THERMODYNAMIC ANALYSIS OF HEAT RECOVERY STEAM GENERATOR IN COMBINED CYCLE POWER PLANT

by

Naradasu RAVI KUMAR, Konijeti RAMA KRISHNA, and Alluru Venkata SITA RAMA RAJU

Original scientific paper
UDC: 536.255
BIBLID: 0354-9836, *11* (2007), 4, 143-156

Combined cycle power plants play an important role in the present energy sector. The main challenge in designing a combined cycle power plant is proper utilization of gas turbine exhaust heat in the steam cycle in order to achieve optimum steam turbine output. Most of the combined cycle developers focused on the gas turbine output and neglected the role of the heat recovery steam generator which strongly affects the overall performance of the combined cycle power plant. The present paper is aimed at optimal utilization of the flue gas recovery heat with different heat recovery steam generator configurations of single pressure and dual pressure. The combined cycle efficiency with different heat recovery steam generator configurations have been analysed parametrically by using first law and second law of thermodynamics. It is observed that in the dual cycle high pressure steam turbine pressure must be high and low pressure steam turbine pressure must be low for better heat recovery from heat recovery steam generator.

Key words: *combined cycle, gas turbine, steam cycle, dual cycle, waste heat recovery*

Introduction

The energy demand world wide especially in the developing countries is growing significantly as a result of economic growth, industrial expansion, high population growth, and urbanization. Thermal power plants play a major role in meeting this ever increasing demand. Selection of proper thermodynamic cycle plays a vital role in extraction of power from thermal power plants. The power cycles are investigated with an overall objective of providing high fuel conversion efficiency. The literature has often suggested combining two or more thermal cycles within a single power plant to achieve this. In all cases, the intention was to increase the efficiency over that of single cycles. Normally the cycles can be classified as a "topping" and "bottoming" cycle. The first cycle, to which most of the heat is supplied, is called the topping cycle.

The waste heat it produces is then utilized in a second cycle which operates at lower temperature level and therefore referred to as a bottoming cycle. Until now, only one combined cycle has found wide acceptance it is the combination of gas turbine/steam turbine power plant. This offers higher thermal efficiency as compared to the gas turbine based plant or steam turbine based plants in isolation. However the performance of the gas/steam combined cycle power plant depends upon the performance of topping and bottoming cycle.

Dallenback [1] proposed an alternative regenerator configuration to improve the efficiency of gas turbine cycle. Ravi Kumar *et al.* [2] has done energy analysis for the alternative regenerator configuration to find exergy losses in different gas turbine cycle 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 decreases. Generally with increase in turbine inlet temperature the specific work output of a gas turbine increases. But increase in turbine inlet temperature has strict metallurgical limitations in terms of maximum temperature that the turbine stage could withstand. Ravi Kumar and Sita Rama Raju [3] analyzed the effect of inlet cooling on heat recovery steam generator (HRSG) performance. It is found that the inlet cooling reduces the work input of the compressor and increases the mass flow rate of air. Similarly 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 in steam turbine is considered from metallurgical point of view of turbine blades. The mass flow rate of steam and steam temperature depends on the amount of heat available in the gas turbine exhaust. The HRSG forms the backbone of combined cycle power plants, providing the link between the gas turbine and the steam turbine. The performance of the HRSG strongly affects the overall performance of a combined cycle power plant. Ongiro *et al.* [4] developed a numerical method to predict the performance of the HRSG. P. K. Nag and S. De [5] designed a heat recovery steam generator with economiser and evaporater for minimum irreversibility. They concluded operating the HRSG at the maximum load reduces entropy generation. Ravi Kumar *et al.* [6] has done performance simulation of HRSG in combined cycle power plant. They discussed the effect of various parameters like pinch point, approach point, steam pressure, steam temperature, gas flow rate on the performance of the HRSG. C. Casarosa and A. Franco [7] done thermodynamic optimization of the operating parameters for heat recovery in combined cycle power plants to minimize thermal energy losses by taking into account only the irreversibility due to temperature difference between the hot and cold.

The aim of the present paper is to study the effect of HRSG configuration of both single pressure and dual pressure on combined cycle power plant efficiency. In addition we analyze the influences of various parameters like pinch point, approach point, steam temperature, steam pressure on the flue gas heat recovery by using energy and energy analysis.

Analysis

A standard combined cycle is considered for the present analysis [8]. The schematic layout of combined cycle is shown in fig. 1. Air after compression in the compressor enters the combustion chamber where its temperature is raised by the combustion of fuel. The gases then expand in the turbine and produce the work output part of which is

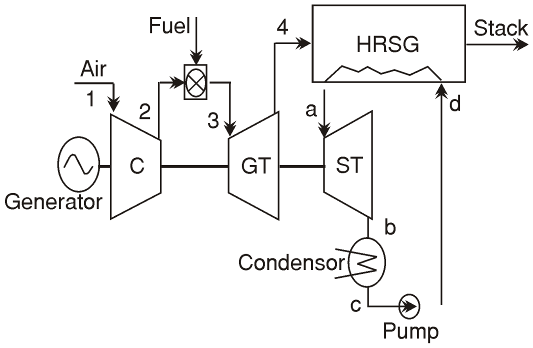

Figure 1. Schematic of combined cycle power plant

supplied to run the compressor. The heat carried by the exhaust gases is recovered in the HRSG to generate steam for expansion in steam turbine.

The following assumptions are made:
(1) the system is in steady-state,
(2) there is no pressure drop either on water or steam side,
(3) the specific heats of exhaust gas and water are constant, and
(4) the pressure drop on the gas side has no significant effect on its temperature.

Further the following operating parameters are assumed for the proposed power plant: maximum high pressure (HP) steam superheat temperature is 500 ºC, minimum pinch point temperature difference 15 ºC, minimum stack temperature100 ºC, maximum steam pressure 200 bar, efficiency of compressor 89%, efficiency of turbine 87%, heat loss factor 0.02, fractional pressure loss in combustion chamber (fpl) 0.03, calorific value of the fuel 48990 kJ/kg, and ratio of chemical energy to net calorific value ψ 1.04.

Specific heat of air is given by [9]:

$$C_{\mathrm{p_{air}}}(T) \quad \frac{28.11 \quad 0.1967 \;\; 10^{\;2}T \quad 0.4802 \;\; 10^{\;5}T^{2} \quad 1.966 \;\; 10^{\;9}T^{3}]}{29}$$

Specific heat of flue gas is given by [9]:

$$C_{\mathrm{p_{gas}}}(T) \quad 1.8083 \quad 2.3127 \;\; 10^{\;3}T \quad 4.045 \;\; 10^{\;6}T^{2} \quad 1.7363T^{3}$$

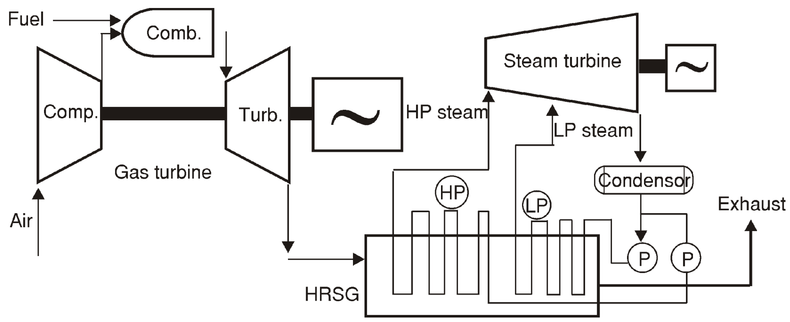

Figure 2. Schematic of dual pressure HRSG

The schematic of combined cycle power plant with dual pressure HRSG is shown in fig. 2. Gas turbine exhaust gases are passed through HRSG to covert the feed water into steam. Feed water is supplied to HRSG at two different pressures which absorbs heat of flue gases and enters the steam turbine as superheated steam. The condensate is circulated as in closed cycle steam power plant. The exhaust gases after passing through HRSG are discharged at atmospheric pressure through stack.

The compressor outlet pressure and temperature are:

$$p_2 = r_\mathrm{p}\, p_1 \tag{1}$$

$$T_{2\mathrm{s}} = T_1 (r_\mathrm{p})^{\frac{\gamma_\mathrm{a} - 1}{\gamma_\mathrm{a}}} \tag{2}$$

The actual temperature of air at the end of compression is given by:

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

Hence work required for compression is obtained by using:

$$W_\mathrm{C} = \int_1^2 C_\mathrm{pa}\, \mathrm{d}T \tag{4}$$

Similarly the pressure after the combustion chamber and temperature after expansion are:

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

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

$$T_4 = T_3 - \eta_\mathrm{T}(T_3 - T_{4\mathrm{s}}) \tag{7}$$

The work output from the gas turbine is:

$$W_\mathrm{T} = \int_4^3 C_\mathrm{pg}\, \mathrm{d}T \tag{8}$$

Hence net work output is:

$$W_\mathrm{net} = W_\mathrm{T} - W_\mathrm{C} \tag{9}$$

and the gas turbine cycle efficiency is:

$$\eta_\mathrm{gt} = \frac{W_\mathrm{net}}{m_\mathrm{f} CV} \tag{10}$$

The process of heat exchange in dual pressure HRSG is explained using fig. 3. The flue gases enter at temperature T_4 and passes through different sections in HRSG and

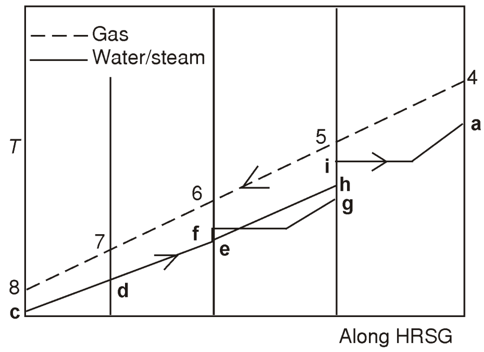

Figure 3. Temperature heat diagram of dual pressure HRSG

leaves at T_8 .The feed water is heated in condensate preheater section of HRSG from T_c to T_d, and enters the economiser section. The LP feed water is heated from T_d to T_e and HP feed water is heated from T_d to T_h in the economiser section. At point e and h the water temperature is just below the saturation temperature to the corresponding pressures. It is the fixed temperature difference that should be considered when designing HRSG. The temperature difference between e to f is called low pressure (LP) approach point and h to i is HP approach point. The LP and HP feed waters enters the evaporator section at pinch points f and i, respectively. The points g and a are final superheated temperature attained by LP and HP steam, respectively.

Let T_i – high pressure saturation temperature, and T_f – low pressure saturation temperature

$$T_5 = T_i + hp_{pp} \tag{11}$$

$$T_6 = T_f + lp_{pp} \tag{12}$$

$$T_h = T_i - hp_{ap} \tag{13}$$

$$T_e = T_f - lp_{ap} \tag{14}$$

where T_h is the temperature of feed water leaving the high pressure economiser, T_f – the temperature of feed water leaving the low pressure economiser, hp_{pp} – the high pressure pinch point, lp_{pp} – the low pressure pinch point, hp_{ap} – the high pressure approach point, and lp_{ap} – the low pressure approach point.

Using energy balance at high pressure evaporator and superheater section (i-a) the mass of high pressure steam generated can be obtained as:

$$m_{hp} = \frac{\int_5^4 m_g C_{pg} \mathrm{d}T(1 - hl)}{\int_h^a \mathrm{d}h} \tag{15}$$

Similarly by taking energy balance at low pressure evaporater and superheater section (f-g) and HP economiser section (e-h) the mass of low pressure steam generated is obtained as:

$$\int_e^g m_{lp}\mathrm{d}h + \int_e^h m_{hp}\mathrm{d}h = \int_6^5 m_g C_{pg}\,\mathrm{d}T(1 - hl) \tag{16}$$

$$m_{\mathrm{lp}} = \frac{\int_6^5 m_{\mathrm{g}} C_{\mathrm{pg}} \mathrm{d}T (1 - hl) - m_{\mathrm{hp}} (h_{\mathrm{h}} - h_{\mathrm{e}})}{h_{\mathrm{g}} - h_{\mathrm{e}}} \tag{17}$$

Now by considering energy balance around condensate preheater and LP economiser outlet as shown below:

$$\int_8^6 m_{\mathrm{g}} C_{\mathrm{pg}} \mathrm{d}T (1 - hl) = \int_e^{c_c} (m_{\mathrm{hp}} + m_{\mathrm{lp}}) \mathrm{d}h \tag{18}$$

From the above equation we can obtain the stack temperature T_8.
The efficiency of the steam turbine is thus obtained as:

$$\eta_{\mathrm{st}} = \frac{W_{\mathrm{st}}}{\int_c^g m_{\mathrm{lp}} \mathrm{d}h + \int_c^a m_{\mathrm{hp}} \mathrm{d}h} \tag{19}$$

The combined efficiency of the cycle is obtained by using:

$$\eta_{\mathrm{I}} = \frac{W_{\mathrm{gt}} + W_{\mathrm{st}}}{m_{\mathrm{f}} CV} \tag{20}$$

Energy analysis is useful in thermal power plants to account for the variable quality of different disordered energy forms. Exergy is nothing but the work that is available in the material stream as a result of the non-equilibrium conditions relative to the reference condition, 1bar and 25 °C.

$$\Delta H_0 = m_{\mathrm{f}}\, CV \tag{21}$$

$$\Delta G_0 = \varphi \Delta H_0 \tag{22}$$

where ΔH_0 is net calorific value, ΔG_0 is chemical exergy, φ is the ratio of chemical exergy to net heating value and T_0 is the environmental temperature.

$$I = T_0 \Delta s_0 = \Delta G_0 - \Delta H_0 \tag{23}$$

Hence the second law efficiency of the combined cycle is given by:

$$\eta_{\mathrm{II}} = \frac{W_{\mathrm{gt}} + W_{\mathrm{st}}}{\Delta G_0} \tag{24}$$

The effectiveness of HRSG is obtained using:

$$\varepsilon_{\mathrm{HRSG}} = \frac{m_{\mathrm{hp}} (h_{\mathrm{a}} - h_{\mathrm{c}}) + m_{\mathrm{lp}} (h_{\mathrm{g}} - h_{\mathrm{c}})}{m_{\mathrm{g}} C_{\mathrm{pg}} (T_4 - T_8)} \tag{25}$$

Solution procedure

For the given pressure ratio the outlet pressure is calculated by using eq. (1). With help of eqs. (2) to (10) compressor work input, gas turbine output, and gas turbine efficiency are calculated. Temperature of flue gas at different sections is calculated by using eqs. (11) and (12). Mass flow of HP steam and LP steam is calculated using eq. (15) to (17). The efficiency of steam turbine and combined cycle efficiency is obtained from eqs. (19) and (20). Second law efficiency and effectiveness of HRSG is calculated by using eqs. (24) and (25).

Results and discussion

In order to establish a systematic comparison between single pressure and dual pressure HRSG performance of the plant is examined for same operating conditions. The effect of gas cycle operating parameters (pressure ratio, gas turbine inlet temperature, mass flow rate of gases) and steam cycle parameters (steam pressure, steam temperature, pinch point and approach point) are studied on the performance of single pressure and dual pressure HRSG. Figures 4-7 shows the effect of pressure ratio and maximum gas temperature on combined cycle efficiency and stack temperature when single pressure and dual pressure HRSGs are used.

Figure 4 shows the effect of pressure ratio on combined cycle efficiency and stack temperature. The combined cycle efficiency decreases with increase in pressure ratio. Compared with single pressure HRSG in dual pressure HRSG the cycle efficiency is more because the waste heat is utilized in generation of steam at low pressure. Further increase in pressure ratio for a fixed maximum inlet temperature to turbine decreases the turbine exhaust temperature which reduces effects the mass of steam generated and decreases the combined cycle efficiency. The stack temperature also increases with increase in pressure ratio causing more exergy loss through exhaust. The effect is very small in dual pressure cycle compared with single pressure cycle because of more recovery from gas turbine exhaust.

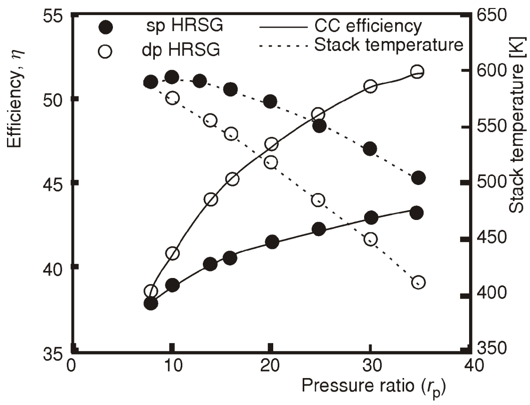

Figure 4. Effect of pressure ratio on stack temperature and combined cycle efficiency

Figure 5 shows the effect of turbine inlet temperature on stack temperature and effectiveness of HRSG. With increase in turbine inlet temperature the turbine exhaust temperature also increases for fixed pressure ratio. For a fixed steam inlet conditions with increase in HRSG inlet temper-

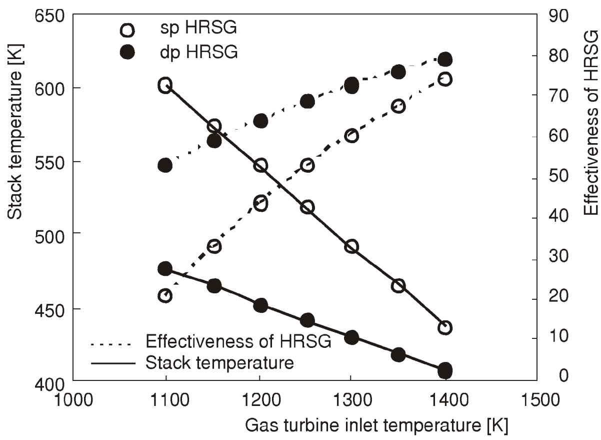

Figure 5. Effect of gas turbine inlet temperature on stack temperature and HRSG effectiveness

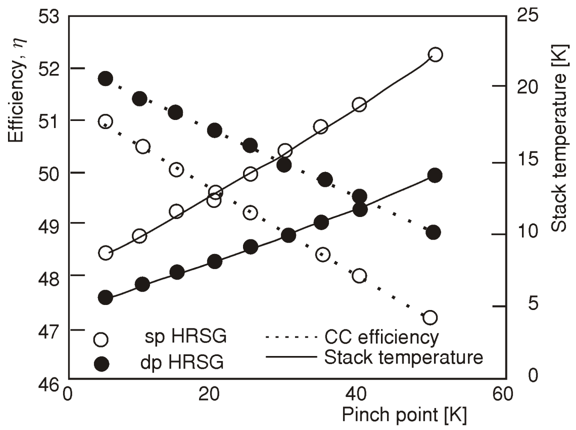

Figure 6. Effect of pinch point on combined cycle efficiency and stack irreversibility

ature the mass flow rate of HP and LP steam increases. This causes high utilization of waste heat in HRSG and decreases the stack temperature. Compared with single pressure HRSG the effectiveness of HRSG is more in dual cycle. This is because of further utilization of waste heat in the LP evaporator of dual cycle.

Figure 6 represents the efficiency of combined cycle and stack irreversibility with change in pinch point. The pinch point is an important parameter in the optimization of the steam cycle. It affects the amount of steam generated in the cycle. In dual pressure HRSG the pinch point of high pressure evaporator is less important than a single pressure system because the heat that is not utilized is recovered in low pressure evaporator. The loss in power output is only to the difference in exergy between the high pressure and low pressure steam portions. In a dual pressure system, the pinch points of both the high pressure and the low pressure evaporators have less effect on the efficiency of the steam process than with single pressure system. The stack irreversibility increases with increase in pinch point. Compared with dual pressure HRSG in single pressure the stack irreversibility is very high which shows large amount of heat is gone to atmosphere unutilized.

Table 1 shows the effect of pinch point on irreversibility of HRSG, steam turbine, and stack in dual pressure cycle. With increase in pinch point the stack irreversibility increases which indicates large amount of heat is gone to atmosphere with out recovery in HRSG. The irreversibility of steam turbine is decreased because of low mass flow rate of steam generation in HRSG.

Table 1. Irreversibility of major components in dual pressure HRSG at various pinch points

Pinch point [K]	Irreversibility [MW]		
	HRSG	Steam turbine	Stack
5	10.428	8.905	5.719
10	11.136	8.769	6.489
15	11.804	8.633	7.298
20	12.435	8.496	8.145
25	13.029	8.361	9.029
30	13.587	8.224	9.948
35	14.112	8.088	10.902
40	14.602	7.952	11.888
50	15.487	7.679	13.958

Figure 7. shows the effects of approach point on steam turbine output and stack temperature. As the approach point increases the steam turbine output decreases. Compared with single pressure HRSG in dual pressure the drop in output of steam turbine is less. The stack temperature also remains almost constant in dual cycle with increase in approach point, which shows high utilization of waste heat. Hence the variation in approach point for dp HRSG has minimal effect on steam turbine output and stack temperature. The change in approach point will not affect the irreversibility of steam turbine, HRSG and stack irreversibility considerably.

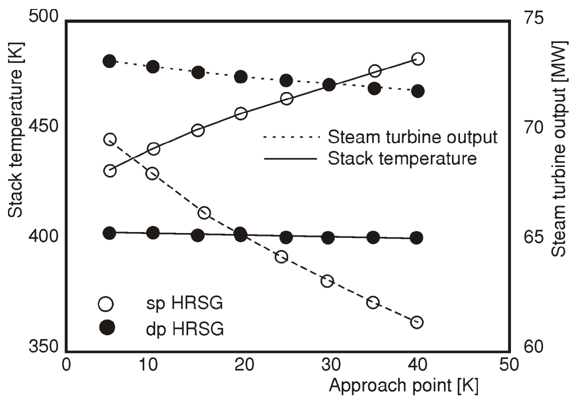

Figure 7. Change in stack temperature and steam turbine output with approach point

Figure 8 shows the variation of second law efficiency with steam pressure. 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. It is advisable raising the steam pressure above the thermodynamic equilibrium, because it

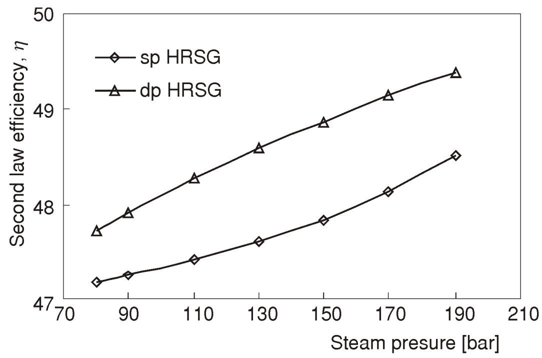

Figure 8. Effect of maximum cycle pressure on second law efficiency

causes a reduction in exhaust steam flow for a given power output thus reducing the size of condensor and cooling water requirement. For a given heat input there is an increase in steam turbine output and combined cycle efficiency because of large mass flow rate of steam. In dual pressure cycle the high pressure steam pressure must be relatively high to attain good exergetic utilization of the waste heat by generating high quality of steam. Hence the LP steam pressure must be low and HP steam pressure must be high to attain good energetic utilization of the waste heat. The effect of LP pressure on steam flow rate is with increase in LP steam pressure mass flow rate of steam decreases and hence steam turbine output is decreased.

The variation of steam cycle efficiency and HRSG effectiveness with pinch point is shown graphically in fig. 9. With increase in pinch point the mass of steam generation decreases and results low output from the steam turbine. The effectiveness of HRSG also decreases due to large temperature difference between steam and gas.

The utilization of waste heat with steam temperature in HRSG is presented in fig.10. As the steam inlet temperature increases the effectiveness of HRSG exponentially drops due to more utilization of available heat in HP section. With decreasing steam temperature the recovery of waste heat is 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.

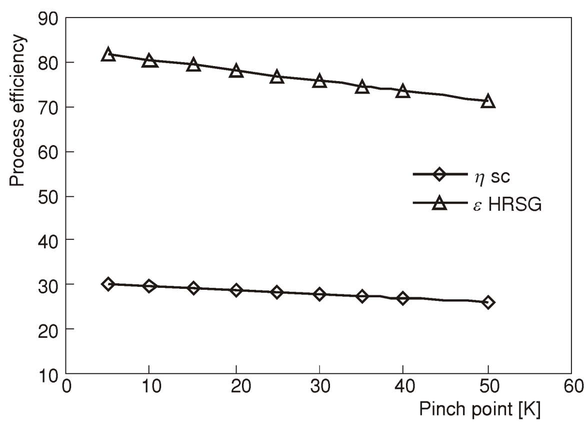

Figure 9. Effect of pinch point on efficiency of steam process and effectiveness of HRSG in dual cycle

In dual pressure HRSG while selecting the 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 on the turbine blades. 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 11 represents the variation of mass of steam with HRSG inlet temperature. As gas inlet temperature increases the mass of steam increases because of large available of heat. A reduction in the exhaust gas temperature lowers the efficiency of the steam process, this reduction, however, is less pronounced in dual cycle than single pressure system, because the energy utilization rate does not drop off quickly.

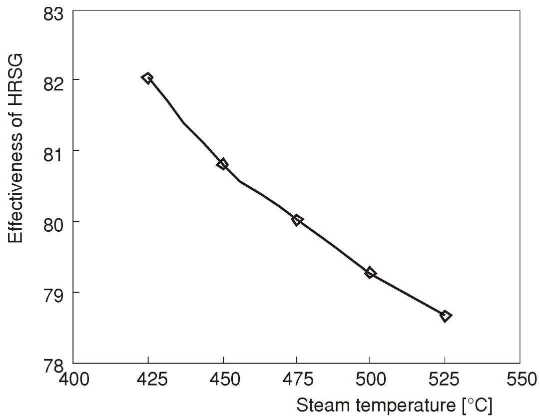

Figure 10. Effect of live steam temperature on rate of waste heat energy utilization

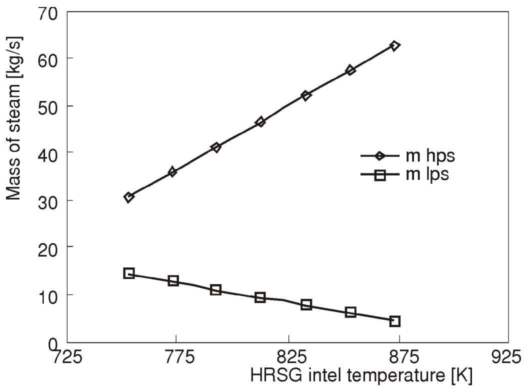

Figure 11. Mass of steam with HRSG inlet temperature

Conclusions

Comparison of various parameters like pressure ratio, steam pressure, steam temperature, pinch point, approach point on the performance of combined cycle power plant with single pressure and dual pressure HRSG is done. It is observed that:

(1) Dual pressure HRSG offers better efficiency and less irreversibility with compared to single pressure HRSG.

(2) In the dual cycle the high pressure steam pressure must be relatively high and low pressure steam pressure must be low for better heat recovery from HRSG. However the pressure in the LP evaporator should not drop below 3 bar causing increased flow of steam requiring large cross-section areas.

(3) While selecting the steam temperature in the dual cycle care should be taken in fixing the LP steam temperature such that the difference between HP steam temperature after expansion and LP steam at mixing point in the turbine is not too high.

(4) The pinch point effects greatly in single pressure HRSG, but its effect on dual cycle is less because the heat that is not utilized in HP section is recovered in the LP evaporator.
(5) A reduction in the exhaust gas temperature lowers the efficiency of the steam turbine output. But this reduction is less pronounced in dual cycle compared with single cycle.

Nomenclature

C_p – specific heat, [$kJkg^{-1}K^{-1}$]
CV – calorific value, [$kJkg^{-1}$]
dp – dual pressure, [–]
fpl – fractional pressure loss
G – chemical exergy input, [$kJkg^{-1}$]
H – net calorific value, [$kJkg^{-1}$]
hl – heat loss, 2%
HP – high pressure, [bar]
I – Irreversibility, [W]
LP – low pressure, [bar]
m – mass flow rate, [kgs^{-1}]
p – pressure, [bar]
P – power output, [W]
R – gas constant, 0.287 [$kJkg^{-1}K^{-1}$]
r_p – pressure ratio, [–]
s – specific entropy, [$kJkg^{-1}K^{-1}$]
sp – single pressure, [–]
T – temperature, [K]
W – work output, [W]

Greek letters

γ – specific heat ratio, [–]
Δ – differential
ε – effectiveness, [–]
η – efficiency, [–]
ψ – ratio of chemical exergy to net calorific value, [–]

Subscripts

a – air
ap – approach point
c – compressor
cc – combined cycle
f – fuel
g – gas
gt – gas turbine
pp – pinch point

s – isentropic condition
st – steam turbine
1, 2, 3 – state points
0 – environmental state

References

[1] Dellenback, P. A., Improved Gas Turbine Efficiency through Alternative Regenerator Configuration, *Transaction ASME*, *Journal of Engineering for Gas Turbines and Power, 124* (2002), 3, pp. 441-446

[2] Ravi Kumar, N., Rama Krishna, K., Sita Rama Raju, A. V., Exergy Analysis of Gas Turbine Power Plant with Alternative Configuration of Regenerator, *Proceedings on CD,* 2nd International Exergy Energy Environment Symposium (IEEES2), Kos, Greece, 2005, VI-13

[3] Ravi Kumar, N., Sita Rama Raju, A. V., The Study of the Effects of Gas Turbine Inlet Cooling on Plant and HRSG Performance, *Proceedings*, National Conference on Advances in Mechanical Engineering (AIM-2005), Hyderabad, India, 2005, pp. 55-60

[4] Ong'iro, A., Ugursal, V. I., A1 Taweel, A. M., Walker, J. D., Modeling of Heat Recovery Steam Generator Performance, *Applied Thermal Engineering, 17* (1997), 5, pp. 427-446

[5] Nag, P. K, De, S., Design and Operation of a Heat Recovery Steam Generator with Minimum Irreversibility, *Applied Thermal Engineering, 17* (1997), 4, pp. 385-391

[6] Ravi Kumar, N., Rama Krishna, K, Sita Rama Raju, A. V, Performance Simulation of Heat Recovery Steam Generator in Combined Cycle Power Plant, *Proceedings* (Eds. S. C. Mishra, B. V. S. S. S., Prasad, S. V., Garimella), 18th National & 7th ISHMT-ASME Heat and Mass Transfer Conference (HMT-2006), IIT Guwahati, India, 2006, pp.1781-1787

[7] Casarosa, C., Franco, A., Thermodynamic Optimization of the Operative Parameters for the Heat Recovery in Combined Power Plants, *Int. J. Applied Thermodynamics, 4* (2001), 1, pp. 43-52

[8] Horlock, J. H., Combined Power Plants, Pergamon Press, Oxford, UK, 1992

[9] Yadav, J. P., Singh, O., Thermodynamic Analysis of Air Cooled Simple Gas and Steam Combined Cycle Plant, Journal of Institution of Engineers (India) – Mechanical Engineering, 2006, Vol. 86, pp. 217-222

Authors' addresses:

N. Ravi Kumar
Department of Mechanical Engineering, MVGR College of Engineering
Vizianagaram (po)-535005
Andhra Pradesh
India

K. Rama Krishna
Freshman Engineering Department, K. L. College of Engineering
Green fields, Guntur
Andhra Pradesh
India

A. V. Sita Rama Raju
Department of Mechanical Engineering, JNTU College of Engineering
Kakinada- 533003
Andhra Pradesh
India

Corresponding author N. Ravi Kumar
E-mail: ravi_naradasu@yahoo.com

Paper submitted: September 10, 2006
Paper revised: October 19, 2007
Paper accepted: November 12, 2007