Int. J. of Thermodynamics **Vol. 14 (No. 1), pp. 29-36, 2011**
ISSN 1301-9724 / e-ISSN 2146-1511 www.ijoticat.com

Parametric Simulation of Combined Cycle Power Pant: A Case Study

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Abstract

In this work a thermodynamic evaluation has been carried out on an existing actual combined cycle plant (LANCO power plant) having a triple pressure heat recovery steam generator (HRSG). In this case study, an attempt has been made to improve the efficiency of the plant through a parametric study using a thermodynamic model. The compressor pressure ratio is evaluated from the terminal temperature difference at HRSG inlet instead of an initial regular fixation. The low pressure (LP) and the intermediate pressure (IP) in HRSG also evaluated from local flue gas temperatures with minimum temperature difference in heaters without initial fixation. The optimized results obtained at high pressures (HP) of 90 and 200 bar are compared with the design results of the plant. The optimum pressures obtained for deaerator, LP and IP heaters at 200 bar of HP pressure are 3.7, 8.3 and 26.5 bar respectively.

Keywords: *Combined cycle; pinch point; thermodynamic analysis; triple pressure.*

1. Introduction

Combined cycle power plants are currently one of the most important options for the construction of new generating capacity as well as for the replacement and re-powering of existing units. A heat recovery steam generator (HRSG), which plays an important role in a combined cycle power plant, utilizes the thermal energy of the exhaust gases from the gas turbine to convert feed water to steam which is then supplied to a steam turbine. Thus, the performance of a HRSG is critical for the efficient operation of combined cycle power plants. In this work, the flow diagram of a combined cycle with a triple pressure heat recovery steam generator is taken from the existing LANCO power plant for thermodynamic optimization. The power plant with the capacity of 355 MW consists of two gas turbine generators, two heat recovery steam generators and one steam turbine. The steam cycle output is maximum at a particular deaerator, LP and IP pressure (correspondingly with temperature ratio) for the given HP pressure and working conditions. Therefore a sensitivity analysis is carried out to find the optimum temperature ratios for deaerator, LP and IP heaters in the combined cycle to improve the existing conditions of the plant. This work gives the complete method to carry out the parametric analysis for a triple pressure HRSG.

Many methods have been adopted to improve the efficiency of the combined cycle. Multi-pressure HRSG is one of the important solutions to get high efficiency from a combined cycle. In modern power plants heat addition at constant pressure in the Rankine cycle is solved by the introduction of a second or third pressure level to reduce the mean temperature difference in the boiler. In order to provide better heat recovery in the steam boiler, more than one pressure level is used. With a single pressure HRSG typically about 30 percent of the total plant output is generated in the steam turbine. A dual pressure arrangement can increase the power output of the steam cycle by up to 10 percent, and an additional 3 percent can result by choosing a triple pressure cycle (Marston and Hyre, 1995). Srinivas et al. (2007) developed a mathematical formulation to study the effect of regeneration in a Rankine cycle. Recently the research is focusing in the area of advanced combined cycles to optimize the system. Ramaprabhu and Roy (2004) developed a computational model of a combined cycle power generation unit and applied the model to a plant operated by a local utility company. They compared the simulation results with available plant test data at rated load to evaluate the model. Franco and Casarosa (2002) proposed an analysis of some possibilities to increase the combined cycle plant efficiency to values higher than the 60 percent without resorting to a new gas turbine technology. They also compared the data from the ABB and Siemens plants with the results obtained from the optimization calculations. Franco and Russo (2002) handled the problem adopting both a thermodynamic and a thermo economic objective function rather than the usual pinch point (PP) method. Bassily (2005) presented the effects of varying the inlet temperature of the gas turbine and PP on the performance of a dual pressure reheat combined cycle. He also modeled a feasible technique to reduce the irreversibility of the HRSG of the combined cycle and compared the optimized results with the regularly designed triple pressure reheat combined cycle (Bassily, 2007). Introduction of a multi-pressure steam process in a coal gasification combined cogeneration plant has been discussed in De and Biswal's (2004) work. A larger pressure drop in the high pressure steam turbine is not recommended in their work. Ragland and Stenzel (2000) specified that the plant designs need to be adjusted to the specific project parameters to achieve optimum results.

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Ongiro et al. (1997) developed a numerical method to predict the performance of the HRSG in a fashion that accounts, as much as possible, for the design and operation constraints, while keeping computational complexity manageable. Casarosa et al. (2004) identified the main parameters of a HRSG which are the number of pressure levels, the pressures, the mass flow ratio, and the inlet temperatures to the HRSG sections. Xiang and Chen (2007) carried out a thermodynamic analysis for a triple pressure HRSG to optimize the combined cycle system.

The main objective of this work is to improve the performance of existing combined cycle plants with a parametric analysis. A thermodynamic model of a combined cycle with triple pressure HRSG is taken from the LANCO power plant for the evaluation. The combined cycle is mathematically formulated with triple pressure heat recovery system with a pre-defined set of constrains. The simulated results obtained at a plant with a HP pressure of 90 bar and the optimized HP pressure of 200 bar are compared with the plant design data. In this work, the exergy value of fuel is determined from the method given by Kotas (1985) to determine the exergy efficiency.

2. Thermodynamic model and analysis

The following assumptions have been made during the analysis of the combined cycle.

1. Atmospheric condition is taken as 25°C and 1.01325 bar.
2. Gas cycle maximum temperature is fixed at 1200°C.
3. Steam cycle maximum temperature is 600°C.
4. The condenser pressure is taken as 0.098 bar.
5. The pinch points in HP, IP and LP evaporators (minimum temperature difference between the flue gas and the saturated steam) are taken as 25°C.
6. The terminal temperature difference (TTD) in the HP, IP and LP super heaters (temperature difference between flue gas and superheated steam) are taken as 25°C.
7. The degree of superheat (DSH) in the LP and IP super heater (temperature difference between superheated steam and the saturated steam) is taken as 60°C.
8. The temperature difference between steam and outlet cooling water in condenser is taken as 15°C.
9. Isentropic efficiency of gas turbine is taken as 90 percent.
10. Isentropic efficiencies of compressor and steam turbine are taken as 85 percent.
11. Pressure drops in combustion chamber, HRSG and condenser are neglected.
12. Heat losses in combustion chamber, HRSG, turbines, condenser, and feed water heaters are neglected.

The schematic diagram of a triple pressure HRSG is shown in Figure 1 and the temperature-entropy diagram for the combined cycle in Figure 2. For 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 decreases rapidly when it's size (capacity) becomes too small. In order to collect enough recovered energy, the combined cycle system is designed with two gas turbines plus one steam turbine. Figure 3 is the temperature-heat transferred diagram for a triple pressure HRSG. The heating devices in the HRSG are arranged to get the minimum temperature difference between the flue gas and the water/steam. The enthalpy rise between the feed water inlet and steam outlet must equal the enthalpy drop of the exhaust gases in the HRSG, and the PP and TTD cannot be less than about 20°C if the HRSG is to be of an economic size (Saravanamuttoo et al, 2003). The arrangement of heaters in the HRSG minimizes the exergetic loss in HRSG.

For an isentropic compression process in the compressor, the expression is given as

$$s_{O_2,29'} - s_{O_2,28} + 3.76(s_{N_2,29'} - s_{N_2,28}) - R\left[\log\left(\frac{P_{29}}{P_{28}}\right) + 3.76\log\left(\frac{P_{29}}{P_{28}}\right)\right] = 0 \qquad (1)$$

The temperature of the compressed air at state 29' is determined by iteration of the above equation. The actual temperature of the compressed air (T_{29}) is calculated with the isentropic efficiency of the compressor.

The combustion equation in gas turbine combustion chamber is

$$CH_4 + x\,(O_2 + 3.76\,N_2) \rightarrow CO_2 + 2H_2O + (x - 2)O_2 + 3.76xN_2 \qquad (2)$$

In Eq. (2), x is the amount of air to be supplied per kg mol of methane into the gas turbine combustion chamber. The amount of air required for combustion to get the required temperature (T_{30}) is determined with the energy balance of reactants and products at the temperature of compressed air (T_{29}).

For an isentropic expansion process in the gas turbine, the equation is given as

$$\Delta s = 0 = s_{30} - s_{31'} \qquad (3)$$

$$s_{30} = [s_{CO2,\,30} + 2\,s_{H2O,\,30} + (x - 2)s_{O2,\,30} + 3.76.xs_{N2,\,30}] - R[\log(P_{30}/m_{pc}) + 2\log(2\,P_{30}/m_{pc}) + (x - 2)\log((x - 2)P_{30})/m_{pc}) + 3.76x\log(3.76xP_{30}/m_{pc})] \qquad (4)$$

$$s_{31}' = [s_{CO2,\,31'} + 2\,s_{H2O,\,31'} + (x - 2)s_{O2,\,31'} + 3.76.xs_{N2,\,31'}] - R[\log(P_{31}/m_{pc}) + 2\log(2\,P_{31}/m_{pc}) + (x - 2)\log((x - 2)P_{31})/m_{pc}) + 3.76x\log(3.76xP_{31}/m_{pc})] \qquad (5)$$

In the above equation m_{pc} is the total number of chemical elements in the products of combustion in kg mol. The temperature of the gas after expansion in gas turbine at state 31′ is estimated from the iteration of Eq. (3). The actual temperature of the gas at the outlet of gas turbine (T_{31}) is determined with the gas turbine isentropic efficiency.

From Figure 1, the HP, IP and LP steam from the HRSG enters into the related steam turbines to generate power. Some quantity of steam bled from the intermediate stage of the turbine is used to heat the feed water in the deaerator. The LP and IP pressures are determined with the deaerator pressure, PPs and TTD in the HRSG. In this work, the deaerator, LP and IP pressures are expressed in temperature ratios.

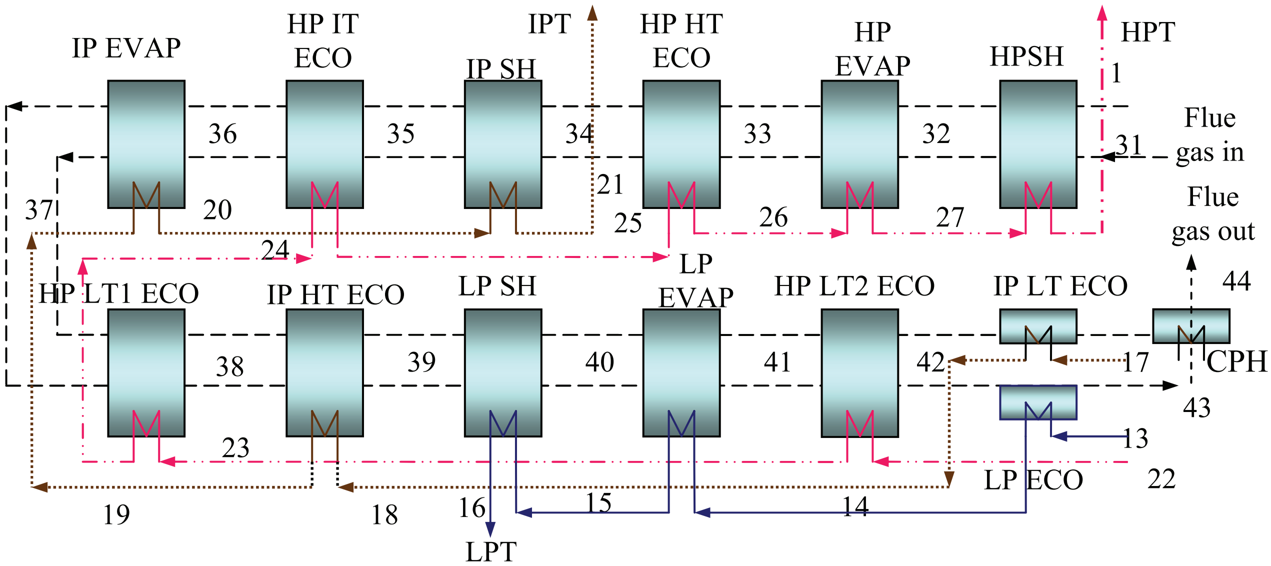

Figure 1. A schematic diagram of a triple pressure HRSG.CPH: condensate preheater; ECO: economizer; EVAP: evaporator; HT: high temperature; IT: intermediate temperature; LT: low temperature; SH: super heater (figure is in color in the on-line version of the paper).

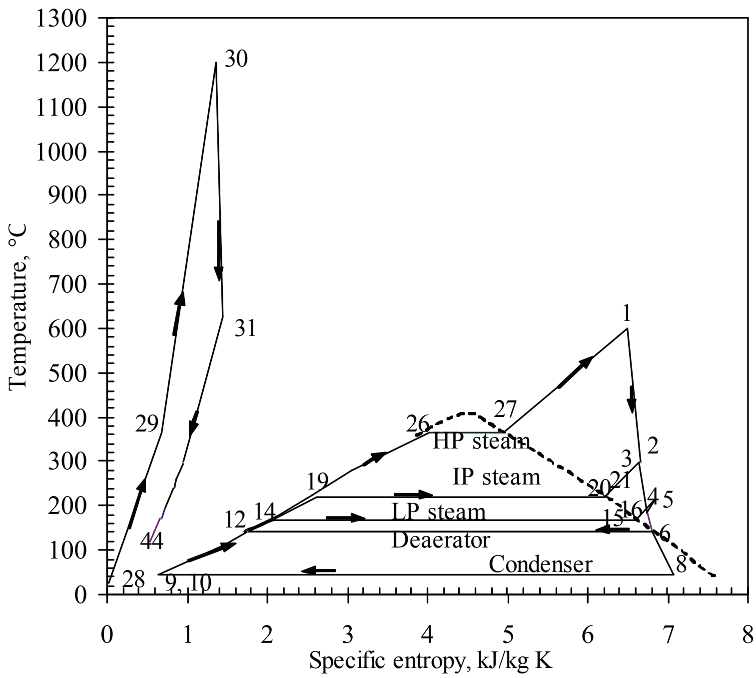

Figure 2. Representation of combined cycle on temperature-entropy diagram with no pressure drop in heating devices.

The overall temperature difference between the HP evaporator and condenser is

$$\Delta T_{OA} = T_{HP\ sat} - T_{co\ sat} \tag{6}$$

The excess temperature ratio for the deaerator to condenser is

$$\theta_{de} = \frac{T_{de\ sat} - T_{co\ sat}}{T_{HP\ sat} - T_{co\ sat}} \tag{7}$$

The excess temperature ratio for the LP pressure to condenser is

$$\theta_{LP} = \frac{T_{LP\ sat} - T_{co\ sat}}{T_{HP\ sat} - T_{co\ sat}} \tag{8}$$

The excess temperature ratio for the IP pressure to condenser is

$$\theta_{IP} = \frac{T_{IP\ sat,} - T_{co\ sat}}{T_{HP\ sat} - T_{co\ sat}} \tag{9}$$

From the PPs in the evaporators, the outlet temperatures of the flue gas at the inlet of HP, IP and LP evaporator are determined.

At HP evaporator the outlet temperature of the flue gas,

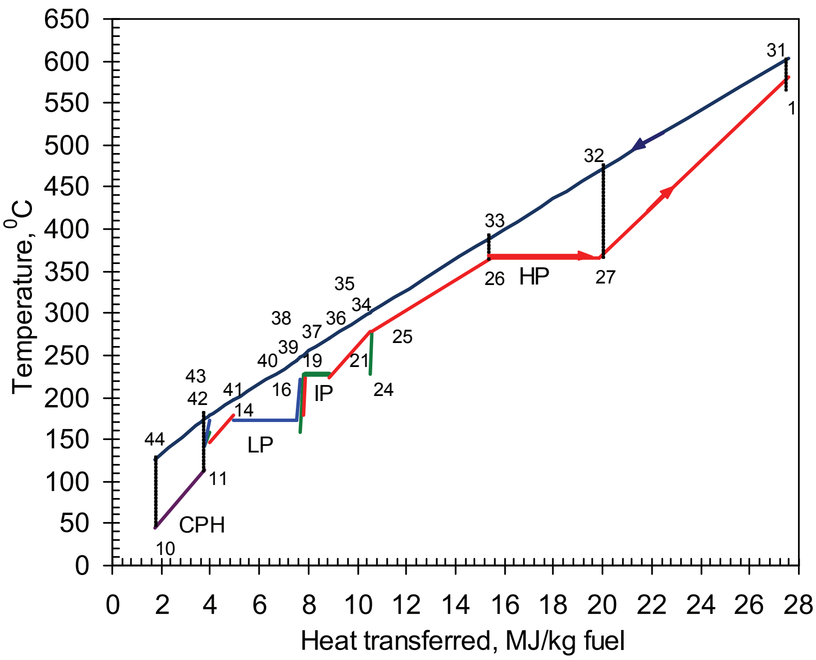

Figure 3. Temperature-transferred heat diagram for triple pressure combined cycle plant for the optimized design at HP pressure of 200 bar (figure is in color in the on-line version of the paper).

$$T_{33} = T_{HP\ sat} + PP_{HP} \tag{10}$$

At the IP evaporator the outlet temperature of the flue gas is

$$T_{37} = T_{IP\ sat} + PP_{IP} \tag{11}$$

At the LP evaporator the outlet temperature of the flue gas is

$$T_{41} = T_{LP\ sat} + PP_{LP} \tag{12}$$

where $T_{IP\ sat}$ and $T_{LP\ sat}$ are the saturation temperatures in IP and LP heaters and determined from the TTD, DSH and with the energy balance equations. From the TTD in the HP super heater and with the gas turbine outlet temperature, the HP superheated steam temperature, i.e. HP steam turbine inlet temperature, is determined.

$$T_1 = T_{31} - TTD_{HP} \tag{13}$$

From the TTD at the IP and LP super heaters the superheated steam temperatures of IP and LP heaters are determined from the flue gas temperature. The flue gas temperatures at states 34 and 39 are obtained from the energy balance equations in HP HT ECO and IP HP ECO respectively.

$$T_{21} = T_{34} - TTD_{IP} \tag{14}$$

$$T_{16} = T_{39} - TTD_{LP} \tag{15}$$

The saturation temperature of the IP evaporator is

$$T_{IP\ sat} = T_{34} - TTD_{IP} - DSH_{IP} \tag{16}$$

The saturation temperature of the LP evaporator is

$$T_{LP\ sat} = T_{39} - TTD_{LP} - DSH_{LP} \tag{17}$$

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 steam generated in the HP heaters is

$$m_1 = \frac{h_{31} - h_{33}}{h_1 - h_{26}} \text{ kg/kg mol fuel} \tag{18}$$

The steam generated in the IP heaters is

$$m_2 = \frac{h_{34} - h_{37} - m_1(h_{25} - h_{24})}{h_{21} - h_{19}} \text{ kg/kg mol fuel} \tag{19}$$

The steam generated in the LP heaters is

$$m_3 = \frac{h_{39} - h_{41}}{h_{16} - h_{14}} \text{ kg/kg mol fuel} \tag{20}$$

The steam extracted for the deaerator is

$$m_4 = \frac{(m_1 + m_2 + m_3)(h_{12} - h_{11})}{h_6 - h_{11}} \text{ kg/kg mol fuel} \tag{21}$$

The enthalpy and temperature of the steam after mixing at the inlet of the IP and LP steam turbines are obtained from the mass and energy balance of the adiabatic mixing processes. The flue gas temperatures at different heating zones are calculated with the energy balance equations. The outputs and the inputs of the gas turbines, steam turbines and compressors are related to the unitary mass flow of fuel. For 1 kg mol of fuel the gas cycle work input and output are as follows

The gas cycle energies are

$$w_{gt} = h_{30} - h_{31} \tag{22}$$

$$w_c = h_{29} - h_{28} \tag{23}$$

$$w_{netgc} = w_{gt} - w_c \tag{24}$$

The steam turbine work is

$$w_{st} = [m_1(h_1 - h_2) + (m_1 + m_2)(h_3 - h_4) + (m_1 + m_2 + m_3)(h_5 - h_6) + (m_1 + m_2 + m_3 - m_4)(h_6 - h_8)] \tag{25}$$

The pump work is

$$w_p = [(m_1 + m_2 + m_3 - m_4)(h_{10} - h_9) + (m_1 + m_2 + m_3)(h_{13} - h_{12}) + (m_1 + m_2)(h_{17} - h_{13}) + m_1 (h_{22} - h_{17})] \tag{26}$$

The net steam cycle output is

$$w_{netsc} = w_{st} - w_p \tag{27}$$

The total work output is

$$w_{net} = w_{netgc} + w_{netsc} \tag{28}$$

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 is

$$m_f = \frac{Capacity\ of\ plant}{2(w_{net,\ gc} + w_{net,\ sc})}\ \text{kg mol} \tag{29}$$

The lower heating value and chemical exergy ($\bar{\varepsilon}^0$) of natural gas are taken as 802,303 kJ/kg mol and 836,420 kJ/kg mol (Kotas, 1985) for energy and exergy efficiency respectively. To determine the exergetic losses, the irreversibilities associated with all the components must be estimated through an exergy analysis. The chemical and physical exergies are determined at each state by using the following equations (Bejan, 1982).

Chemical exergy,

$$e_{ch} = \sum_k n_k \overline{\varepsilon^0{}_k} + RT_0 \Sigma_k n_k \ln .[P.x_k] \tag{30}$$

where x_k is the mol fraction of x^{th} compound.

$$\text{Physical exergy} = e_{ph} = h - \Sigma_k T_0 s_k \tag{31}$$

$$\text{Exergy, } e = e_{ch} + e_{ph} \tag{32}$$

The energy efficiency of the combined cycle is

$$\eta_{1,cc} = \left(\frac{W_{net\ cc}}{2m_f\ LHV_{CH4}} \right) \times 100 \tag{33}$$

The exergy efficiency of the combined cycle is

$$\eta_{2,cc} = \left(\frac{W_{net\ cc}}{2m_f\ \bar{\varepsilon}^0{}_{CH4}} \right) \times 100 \tag{34}$$

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

3. Results and Discussions

In this work, a sensitively analysis has been carried out to examine the effect of TTD at HRSG inlet, combustion chamber temperature, deaerator temperature ratio and HP pressure on the efficiency of the combined cycle.

In the current study, the IP and LP temperature ratios are determined with the available flue gas temperatures to transfer the heat between fluids with minimum temperature difference at the fixed deaerator temperature ratio. The LP, IP and HP pressures are maintained above the deaerator pressure. The IP, LP and deaerator pressure increases with increases in the HP pressure. At 200 bar the deaerator, LP and IP pressures are 3.7, 8.3 and 26.5 bar respectively at the given working conditions.

Figure 4 shows the effect of TTD at HRSG inlet (temperature difference between flue gas and HP superheated steam temperature) with gas turbine inlet temperature on compressor pressure ratio. The steam plant parameters are kept fixed for this study while the gas plant parameters (gas turbine inlet temperature and compressor pressure ratio, and therefore gas turbine exhaust temperature) are varied. The TTD at HRSG inlet is varied from 10 to 50°C. The compressor ratio increases with an increase in gas turbine inlet temperature to get the same exhaust temperature. A simple gas cycle results in low pressure ratios (6 to 16) with steam turbine inlet temperature of 600°C at 200 bar HP. The compressor pressure ratio is independent of HRSG and steam plant variations. It is iterated from TTD at HRSG inlet (function of gas turbine exhaust temperature) and gas turbine inlet temperature (T_{max}).

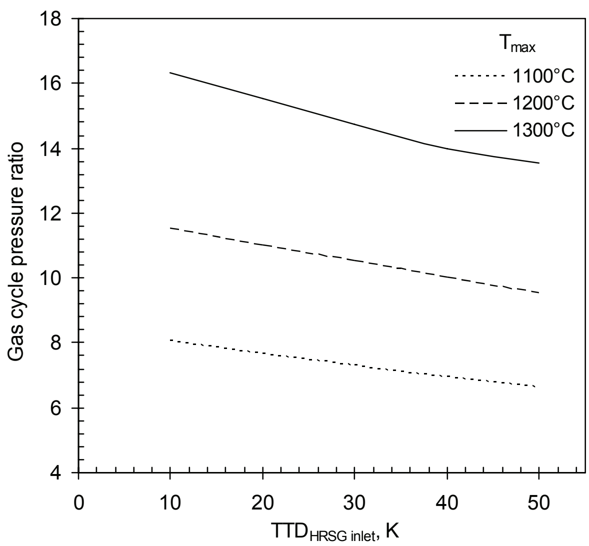

Figure 4. Cycle pressure ratio as a function of TTD at HRSG inlet and gas turbine inlet temperature.

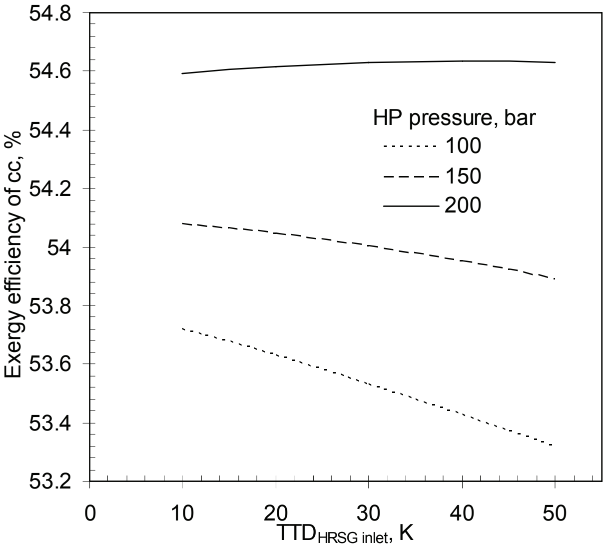

Figure 5. Influence of TTD at HRSG inlet and HP pressure on exergy efficiency of combined cycle plant.

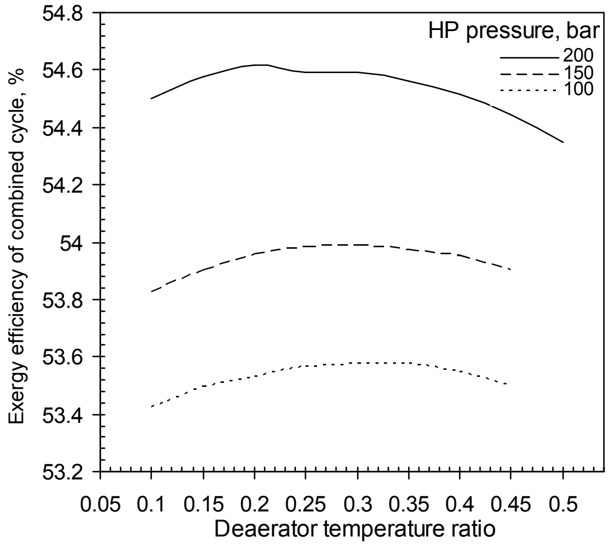

Figure 6. Effect of deaerator temperature ratio on exergy efficiency of combined cycle with HP pressure in HRSG.

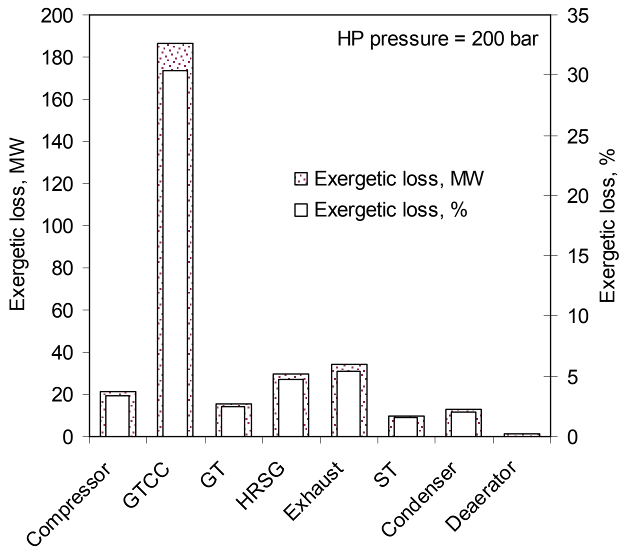

Figure 7. Comparison of exergetic losses in the combined cycle plant components expressed in MW and percent of exergy of fuel at HP pressure of 200 bar.

The variation in combined cycle exergy efficiency with the TTD at HRSG inlet and HP pressure is shown in Figure 5. The exergy efficiency of the combined cycle is evaluated in percent of standard chemical exergy of fuel. The efficiency of the cycle increases with increases in the HP pressure. The variation in exergy efficiency decreases with an increase in HP pressure. The optimum pressure ratio decreases and optimum TTD increases with an increase in HP pressure because these two (pressure ratio and TTD) are inversely related with each other. The lower values of HP pressure demands low TTD at HRSG inlet.

Figure 6 develops the effect of the deaerator temperature ratio on exergy efficiency of the combined cycle with different HP pressures. To get a maximum efficiency of the combined cycle, the deaerator location should be optimized. The deaerator is an open feedwater heater and its pressure is maintained in between the condenser and LP pressures. Srinivas et al. (2007) generated efficiency curves for a single feed water heater and got the optimum temperature ratio in between the condenser and boiler temperature ratios. Similarly in the current combined cycle plant also the optimum deaerator temperature ratio lies in between the condenser and LP temperature ratios. The efficiency of the combined cycle initially increases with an increase in the deaerator temperature ratio but then decreases with further increases in the temperature ratio. The optimum deaerator temperature ratio decreases with increases in the HP pressure. The optimum deaerator temperature ratio at 100, 150 and 200 bar HP pressure is 0.35, 0.3 and 0.25 respectively and the maximum exergy efficiency is 53.7, 54.1 and 54.6 percent respectively.

Figure 7 compares the exergetic loss in MW and percent of exergy of fuel for the components of the combined cycle plant and the exhaust. The major exergetic loss occurs in the gas turbine combustion chamber (187 MW @ 30 percent) and subsequently in the exhaust gas. The exergetic loss in the steam bottoming cycle (components) is less compared to the loss in the gas cycle. The smallest exergetic loss occurs in the deaerator due to a small quantity of steam being used for feed water heating. The exergetic loss in HRSG is only on the order of 4.5 percent (30 MW) where as for steam boiler in a conventional steam power plant this loss is more.

Table 1 compares the plant design parameters of HP, IP, LP and deaerator pressures with the optimized results obtained through the thermodynamic analysis at 90 bar and 200 bar of HP pressure. The results obtained from the analysis at the plant HP pressure (90 bar) and at given working conditions are reasonable and agree with the data obtained from the plant. The same results are again determined with the HP pressure of 200 bar to get the higher efficiency for the combined cycle.

Table 2 compares the results of net work output from the gas, steam and combined cycle at HP pressure of 90 and 200 bar with the plant design values. A thermodynamic model is prepared at the net plant output of 355 MW. The obtained gas cycle output is slightly increased compared to the existing plant gas cycle output. But the steam cycle output is decreased compared to the actual steam cycle output. The energy efficiency of the combined cycle based on the lower heating value of the fuel is compared with the plant efficiency. The optimized efficiencies from the thermodynamic evaluation at the HP pressure of 90 and 200 bar are higher compared to the actual plant efficiency.

Table 1. A comparison of HP, IP, LP and deaerator pressures in HRSG with actual values.

Section	Pressure, bar		
	Plant design	**Simulated results at plant HP pressure (90 bar)**	**Simulated results at HP pressure of 200 bar**
HP Steam	90	90	200
IP Steam	20	16.4	26.5
LP Steam	4.8	4.4	8.3
Deaerator	1.4	2.2	3.7

Table 2. A comparison of net work outputs with the actual values.

Cycle	Net output of plant, MW		
	Plant design	**Simulated results at plant HP pressure (90 bar)**	**Simulated results at HP pressure of 200 bar**
Gas cycle	224.8	234.3	230.3
Steam cycle	128.5	120.6	124.7
Combined cycle	353.3	355.0	355.0
Energy efficiency, %	47.0	55.9	56.9

4. Conclusions

An existing combined cycle power plant with triple pressure HRSG is examined thermodynamically. The analysis is carried out with the HP pressure of 90 bar (plant reading) and 200 bar. The gas cycle pressure ratio decreases with an increase in TTD at HRSG inlet. The optimum deaerator pressure decreases with increases in the HP pressure. The major and minor exergetic losses in the combined cycle occurs in the combustion chamber and deaerator respectively. The optimized pressures for the deaerator, LP and IP heaters at 200 bar of HP pressure are 3.7, 8.3 and 26.5 bar respectively. The pressure ratio obtained for the gas cycle is around 10-12 at 1200°C with the variation in TTD at HRSG inlet.

Acknowledgements:

Authors T. Srinivas and B.V. Reddy acknowledge the financial support from NSERC, Canada for the research work. The authors would like to express their thanks to the members of LANCO power plant for their warm consideration and data support in this research work.

Nomenclature

e	specific exergy, kJ/kg mol
h	specific enthalpy, kJ/kg mol
m	mass, kg/kg mol fuel
LHV	lower heating value, kJ/kg mol
P	pressure, bar
R	universal gas constant, kJ/kg mol K
s	specific entropy, kJ/kg mol
T	temperature, K
w	specific work, kJ/kg mol fuel
W	work, kJ
η	efficiency
$\bar{\varepsilon}$	specific exergy at ground state, kJ/kg mol
θ	excess temperature ratio

Subscripts

c	compressor
cc	combined cycle
co	condenser
de	deaerator
f	fuel
gc	gas cycle
gt	gas turbine
HP	high pressure
IP	intermediate pressure
LP	low pressure
OA	over all
p	pump
pc	products of combustion
sat	saturation
sc	steam cycle
st	steam turbine
0	reference state
1	first law
2	second law

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