eq 9)

Total work output, $w_{net} = w_{netgc} + w_{netsc}$ (Eq 10)

The analysis is carried out for one stage of the power plant. The steams from both HRSGs are mixed and carried for expansion in the single steam turbine.

The fuel supply in each gas generator unit,

$$m_f = \frac{\text{Capacity of plant}}{2(W_{net,\ ge} + W_{net,\ se})} \ \text{kg mol} \quad \text{(Eq 11)}$$

Energy efficiency of combined cycle,

$$\eta_{1,cc} = \left(\frac{W_{net\ cc}}{2m_f\ LHV_{CH4}}\right) \times 100 \quad \text{(Eq 12)}$$

Where $W_{net\ cc}$ is the total capacity of the combined-cycle plant.

## RESULTS AND DISCUSSIONS

A sensitivity analysis is performed to study the performance of the gas, steam and combined cycle at part load operation at different HP pressures. The part-load behavior of the power plant is simulated by control of the fuel supply to the gas turbine, and this results in a drop in combustion temperature with lower fuel supply by keeping the air flow rate constant. In all cases considered, there is no supplementary firing. At a particular outside air temperature, the air flow rate is kept constant. The net power output of combined cycle at part load is expressed as fraction of net output at full load.

Figure 2 shows the change in the gas, steam and combined-cycle network output at part load. The work output decreases with decreases in load. Figure 3 shows the topping, bottoming and combined-cycle thermal efficiencies with part load. The efficiency deteriorates very fast at part load. Gas turbine entry temperature and energy input to bottoming cycle decreases at part load because fuel is controlled at part loads with a fixed air supply. It results in a decrease in power and efficiency of the respective cycles. Therefore, the output and efficiency are at their maximum at full load. At full load condition, the net outputs of gas, steam and combined cycle are 214 MW, 141 MW and 355 MW with a thermal efficiency of 34, 22 and 56 percent, respectively.

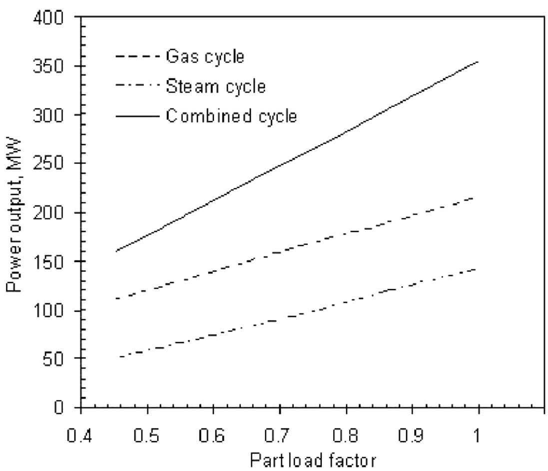



**Figure 2. Cycle net outputs at part load.**

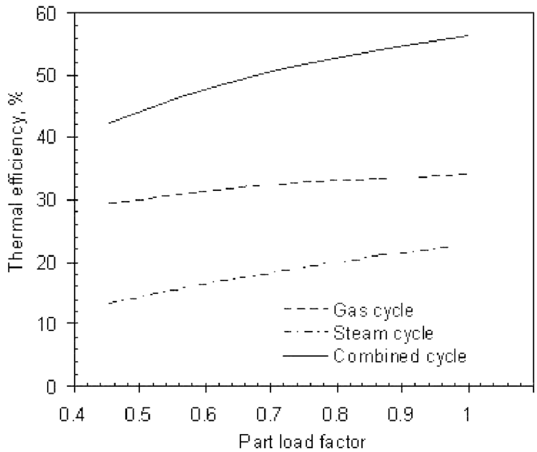



**Figure 3. Thermal efficiencies of cycles at part load**

Figure 4 shows that at a specified outside air temperature, the gas turbine entry and exit temperatures decrease with a decrease in load. At part load, the partial supply of fuel increases the air fuel ratio in the gas turbine combustion chamber. The excess air in the combustion chamber cools the products of combustion. For the same pressure ratio, the exit temperature of the gas turbine decreases with decreases in load. Figure 5 shows the steam turbine inlet temperature at part loads. The steam turbine entry temperature decreases with decreases in load. Steam turbine temperature is determined from gas turbine outlet temperature subtracting the terminal temperature difference. Therefore, low temperature at the exit of the gas turbine decreases the steam temperature at part load.

Figure 6 depicts the effect of the part load factor on stack temperature at different HP pressures. The flue gas temperature at the outlet of the HRSG decreases with increases in load, and increases with increases in HP pressure. At a particular HP pressure, steam generation in HRSG increases from part load to full load. The improved heat recovery at full loads decreases the stack temperature (flue gas exit at HRSG). The LP and IP pressure increases with HP pressure to get a positive heat transfer from the flue gas to the water/steam. The increased pressures with increased saturation temperatures effect so that heat recovery takes place at relatively high temperatures. This leads to an increase in the

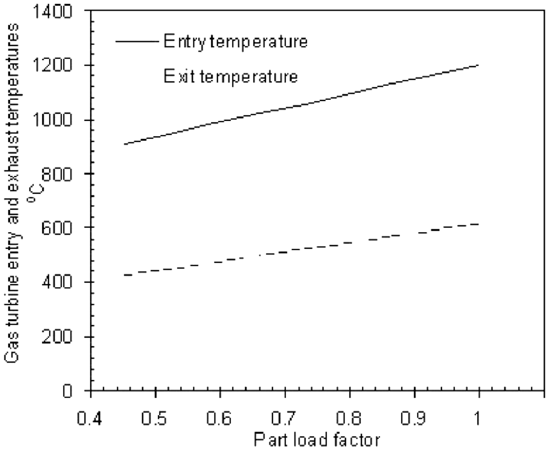



**Figure 4. Gas turbine entry and exhaust temperatures at part load**

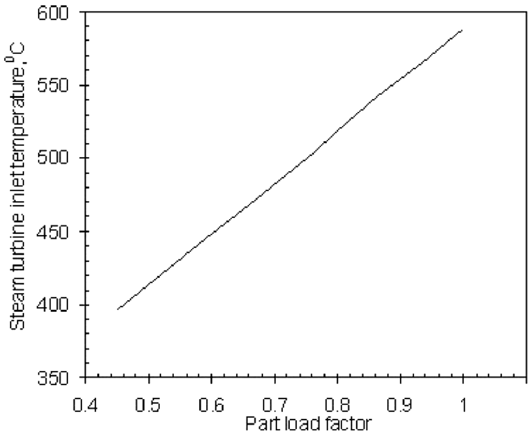



**Figure 5. Temperature of steam at turbine entry at part load**

stack temperature with HP pressure.

Figure 7 shows the effect of part load factor along with HP pressure on thermal efficiency of combined cycle. As mentioned before, the efficiency of a combined cycle decreases with decreases in load. Near full load, the efficiency increases with HP pressure, especially from load factor of 0.7. Below this load factor, a lower HP pressure is preferable because the efficiency decreases with increases in HP pressure. The temperatures of flue gas and steam decrease with decrease in load. So, at this condition, HRSG needs a low HP pressure to get high efficiency.

Figure 8 shows the dryness fraction of steam at the turbine exit

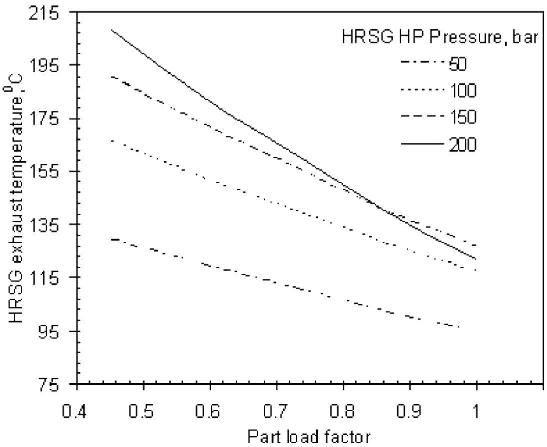



**Figure 6. Stack temperature at part load operation at different HP pressures**

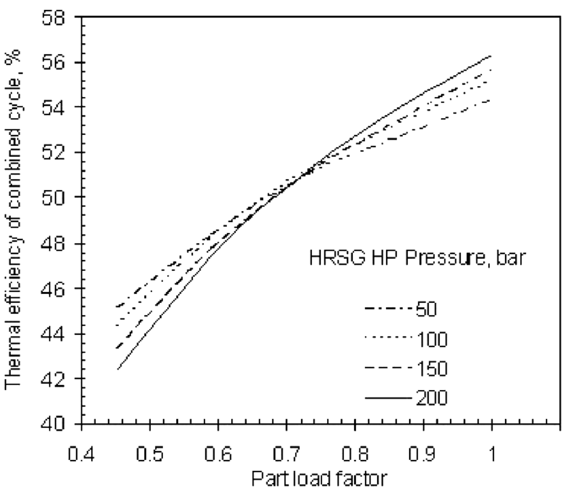



**Figure 7. Thermal efficiency of combined cycle at part load operation at different HP pressures**

with part load factor and HP pressure. Dryness fraction increases with increases in load and decreases with increases in HP pressure. The exit steam is relatively wet at part load and high HP pressure. To run the plant at high HP pressure demands a steam reheater to get a satisfactory dry steam.

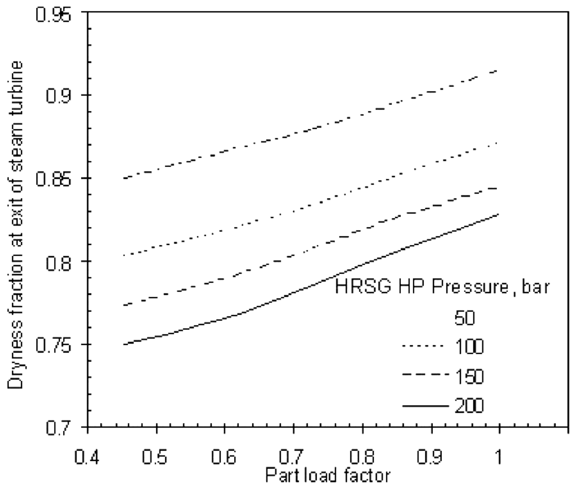



**Figure 8. Dryness fraction of steam at exit of steam turbine at part load operation at different HP pressures**

CONCLUSION

The use of computer simulation software has been used to study the performance of a combined cycle power plant. The responses of the power plant to load changes and HP pressure changes have been presented. The power plant was optimized at part load operation from 50 percent for maximum efficiency and output taking into consideration the associated constraints.

The power output, thermal efficiency, gas temperatures, steam entry temperature to turbine, and dryness fraction of steam decreases with decrease in part load. At part loads, the plant demands low HP pressure in order to get a satisfactory dry steam at turbine exit with high efficiency. It has been recommended to the proposed model to run the plant at full loads to get good results. To run the plant at high HP pressure, it demands a steam reheater.

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