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Parametric Analysis of Triple Pressure HRSG in Combined Cycle Power Plant (September 2006)

N. Ravi Kumar, Sk. Jaheeruddin, Dr. K. Rama Krishna, and Dr. A. V. Sita Rama Raju

Abstract - *Combined cycle power plants play an important role in the present energy sector. Combined Cycle plants couple a Brayton cycle with a Rankine cycle. For the combined power plants, the optimization of the heat recovery steam generator (HRSG) is of particular interest in order to improve the efficiency of the heat recovery from gas turbine exhaust to maximize the power production in the steam cycle. In the present study effect of different operating variables such as exhaust gas flow rate, exhaust gas temperature, pinch point, approach point, steam temperature, steam pressure on the performance of HRSG has been investigated. Triple pressure steam cycle is considered for the bottoming cycle to reduce irreversibilities during heat transfer from gas to water/steam. The cycle is analysed by using energy and exergy analysis. It is observed that in triple pressure cycle selection of IP and LP pressures, and LP pinch are identified as dominant parameters having impact on heat recovery steam generator performance.*

Key words - **Combined cycle, exergy analysis, heat recovery steam generator, optimization**

1. INTRODUCTION

Combined cycle power plants pose a very good alternative to conventional thermal power plants due to better thermodynamic performance and reduced environment pollution. In a typical combined cycle plant gas turbine is the topping cycle and the steam turbine is the bottoming cycle. The major components that make up a combined cycle are the gas turbine, the HRSG, the steam turbine. There are many concepts of the combined cycle, these cycles range from the single pressure cycle, in which the steam for the turbine is generated at only one pressure, to the triple pressure cycles where steam generated for the steam turbine is at three different levels. The schematic of Simple combined cycle plant is shown in Fig. 1.The practical design of the HRSG is usually based on the concepts of pinch point and approach point that govern the gas and steam temperature profiles given by Linhoff and Hindmarsh [1]. The pinch point represents the difference between the gas temperature leaving the evaporator and the saturation temperature, while the approach-point temperature is the difference between the water temperature leaving the economizer and the saturation temperature. Pinch and approach points take into account both thermodynamic and economical points of view. Subrahmanyam et al. [2] discussed various factors affecting the HRSG design for achieving highest combined cycle efficiency with cheaper, economical and competitive designs and with highest requirements to meet the shorter deliveries. P.K. Nag and D.Raha [3] made thermodynamic analysis of a coal based combined cycle power plant from the view points of both first law and second law. Horlock [4] presented detailed thermodynamic analyses of various schemes of combined power plants. A comprehensive computer simulation of an HRSG was developed by A.Ongiro et al. [5] to predict the performance of heat recovery steam generator (HRSG).Nag and Mazumder, [6] reported the thermodynamic optimization of a waste heat recovery boiler with an economizer and evaporator and producing saturated Steam. They have reported the effect of different operating parameters on entropy generation for the waste heat recovery boiler.

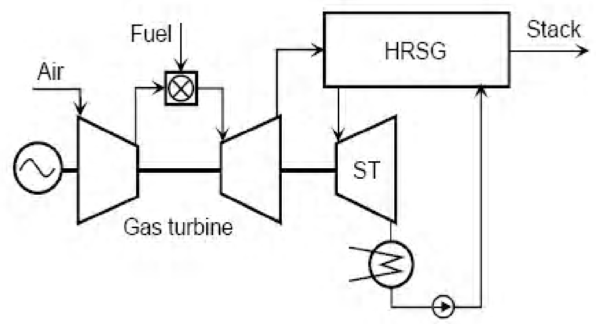

Fig. 1. Schematic of combined cycle power plant.

Dallenback [7] proposed an alternative regenerator configuration to improve the efficiency of gas turbine cycle. Ravi kumar et al. [8] has done exergy analysis for the alternative configuration to know exergy losses in different components. It is observed that the irreversibility in exhaust gases is low which indicates effective utilization of heat energy, but the specific work output of the turbine is less. Gas turbine is seen to offer high specific work output if the turbine inlet temperature (TIT) could be increased. Increase in TIT has strict metallurgical limitations in terms of maximum temperature that the turbine stage could withstand. Ravi kumar and Sita rama raju [9] analyzed the effect of inlet

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cooling on HRSG performance. It is found that the inlet cooling reduces the work input of the compressor and increases the mass flow rate of air. The efficiency of steam cycle can be improved by increasing the temperature of steam entered into the steam turbine. The maximum temperature of steam that can be used is fixed from metallurgical considerations. P. K. Nag and S. De [10] applied thermodynamic analysis to the optimal design and operation of a heat recovery steam generator (HRSG) generating saturated steam for a combined gas and steam power cycle. The aim of the present paper is to study the effect of different operating parameters on triple pressure HRSG configuration in combined cycle power plant by using energy and exergy analysis.

2. ANALYSIS

In a combined cycle power plant the HRSG represents the interface element between the gas turbine and the steam cycle. Here, the gas turbine exhaust gas is cooled down and the recuperated heat is used to generate steam. In order to provide better heat recovery in the HRSG more than one pressure level is used. HRSGs for gas turbine exhaust are usually designed in unfired conditions and the performance evaluated at other unfired or fired conditions. The reason for this is that two of the important variables which affect the gas and steam temperature profiles, namely the pinch and approach points, cannot be arbitrarily selected in the fired mode. The pinch point is the difference between the gas temperature leaving the evaporator and the temperature of saturated steam. Approach point is the difference between the temperature of saturated steam and the temperature of water entering the evaporator. The schematic diagram of the combined cycle power plant with triple pressure HRSG, for which the analysis has been conducted, is shown in Fig. 2.The air is taken into the compressor at atmospheric pressure and compressed to the desired pressure. The compressed air is sent into the combustion chamber and raised its temperature by burning the fuel. The steam turbine utilizes the energy in the exhaust gas of the gas turbine as its input energy. The following assumptions are taken for the present analysis

(1) The system is in steady state
(2) There is no pressure drop in water and steam side
(3) There is no external heat and mass transfer
(4) The specific heats of exhaust gas and water are constant
(5) The pressure drop on the gas side does not have a significant effect on its temperature, Maximum HP steam superheat temperature is 500^{0}C, Minimum pinch point temperature difference 15^{0}C, Minimum stack temperature100^{0}C, Maximum working pressure 200 bar, C_{pa} = 1.005 kJ/kgK, C_{pg} = 1.147kJ/kgK, R_g = 0.287, γ_a = 1.4, γ_g = 1.33, η_c = 89%, η_t = 87%, hl = 0.02, fpl = 0.03, ψ = 1.04.

$$p_2 = r_p * p_1 \tag{1}$$

$$T_{2s} = T_1 (r_p)^{(\gamma_a - 1)/\gamma_a} \tag{2}$$

$$T_2 = T_1 + \frac{(T_{2s} - T_1)}{\eta_C} \tag{3}$$

$$W_C = C_{pa}(T_2 - T_1) \tag{4}$$

$$P_3 = 1 - fpl * P_2 \tag{5}$$

$$T_{4s} = T_3 \left(\frac{P_4}{P_3}\right)^{\frac{(\gamma_g - 1)}{\gamma_g}} \tag{6}$$

$$W_{gt} = \eta_t C_{pg} (T_3 - T_{4s}) \tag{7}$$

$$W_{net} = W_t - W_C \tag{8}$$

For the gas turbine process

$$\eta_{gt} = \frac{W_{net}}{m_f * CV} \tag{9}$$

The combined efficiency of the cycle is

$$\eta_{CC} = \frac{W_{gt} + W_{st}}{m_f * CV} \tag{10}$$

$$\Delta H_0 = m_f * CV \tag{11}$$

$$\Delta G_0 = \varphi * \Delta H_0 \tag{12}$$

$$T_0 * \Delta s_0 = \Delta G_0 - \Delta H_0 \tag{13}$$

$$\eta_{sec} = \frac{W_{gt} + W_{st}}{\Delta G_0} \tag{14}$$

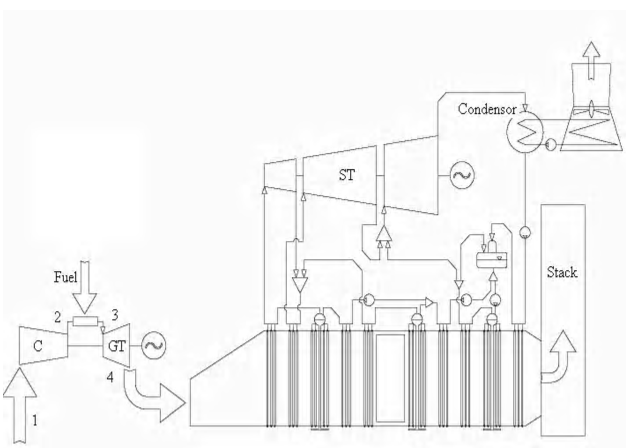

Fig. 2. Schematic of triple pressure HRSG.

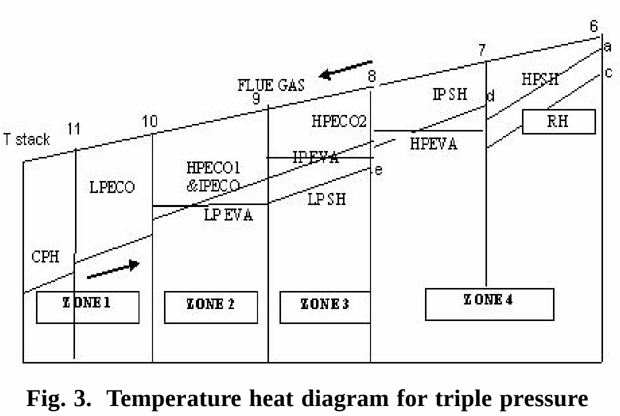

Fig. 3. Temperature heat diagram for triple pressure HRSG.

A computer program is developed to calculate the mass of steam generated in different sections, dryness fraction of steam, stack temperature, work output, irreversibilities in turbine and HRSG are calculated. The results obtained from the analysis are represented in form of graphs and discussed in the next section.

3. RESULTS AND DISCUSSION

Figure 4 discussed the effect of maximum steam temperature on mass flow rate of steam. With increase in steam temperature the HP mass is decreased because more heat is required and the IP, LP mass of steam is increased.

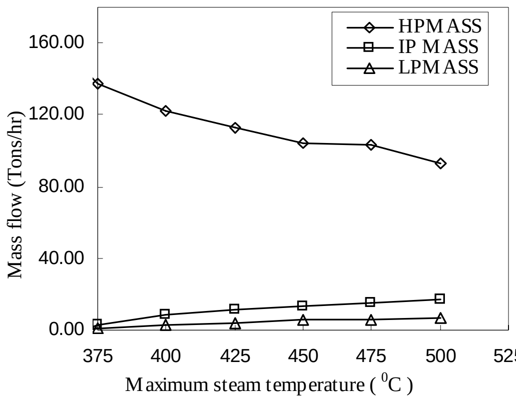

Fig. 4. Effect of steam temperature on mass flow of steam.

Figure 5 discussed the effect of steam temperature on efficiency of the bottoming cycle and stack temperature. With increase in steam temperature the stack temperature and efficiency also increased. For steam turbine increasing the live steam temperature means less erosion in the final stages but too high a live steam temperature can also cause a disproportionate increase in plant costs since a great amount of expensive material is required for the piping, the superheater and the steam turbine. In most cases the exhaust gas temperature sets the limit for the live steam temperature level because a sufficient difference in temperature is necessary between the exhaust gas and the live steam temperature in order to limit the size of the superheater. In triple cycle while selecting the intermediate pressure and low pressure steam temperature, the difference in temperature between the high pressure steam after expansion and the low pressure steam at the mixing point in the turbine must be taken into account. If the difference is too high, it causes unnecessary thermal stresses with in the turbine. But a high low pressure steam temperature presents the advantage of a kind of a low 'reheating', reducing the risk of erosion due to wetness in the turbine.

Figure 5 represents the change in mass flow rate of steam with gas turbine exhaust (HRSG inlet) temperature. With increase in temperature the HP mass of steam increases and LP mass of steam is decreased.

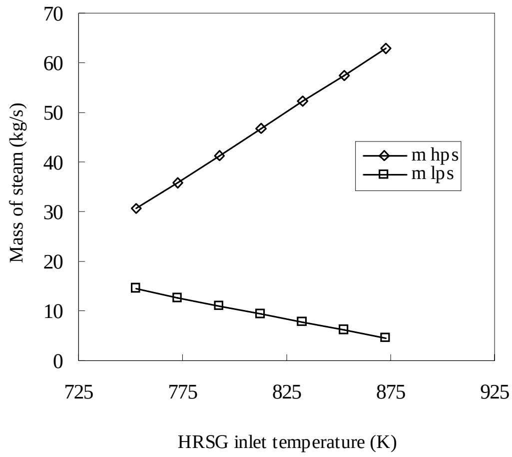

Fig. 5. Effect of Gas turbine exhaust on mass of steam.

Figs. 6 and 7 shows the effect of steam pressure on optimum IP and LP pressures and mass of steam. With increase in steam pressure the optimum IP and LP pressures are increased. In a combined cycle plant, a high live steam pressure brings an increased efficiency of the water/steam cycle due to the greater enthalpy gradient in the turbine. The rate of waste heat energy utilization in the exhaust gases however drops off sharply. From economical point of view it is advisable raising the steam pressure above the thermodynamic equilibrium, because it causes a reduction in exhaust steam flow which requires a smaller condensor and less cooling water requirement.

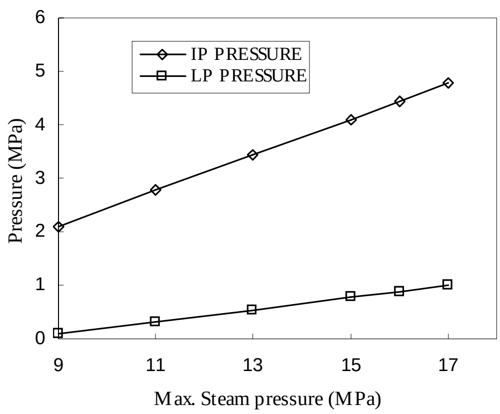

Fig. 6. Effect of maximum steam pressure on optimum IP and LP steam pressures.

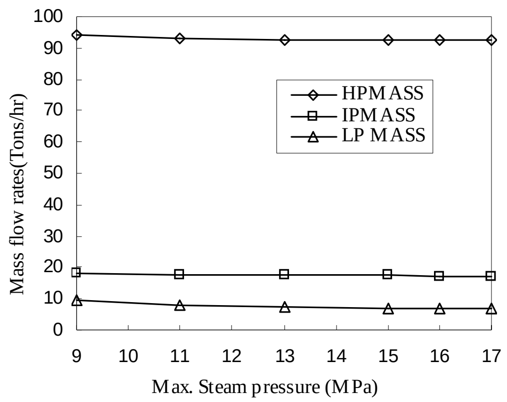

Fig. 7. Effect of maximum steam pressure on mass of steam.

Table 1. Effect of Pinch Points on Stack Temperature, Efficiency and Stack Irreversibility

HP pinch	IP pinch	LP pinch	Stack	Efficiency	I stack
10	10	10	93.73	38.66244	5687
15	10	10	93.54	38.50814	5650
20	10	10	93.35	38.35367	5628
25	10	10	93.16	38.19903	5598
30	10	10	92.97	38.04421	5574
10	15	10	93.66	38.57491	5675
10	20	10	93.59	38.48726	5664
10	25	10	93.52	38.39949	5653
10	30	10	93.44	38.31160	5643
10	10	15	100.08	38.82380	6692
10	10	20	106.42	38.98890	7720
10	10	25	112.76	39.15788	8809
10	10	30	119.11	39.33087	10156

Figure 8 discussed the effect of steam temperature on dryness fraction of exhaust steam. With increase in steam temperature the dryness fraction of steam also increases, which is helpful in avoiding the erosion in final stages of turbine. Figure 9 shows the change in irreversibilities of steam turbine and exhaust of HRSG. The irreversibility of steam turbine decreases with increase in temperature, but the stack irreversibility increases with increase in steam temperature. To reduce the stack temperature the LP pressure must be low.

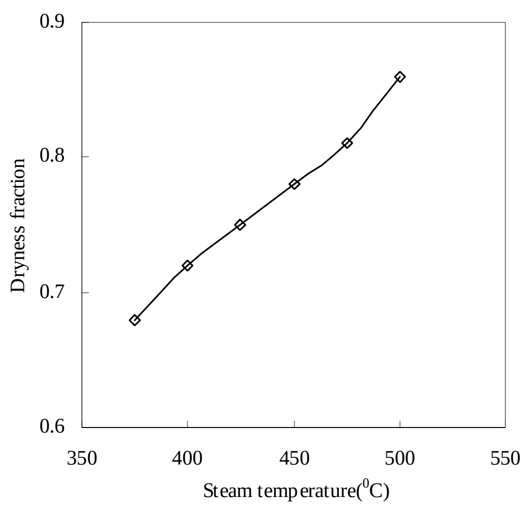

Fig. 8. Effect of steam temperature on dryness fraction.

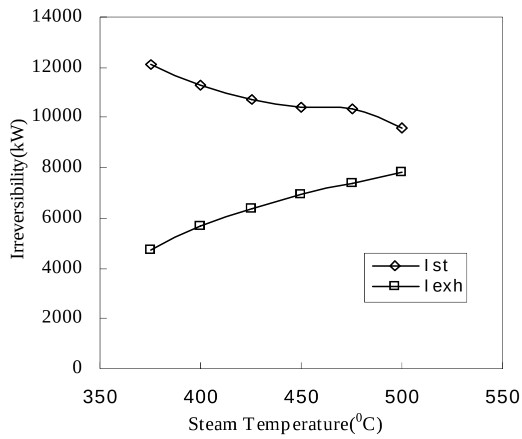

Fig. 9. Effect of steam temperature on Stack and steam turbine irreversibility.

Table 2 shows the effect of steam pressure on efficiency of the cycle, stack temperature and stack irreversibility. At higher pressures the change in stack temperature and stack irreversibility is small. For better exergetic utilization the high pressure must be as high as possible. Table 3 shows the change in work output and mass of flue gases with plant capacity. From the table it is clear that for any capacity of the plant one third of total power is produced by the steam turbine and the remaining two third power is produced by the gas turbine.

Table 2. Effect of Maximum Steam Pressure on Stack Temperature, Efficiency and Stack Irreversibility

Pressure (bar)	Efficiency	Stack	I stack
90	34.61	81.34	3912
110	35.75	92.57	5507
130	36.42	101.38	6923
150	36.92	103.25	7241
160	37.13	105.04	7551
170	37.32	106.61	7828

Table 3. Change in Steam Turbine Output with Plant Capacity for Fixed Steam Inlet Conditions at Optimum Conditions

Plant Capacity (MW)	Mass of flue gas (kg/s)	Work Output (kW)	Stack (K)	Efficiency
100	180.437	36699.081	379.6	37.327
200	360.874	73398.163	379.6	37.327
300	541.311	110097.240	379.6	37.327
400	721.748	146796.330	379.6	37.327
500	902.185	183495.410	379.6	37.327
600	1082.622	220194.490	379.6	37.327
700	1263.059	256893.570	379.6	37.327
800	1443.496	293592.650	379.6	37.327
900	1623.933	330291.730	379.6	37.327
1000	1804.370	366990.810	379.6	37.327
1100	1984.807	403689.900	379.6	37.327
1200	2165.244	440388.980	379.6	37.327

4. CONCLUSIONS

Thermodynamic analysis of triple pressure HRSG is done using energy and exergy analysis. Effect of various operating parameters on performance of HRSG is studied. It is observed that by increasing steam temperature the irreversibility of steam turbine decreases and dryness fraction of steam is increased. For maximum pressure of 170 bar the optimum IP and LP steam pressures are found as 47.6 bar and 9.8 bar. For given capacity of the combined cycle power plant 2/3 power is produced from gas turbine and 1/3 power is produced from the steam turbine.

NOMENCLATURE

C_p = specific heat, kJ/kg K
CV = calorific value, kJ/kg
fpl = fractional pressure loss
G = chemical exergy input
H = net calorific value
hl = heat loss
hp = high pressure, bar
lp = low pressure, bar
p = pressure, bar
P = Power output, MW
R = gas constant
r_p = pressure ratio
T = Temperature, K
W = work output, MW

Suffix

a = air
ap = approach point
c = compressor
g = gas
gt = gas turbine
pp = pinch point
s = steam
st = steam turbine

Greek Symbols

ς = efficiency
γ = specific heat ratio
ø = ratio of chemical exergy to net calorific value
Δ = differential

REFERENCES

[1] Linhoff B., Hindmarsh E. 1983. The pinch design method for heat exchanger networks. Che*mical Engineering Science* (38): 745-763.

[2] Subrahmanyam.N.V.R.SS, Rajaram.S, Kamalnathan.N. 1995. HRSGs for combined cycle power plants, *Heat Recovery Systems & CHP* 15(2):155-161.

[3] P.K.Nag and D.Raha.1995. Thermodynamic analysis of a coal-based combined cycle power plant, *Heat Recovery Systems &CHP*, 15(2): 115-129.

[4] Horlock, J.H. 1992. *Combined Power Plants* Oxford, Pergamon Press.

[5] A. Ong'iro, V. I. Ugursal, A. M. A1 Taweel, and J. D. Walker.1997. Modeling of heat recovery steam generator performance, *Applied Thermal Engineering*, 17(5):427-446.

[6] P.K.Nag, S. Mazumder,1991. Thermodynamic optimization of a waste heat boiler in: *Proceedings of the 11th National Heat and Mass Transfer Conference*, Madras, India.

[7] Dellenback, P.A.2002. Improved gas turbine efficiency through alternative regenerator configuration. *Trans ASME, Journal of Engineering for Gas turbines and power.* 124. 441-446.

[8] Ravi kumar.N. Rama krishna.K. Sita rama raju.A.V. 2005. Exergy Analysis of Gas Turbine Power Plant with Alternative Configuration of Regenerator. *Proceedings, 2nd International Exerg, Energy Environment Symposium(IEEES2)*. Kos-Greece.

[9] Ravi kumar.N., Sita rama raju.A.V. 2005. The Study of the Effects of Gas Turbine Inlet Cooling on Plant and HRSG Performance, *Proceedings of National Conference on Advances in Mechanical Engineering (AIM-2005)*.Hyderabad. 55-60.

[10] P. K. Nag and S. De. 1997. Design and operation of a heat recovery steam generator with minimum Irreversibility, *Applied Thermal Engineering*, 17(4)385-391.