Thermodynamic modeling and optimization of multi-pressure heat recovery steam generator in combined power cycle

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Optimum configuration for single pressure (SP), dual pressure (DP) and triple pressure (TP) heat recovery steam generator (HRSG) is presented to improve heat recovery and thereby exergy efficiency of combined cycle. Deaerator was added to enhance efficiency and remove dissolved gases in feed water. A new method was introduced to evaluate low pressure (LP) and intermediate pressure (IP) in HRSG from local flue gas temperature to get minimum possible temperature difference in heaters instead of a usual fixation of pressures. Optimum location for deaerator was found at 1, 3 and 5 bar respectively for SP, DP and TP in heat recovery at a high pressure (HP) of 200 bar. Results also showed optimum pressure for air compression and steam reheater by means of three categories of heat recovery.

Key words: Combined cycle, Deaerator, Exergy analysis, Heat recovery, Multi pressure, Steam generator

Introduction

Combined cycle (CC) power plants are gaining wider acceptance due to more and more availability of natural gas, higher overall thermal efficiencies, peaking and two-shifting capabilities, fast start-up capabilities and lesser cooling water requirements. Optimization of heat recovery steam generator (HRSG) is particularly interesting for combined plants design in order to maximize work obtained in vapor cycle. Pasha & Sanjeev presented a discussion about parameters that influence type of circulation and selection for HRSG. Ongiro et al developed a numerical method to predict performance of HRSG for design and operation constraints. Ganapathy et al described features of HRSG used in Cheng cycle system, where a large quantity of steam is injected into a gas turbine to increase electrical power output. Subrahmanyam discussed factors affecting HRSG design for achieving highest CC efficiency with cheaper, economical and competitive designs. Noelle & Heyen designed once-through HRSG, which is ideally matched to very high temperature and pressure, well into supercritical range.

Ragland & Stenzel compared four plant designs using natural gas with a view of cost benefits achieved through HRSG optimization. Casarosa et al determined operating parameters means both of a thermodynamic and of a thermoeconomic analysis, minimizing a suitable objective function by analytical or numerical mathematical methods.

Optimal design and operation of a HRSG is possible with minimization of entropy generation. Reddy et al applied second law analysis for a waste HRSG, which consists of an economizer, an evaporator and a super heater. Multi pressure steam generation in HRSG of a combined power plant improves performance of plant. Pelster et al compared results of a reference CC with dual and triple pressure HRSGs and also with and without steam reheating models. Bassily modeled a dual and triple pressure reheat CC with a preset in constraints on minimum temperature difference for pinch points, temperature difference for superheat approach, steam turbine inlet temperature and pressure, stack temperature, and dryness fraction at steam turbine outlet without a deaerator in steam bottoming cycle.

This study presents optimization of single pressure (SP), dual pressure (DP) and triple pressure (TP) of
HRSG in CC with a deaerator. Variations in CC exergy efficiency are plotted with compressor pressure ratio, gas turbine inlet temperature, high pressure (HP) in HRSG, steam reheat pressure, deaerator pressure and pinch point (PP). Optimized pressures for combustion, HP steam, steam reheater and deaerator for each of HRSG configuration are presented.

Materials and Methods
Thermodynamic Model of Combined Cycle with SP, DP and TP HRSG

Optimized configuration in HRSG improves steam generation rate and hence steam turbine output. Simple gas cycle gives higher efficiency at low compressor ratio compared to intercooled-reheat gas cycle, thereby gas cycle gives higher efficiency at low compressor pressure level is depicted in Fig. 2. In each configuration with SP, DP and TP level configurations in HRSG as a bottoming cycle in CC, which is modeled separately for each of pinch point (PP). Optimized pressures for combustion, HRSG, steam reheat pressure, deaerator pressure and gas turbine inlet temperature, high pressure (HP) in efficiency are plotted with compressor pressure ratio, (Fig. 1).

Temperature-heat transferred diagram for each pressure level is depicted in Fig. 2. In each configuration of HRSG, a deaerator is located to gain higher efficiency as well as to remove dissolved gases in feed water. Condensate preheater (CPH) is located in HRSG as the last heat transfer surface to improve heat recovery. In this work, steam turbine inlet temperatures, low pressure (LP) and intermediate pressure (IP) are not fixed as in a regular manner. Steam temperature and pressures are determined with local flue gas temperature in heaters. Pinch point (PP)-minimum temperature difference between gas turbine exhaust leaving evaporator and saturation temperature of steam in evaporators and terminal temperature difference (TTD)-temperature difference between exhaust entry and super heated steam in super heaters are maintained constant. Zero approach point (temperature difference between saturation temperature of steam and incoming water temperature) is assumed in economizer.

Thermodynamic Analysis of Combined Cycle
Assumptions for analysis of CC are tabulated (Table 1). Energy efficiency of CC is determined based on lower heating value (50, 145 kJ/kg) of fuel. Net work output (%) of CC of standard chemical exergy of fuel (52, 275 kJ/kg) is expressed as exergy efficiency of CC. For an isentropic compression process, entropy change for air is zero, $\Delta s = 0$.

$$s_{o_{31}} - s_{o_{30}} + 3.76(s_{N_{i_{31}}} - s_{N_{i_{30}}}) - R \left[\log \left(\frac{P_u}{P_{30}}\right) + 3.76 \log \left(\frac{P_u}{P_{30}}\right)\right] = 0$$

... (1)

Temperature of air after isentropic compression is estimated from iteration of Eq. (1). Actual temperature of compressed air is determined from compressor isentropic efficiency. Combustion equation in gas turbine combustion chamber is

$$CH_4 + x (O_2 + 3.76 N_2) \rightarrow CO_2 + 2 H_2O$$

$$+ (x - 2)O_2 + 3.76x N_2 \quad \ldots (2)$$

In Eq. (2), $x$ is amount of air to be supplied. Air supply is determined by energy balance of Eq. (2) to get required combustion temperature. $\Delta s$ for isentropic expansion in gas turbine is

$$\Delta s = 0 = s_{j2} - s_{j3}. \quad \ldots (3)$$

where, $s_{j2} = [s_{CO_2} + 2 s_{H_2O} + (x - 2) s_{O_2}$$

$$\quad + 3.76 x s_{N_2}]_{j2} - R [log (P_{j2}/m_{pce})$$

$$\quad + 2 \log(2 P_{j2}/m_{pce}) + 3.76x \log (3.76x P_{j2}/m_{pce})$$

$$\quad + (x - 2) \log ((x - 2) P_{j2}/m_{pce})] \quad \ldots (4)$$

and $s_{j32} = [s_{CO_2} + 2 s_{H_2O} + (x - 2) s_{O_2} + 3.76 xs_{N_2}]_{j32}$

$$\quad - R [log (P_{j3}/m_{pce})$$

$$\quad + 2 \log(2 P_{j3}/m_{pce}) + 3.76x \log (3.76x P_{j3}/m_{pce})$$

$$\quad + (x - 2) \log ((x - 2) P_{j3}/m_{pce})] \quad \ldots (5)$$

where $m_{pce}$ (kg mol) is total mass of chemical elements in products of combustion.

Gas temperature after expansion in gas turbine is determined with iteration of entropy in Eq. (3). Actual temperature of expanded gas is obtained from gas turbine isentropic efficiency. IP and LP are evaluated from saturation temperatures. Steam flow rates and local exhaust temperature in heating devices are determined from heat balance equations. Saturation temperature of IP evaporator is

$$T_{IP sat} = T_{IP ex out} - TTD_{IP} - DSH_{IP} \quad \ldots (6)$$
Fig. 1—Schematic representation of combined cycle with: a) Single pressure HRSG; b) Dual pressure HRSG; and c) Triple pressure HRSG (HP, high pressure; IP, intermediate pressure; LP, low pressure; HT, high temperature; LT, low temperature; GT, gas turbine; ST, steam turbine; RH, reheater; SH, superheater; EVAP, evaporator; ECO, economizer; CPH, condensate preheater; FP, feed pump; HRSG, heat recovery steam generator; CEP, condensate extraction pump)
Saturation temperature of LP evaporator is

$$T_{LP \text{ sat}} = T_{LP \text{ ex out}} - TTD_{LP} - DSH_{LP} \quad \ldots(7)$$

At HP evaporator, outlet temperature of flue gas

$$= T_{HP \text{ sat}} + PP_{HP} \quad \ldots(8)$$

At IP evaporator, outlet temperature of flue gas

$$= T_{IP \text{ sat}} + PP_{IP} \quad \ldots(9)$$

At LP evaporator, outlet temperature of flue gas

$$= T_{LP \text{ sat}} + PP_{LP} \quad \ldots(10)$$

Work outputs and inputs in gas and steam cycles are related to unitary mass flow of fuel.

Total net work output by CC, $w_{net, cc} = w_{net, gc} + w_{net, sc} \quad \ldots(11)$

To determine exergetic losses, irreversibilities associated in all components have to be estimated for exergy analysis. In this analysis, efficiency and losses are determined for 1 kg mol of fuel. Chemical exergy and physical exergy can be determined at each state as

Chemical exergy, $e_{ch}$

$$= \sum n_i e_k + RT_0 \sum n_i \ln \left( \frac{P}{P_0} \right) x_k \quad \ldots(12)$$

where $x_k$ is mole fraction of $k^{th}$ component.
Physical exergy = \( e_{ph} = h - \sum k T_0 s_k \) ...(13)

Exergy, \( e = e_{ch} + e_{ph} \) ...(14)

Exergetic loss (irreversibility) in compressor,
\[ i_c = e_{30} + w_c \cdot e_{31} \] ...(15)

Exergetic loss in gas turbine combustion chamber,
\[ i_{gtcc} = e_{CH4} + e_{31} \cdot e_{32} \] ...(16)

Exergetic loss in gas turbine, \( i_{gt} = e_{32} \cdot e_{33} \cdot w_{gt} \) ...(17)

Exergetic loss in HRSG, \( i_{HRSG} \)
\[ = (e_{HRSG in} - e_{HRSG out}) + \sum (e_{water in} - e_{steam out}) \] ...(18)

Exergetic loss in exhaust from HRSG, \( i_{ex} = e_{ex} \) ...(19)

Exergetic loss in steam turbines, \( i_{st} = T_0 \sum m(s_{out} - s_{in}) \) ...(20)

Total exergetic loss of combined cycle,

\[ i_{cov} = T_0 \left( m_{cov, st} (s_{out} - s_{in}) + m_w \times 4.18 \times \log \left( \frac{T_{w, out}}{T_{w, in}} \right) \right) \] ...(21)

Exergetic loss of hot water from condenser,
\[ i_{w, wc} = m_w \times 4.18 \times (T_{w, out} - T_{w, in}) - T_0 m_w \times 4.18 \times \log \left( \frac{T_{w, out}}{T_{w, in}} \right) \] ...(22)

Total (internal + external) exergetic loss in condenser,
\[ i_{cov} = i_{cov} + i_w \] ...(23)

Exergy loss in deaerator,
\[ i_{de} = T_0 \left( m_{dea} (s_{after mix} - s_{before mix}) + m_{feed} (s_{after mix} - s_{before mix}) \right) \] ...(24)

Table 1—Assumptions made for thermodynamic evaluation of HRSG in combined cycles

<table>
<thead>
<tr>
<th>Assumption</th>
<th>Condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>Atmospheric condition</td>
<td>25°C and 1.01325 bar</td>
</tr>
<tr>
<td>Gas cycle pressure ratio and maximum temperature</td>
<td>12 and 1200°C</td>
</tr>
<tr>
<td>Inlet pressure for HP steam turbine</td>
<td>200 bar</td>
</tr>
<tr>
<td>Terminal temperature difference (TTD) - temperature difference between flue gas and reheated/superheated steam</td>
<td>25</td>
</tr>
<tr>
<td>Condenser pressure</td>
<td>0.05 bar</td>
</tr>
<tr>
<td>Steam reheat pressure</td>
<td>50 % of HP pressure</td>
</tr>
<tr>
<td>Pinch Point (PP) - minimum temperature differences between flue gas and steam in evaporators of HRSG</td>
<td>25</td>
</tr>
<tr>
<td>Degree of superheat (DSH) in superheater</td>
<td>50</td>
</tr>
<tr>
<td>Isentropic efficiency of gas turbine</td>
<td>90 %</td>
</tr>
<tr>
<td>Isentropic efficiencies of compressor and steam turbine</td>
<td>85 %</td>
</tr>
<tr>
<td>Pressure drop in combustion chamber</td>
<td>5 % of combustion chamber pressure</td>
</tr>
<tr>
<td>Pressure drop in HRSG, deaerator and condenser is neglected; Heat loss in HRSG, turbines, condenser, and deaerator is neglected</td>
<td></td>
</tr>
</tbody>
</table>
Fig. 3-Exergy efficiency of combined cycle with different configurations (SP, DP and TP) of HRSGs versus: (a) Compressor pressure ratio; (b) Gas turbine inlet temperature; (c) HP pressure; (d) Steam reheater pressure; (e) Deaerator pressure; and (f) Pinch point

\[ i_{\text{tot}} = i_c + i_{\text{gcc}} + i_{g_t} + i_{HRSG} + i_{es} + i_{st} + i_{\text{cot}} + i_{de} \ldots (25) \]

Exergy efficiency of combined cycle,

\[ \eta_{cc} = \left( \frac{W_{net \ cc}}{\epsilon^{CH4}} \right) \times 100 \quad \ldots (26) \]

**Results and Discussion**

CC with SP, DP and TP HRSG has been studied parametrically to find maximum obtainable exergy efficiency from CC. Deaerator pressure, at which steam is separated from steam turbine to heat feed water, is a key parameter to get optimum heat recovery from
Effects of topping cycle parameters (compressor pressure ratio and gas turbine inlet temperature) and bottoming cycle parameters (HP pressure, steam reheater pressure, deaerator pressure and PP in HRSG) on exergy efficiency of CC are plotted to identify maximum efficiency.

Optimum pressure ratio increases with increase in pressure level in HRSG (Fig. 3a). Gas turbine outlet temperature decreases with increase in pressure ratio. Therefore, efficiency of cycle decreases with further increase in pressure ratio from optimum value. Heat recovery improves from SP to TP configuration and hence efficiency of CC also rises with single to multi pressure effect. Optimum pressure ratio with SP, DP and TP HRSGs is 8, 10 and 12 respectively at gas turbine inlet temperature of 1200°C. At a fixed compressor ratio of 12, for SP, DP and TP configurations, heat recovery and exergy efficiency of CC increases with increase in temperature (Fig. 3b). But, at high temperature, there is a less increment in efficiency with multi pressure HRSG compared to at low temperature.

Efficiency of cycle increases with increase in steam turbine inlet pressure for all configurations in HRSG (Fig. 3c). There is a significant increase in efficiency from DP to TP effect at high pressure (HP). At a fixed steam turbine inlet pressure, 200 bar, SP HRSG exergy efficiency of CC increases all time with increase in steam reheater pressure (Fig. 3d). A steam re heater is added to increase dryness fraction of steam at turbine exit and to avoid erosion of blades during wet expansion. For DP and TP HRSGs, exergy efficiency maximizes at optimum steam reheater pressure of 100 bar with HP pressure at 200 bar. Optimum deaerator pressure with SP, DP and TP levels in HRSG is around 1, 3 and 5 bar respectively with HP pressure at 200 bar and steam reheating at 100 bar (Fig. 3e). Optimized exergy efficiency at these conditions is 52.5, 54.5 and 55.5 % with SP, DP and TP HRSGs respectively. Exergy efficiency decreases with increase in PP (Fig. 3f). Irreversibility losses in heat transfer between exhaust and steam increases with increase in temperature difference. Heat recovery also
drops with increase in temperature difference (PP). Therefore, at low PP, higher efficiency for CC can be achieved. But, close temperature difference increases size of heating device and hence the cost. Their choice is the consequence of a compromise between thermodynamic efficiency and investment costs. Thus, gain in efficiency from SP to DP is more compared to the gain from DP to TP level in HRSG.

For SP, DP and TP heat recovery, compressor pressure ratio and gas turbine inlet temperature are fixed. Therefore, topping cycle components exhibit no change on exergetic losses, which are compressor, gas combustion chamber and gas turbine. Steam generation rate increases from SP to TP in HRSG, thereby exergetic loss associated with heat transfer increases (Fig. 4). Exhaust temperature and hence exergetic loss in exhaust decreases from SP to TP levels in HRSG. Exergetic loss in expansion of steam in turbine rises with pressure levels due to excessive steam flow rate. This also increases condenser exergetic loss with increase in pressure levels. More amount of steam is required for feed water heater in case of SP HRSG whereas in DP and TP effect, amount of bled steam and hence exergetic loss in deaerator decreases. On overall basis, total exergetic loss decreases from SP to TP HRSG. Keeping exhaust gas flow rate fixed (650 kg/s), results (Table 2) for SP, DP and TP heat recovery agreed with literature values.

Conclusions

Optimum pressure ratio for compressor with SP, DP and TP effects in heat recovery are 8, 10 and 12 respectively at 1200°C of gas turbine inlet temperature. Optimum deaerator pressure is obtained at 1, 3 and 5 bar for SP, DP and TP levels respectively at steam turbine inlet pressure of 200 bar. Similarly, at 200 bar of HP pressure for DP and TP, steam reheater demands 100 bar to maximize exergy efficiency for CC. Parametric analysis exhibits that gain in efficiency form single pressure heat recovery to DP and TP recovery increasing with diminishing rate.

References