Thermoeconomic Evaluation of Combined Cycle Cogeneration Systems

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ABSTRACT

In this paper, several configurations of combined cycle cogeneration systems are investigated by energy, exergy and thermoeconomic analyses. In each of these configurations, varying steam demand is considered rather than fixed steam demand. Among the different approaches for thermoeconomic analysis in literature, SPECO method is applied. Since the systems have more than one product (process steam and electrical power), systems are divided into several subsystems and cost balances are applied together with the auxiliary equations. Hence, cost of each product is calculated. Comparison of the configurations in terms of performance assessment parameters and costs per unit of exergy are also given in this study.

NOMENCLATURE

BFW Boiler feed water
c Cost per unit of exergy [$/GJ]
\dot{C} Cost rate associated with exergy [$/h]
 CC Carring charges
\dot{E} Exergy flow rate [kW]
FUE Fuel utilization efficiency
H Enthalpy rate [kW]
HP High pressure
HRSG Heat recovery steam generator
IP Intermediate pressure
LHV Lower heating value
LP Low pressure
m Mass flow rate
O&M Operating and maintenance
PEC Purchased equipment costs
\dot{Q} Heat rate[kW]
T Temperature [°C]
W Power [kW]
\dot{Z} Cost rate associated with the capital and operating and maintenance expenses [$/h]
\varepsilon Exergetic Efficiency

Subscripts
ap Approach
El Electrical
f Fuel
pp Pinch point

INTRODUCTION

Combined cycle cogeneration systems produce electrical (or mechanical) power and useful thermal energy. Principal applications are in industrial sites and district heating.

The most widely used combined cycle systems are those of gas turbine-steam turbine. Main configuration differences arise due to using different types of steam turbines and heat recovery steam generators.

Steam turbines may be condensing and back-pressure (non-condensing) type. In back-pressure steam turbines, steam exits the turbine at a pressure higher or at least equal to the atmospheric pressure, which depends on the needs of the thermal load. In condensing steam turbines, steam for the thermal load is obtained by extraction from one or more intermediate stages at the appropriate pressure and temperature. The remaining steam is exhausted to the pressure of the condenser [12].

Depending on the steam requirements, HRSGs can take three forms: unfired, supplementary-fired and exhaust-fired [11]. Besides, multi-pressure HRSGs may be used to increase the heat recovery from the exhaust gas of the gas turbine.

Description of systems used in this study

Three different configurations of combined cycle cogeneration systems are chosen in this study. They are shown in Figure 1-3. Each of these cases has the same gas turbine system. The gas turbine system is taken same as
that of CGAM system (i.e. Air compressor, air preheater, combustion chamber and gas turbine of that system)[1].

Besides the gas turbine and auxiliary systems, case-1 consists of a two-pressure HRSG and a back-pressure steam turbine. Case-2 consists of a two-pressure HRSG and a condensing steam turbine. Case-3 consists of a three-pressure HRSG and a condensing steam turbine. In each of them, one of the steam generators supply steam to the integral deaerator to heat up the condensate to the saturation temperature corresponding to the pressure of its drum and degass that condensate. For controlling the steam turbine inlet temperature, a desuperheater may be used between first and second superheater stages. Besides, some of the steam needed to deaerate the condensate could be sent from a higher pressure steam drum to control stack temperature. Stack temperature should be controlled due to corrosion considerations. These security components are shown with dotted lines in Figures 1-3.

The systems used in this study can cope with their steam demand change. A dump condenser is used with the back pressure steam turbine in case-1 for this purpose. Extraction-condensing steam turbine is used for that in case-2 and case-3. The cooling water of the condensers are considered to be pumped from sea or lake.

The operation data used in the case studies are given in Table 1. Data for gas turbine system and environmental conditions are taken same as the base-case design values [1] of the gas turbine system.

<table>
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<th>Operation Data of Case Studies</th>
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<tr>
<td>HP Steam Drum Pressure</td>
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<td>Pinch Point and Approach Temp.</td>
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<tr>
<td>Steam Turbine Inlet Temp.</td>
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<td>Steam Export (Sat.vapor) Pressure</td>
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<td>Condenser Pressure</td>
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<td>Dump Condenser Pressure</td>
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<td>LP Steam Drum Pressure</td>
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<tr>
<td>Condensate Return Temp.</td>
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<td>Steam Turbine Isentropic Efficiency</td>
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<td>Pumps Isentropic Efficiency</td>
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<tr>
<td>Electric Generator Efficiency</td>
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</table>

Table 1: Operation data of case studies

Besides the assumptions for the gas turbine system given in [1], the assumptions done in this study are as follows: Heat loss from HRSG is assumed to be 2% of heat absorbed. Pressure drop on the gas side of HRSG is 5% and that of on the water side is neglected. Blow down requirements and deaerator vent flows are not taken into account. All the steam export from the system returns as condensate.

**THERMODYNAMIC ANALYSIS**

**Energy Analysis**

Basic thermodynamics, involving first and second law of thermodynamics, is applied to systems to find the thermodynamic properties of each state, and calculate the electrical output of bottoming cycle and thermal energy of process. In the analyses, kinetic and potential energy effects are ignored.

For ideal gases, formulations in [1] is used which takes into account the variation of enthalpy and absolute entropy with temperature for various substances. For water, steam tables at [9] are used. HRSG temperature profile for case-1 and case-2 is given in Figure 4. That for case-3 is given in Figure 5. Energy balances of a one-pressure HRSG may be found in [5]. With similar efforts, mass flow rate of steam generated at steam drums and thermodynamic properties of combustion gases at several points within HRSG may be found.

**Results of Energy Analysis**

Several energy balances are applied to HRSG sections. Then, steam generation rate at steam drums and stack temperature are calculated. Hence, it is found that in each of the cases, 11.1 kg/s steam is produced in the HP steam drums. In the third case, 1.78 kg/s steam is produced in the IP steam drum.

Stack temperature changes with steam demand in a small range. Average stack temperatures for the case-1, case-2 and case-3 are 184°C, 169 °C and 123 °C, respectively.

Electrical output of steam turbine does not change with steam demand in case-1. Besides, this case has the lowest electricity production because of the high exit pressure of steam turbine. For cases 2 and 3, the electrical output of steam turbine decreases as steam demand increases. For the same steam demand, case-3 has a higher electrical output because extraction of steam is less due to steam going to process from LP steam drum.
Prior to thermoeconomic analysis, exergy analysis should be done. In this study, cost of products of the system and cost formation within the system are to be calculated. Hence, it is sufficient to calculate the exergy rate of states in the systems (i.e. Exergy balances are not needed to be applied to subsystems of the systems). The exergy rates consist of physical and chemical exergy terms. Formulation of these terms may be found in [1]. Exergy is lost, in general, when the energy associated with a material or energy stream is rejected to the environment [4]. In the case studies, exergy rate of stream exiting HRSG and exergy rate difference between condenser cooling water inlet and outlet streams are exergy losses. The exergy loss due to heat loss in HRSG may be taken as zero by considering large enough control volume, so that heat loss occurs at the ambient temperature.

**Performance Assessment Parameters**

There are many performance assessment parameters related to energy analysis [6]. Among them, fuel utilization efficiency is the most widely used one. However, qualities of different kinds of energies are not taken into account in this parameter. The change of FUE with steam demand for the case studies are given in Figure 6.

\[
FUE = \frac{(\dot{W}_{net})_{plant} + \Delta \dot{H}_{process}}{m_{fuel} \cdot LHV}
\]  

Figure 6: Change of fuel utilization efficiency with process steam demand

Exergetic efficiency is the most meaningful performance parameter since different qualities of energies are taken into account. That efficiency for the case studies may be given as

\[
\epsilon_{plant} = \frac{(\dot{W}_{net})_{plant} + \Delta \dot{E}_{process}}{\dot{E}_f}
\]  

Figure 7: Change of exergetic efficiency with process steam demand

It is seen from Figure 6 and 7 that case-3 has the highest fuel utilization and exergetic efficiency for
a given steam demand. Case-1 has the lowest of them and it gets closer for higher steam demands.

**THermoEconomic Analysis**

There are several approaches for thermoeconomic analysis. Among them, SPECO method [2,3] is used in this study. The basic principle of the SPECO method is initially suggested by Tsatsaronis and Lin (1990). Then, it is developed by Tsatsaronis and his coworkers. The aim of the thermoeconomic analysis in this study is to calculate the cost of each product and investigate the cost formation process. A powerful economic analysis increases the accuracy of the results of thermoeconomic analysis. Levelized capital and O&M costs associated with each plant component and fuel cost are calculated in economic analysis. Then, they are used as inputs in thermoeconomic analysis. A detailed description of engineering economic analysis can be found in [1]. Each system is divided into several subsystems and cost balance is applied to each subsystem. Auxiliary equations are applied by using fuel and product rules. Hence, a set of linear equations are formed and solved.

The steady-state form of control volume cost balance is

\[
\sum_{j=1}^{n} C_{j,k,\text{in}} + \dot{Z}_k = \sum_{j=1}^{m} \dot{C}_{j,k,\text{out}} \quad (3)
\]

\[
\dot{C}_j = c_j \cdot \dot{E}_j \quad (4)
\]

Purchased equipment costs of components of gas turbine system, total annual number of hours of system operation at full load and levelized fuel costs are taken same as CGAM system. Levelized carrying charges and O&M costs should be greater than that of the CGAM system due to increase in plant components. So; for simplicity, it is assumed that summation of carrying charges and O&M costs of combined cycle cogeneration systems change such that $\dot{Z}_k$ of gas turbine components remain constant. Hence, cost of exhaust gas and power of gas turbine system remains constant for all cases.

Purchased equipment cost of HRSG is calculated by using the same formula given in [1]. However, the formula is extended for other HRSG sections and coefficients of the formula and unit costs are assumed to be same also for other sections. There is no formula given for other plant equipments; i.e., steam turbine, desuperheater, etc. Thus, back-pressure steam turbine price is taken for two values ($1000 and $2000), extraction-condensing steam turbine price is also taken for two values ($3000 and $4000). For other plant components, since their price are lower, reasonable values are taken.

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<th>Component</th>
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<th>$\dot{Z}$ ($/h$)</th>
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<td>Other Plant eq.</td>
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Table 2: Purchased equipment costs and costs associated with levelized capital and O&M costs of plant components

In Table 2, superscripts show the cases that the values are valid for.

For illustration, cost balances and auxiliary equations for case-2 are given below.

For the control volume around HRSG (including deaerator),

\[
(C_3 - \dot{C}_4) + (\dot{C}_{17} - \dot{C}_5) + (\dot{C}_{12} - \dot{C}_{13}) + \dot{Z}_{HRSG} = 0 \quad (5)
\]

\[
\frac{\dot{C}_3}{\dot{E}_3} = \frac{\dot{C}_4}{\dot{E}_4} \quad (6)
\]

\[
\frac{\dot{C}_5 - \dot{C}_{17}}{\dot{E}_5 - \dot{E}_{17}} - \frac{\dot{C}_{12} - \dot{C}_{13}}{\dot{E}_{12} - \dot{E}_{13}} = \frac{\dot{C}_3}{\dot{E}_3} - \frac{\dot{C}_4}{\dot{E}_4} \quad (7)
\]

For the control volume around steam turbine (including electric generator),

\[
(C_5 - \dot{C}_6 - \dot{C}_7) - \dot{C}_b + \dot{Z}_{ST} = 0 \quad (8)
\]

\[
\frac{\dot{C}_5}{\dot{E}_5} = \frac{\dot{C}_6}{\dot{E}_6} = \frac{\dot{C}_7}{\dot{E}_7} \quad (F rule) \quad (9)
\]

The electricity generated in the steam turbine splits into three streams. Two streams are sent to pumps. Remaining is the net power output of the
bottoming cycle. If cost balance is applied around this splitting point and from product rule, it may be shown that
\[ \frac{\dot{C}_B}{W_B} = \frac{\dot{C}_C}{W_C} = \frac{\dot{C}_D}{W_D} = \frac{\dot{C}_E}{W_E} \] (10)
For the control volume around desuperheater,
\[ \dot{C}_{15} + \dot{C}_6 - \dot{C}_8 + \dot{Z}_{\text{Desph}} = 0 \] (11)
For the control volume around condenser and condensate pump,
\[ (\dot{C}_7 - \dot{C}_{10}) + (\dot{C}_{18} - \dot{C}_{19}) + \dot{C}_c + \dot{Z}_{\text{con.}} + \dot{Z}_{\text{c.pump}} = 0 \] (12)
\[ \frac{\dot{C}_{18}}{E_{18}} = \frac{\dot{C}_{19}}{E_{19}} \quad \text{(F rule)} \] (13)
It’s assumed that \( \dot{C}_{18} = 0 \), so, from eqn (10) \( \dot{C}_{19} = 0 \). Besides, cost rate of condensate return is assumed to be zero. Hence,
\[ \frac{\dot{C}_{12}}{E_{12}} = \frac{\dot{C}_{10}}{E_{10}} \] (14)
For the control volume around BFW pump,
\[ \dot{C}_{13} + \dot{C}_D - \dot{C}_{14} + \dot{Z}_{\text{BFW Pump}} = 0 \] (15)
For the control volume around splitting point after BFW pump,
\[ \frac{\dot{C}_{14}}{E_{14}} = \frac{\dot{C}_{15}}{E_{15}} = \frac{\dot{C}_{17}}{E_{17}} \] (16)
Solving linear equation (5)-(16), cost formation within the system and cost of products may be found.

Cost of other plant equipment and the monetary loss associated with exergy lost should be apportioned between products. One assumption may be apportioning products according to their exergy rate.
\[ \dot{C}_{8-\text{final}} = \dot{C}_8 + (\dot{Z}_{\text{other}} + \dot{C}_4) \cdot \frac{\dot{E}_8}{W_A + W_E} + \frac{\dot{E}_{11}}{W_A + W_E} \] (17)
\[ \dot{C}_{E-\text{final}} = \dot{C}_E + (\dot{Z}_{\text{other}} + C_4) \cdot \frac{W_E}{W_A + W_E} + \frac{\dot{E}_8}{W_A + W_E} \] (18)

Cost balances and auxiliary equations for case-1 and case-3 may be written with similar efforts.

**Results of Thermoeconomic Analysis**

It is seen from Figure 8 that for the same process steam demand, case-1 has the highest specific cost of process steam and case-2 is slightly lower than case-3. In the case of bottoming cycle electrical output cost, cost of it for case-1 decreases while cost of it for case-2 and case-3 increases as steam demand increases. For case-1, it has lower cost only for high steam demands. When overall steam demand range is considered, it may be seen that case-3 has the lowest specific cost for bottoming cycle electrical output. Case-2 and case-3 follow it, respectively.

![Figure 8: Change of cost per unit of exergy of process steam with process steam demand](image)

![Figure 9: Change of cost per unit of exergy of bottoming cycle electrical output with process steam demand](image)

In the case of cost of electrical output of gas turbine system, cost of it found after cost balances and auxiliary equations are same in all cases due to the economic assumptions described in economic analysis section. However, when the cost associated with exergy loss is apportioned between products, case-1 has the highest cost of it. Case-2 and case-3 follow it, respectively.

Cost formation for 6 kg/s steam demand is given in Table 5 for illustration. Some cost values are not given (i.e. shown with dash line). This is because these states stay inside the subsystems, i.e. they are not inflows or outflows to the subsystem. In each of the configurations, states 1, 2, 3 and A correspond to air inlet, fuel inlet, exhaust gas and power output of gas turbine system. Other states are given in schematic drawings of systems.
CONCLUSIONS

In this paper, comparisons of combined cycle cogeneration systems are investigated through thermodynamic and thermoeconomic analyses. Results of these analyses are given in related sections.

For the same process steam demand, thermal energy and exergy rate of process steam are same for all cases. However, electrical power output of bottoming cycle is different. So; to make comparison, cost per unit exergy of products is selected.

From the results, it is seen that combined cycle cogeneration systems with extraction-condensing steam turbine are thermoeconomically more efficient. Increasing the pressure level of HRSG, increases slightly the cost of process steam and decreases slightly the cost of electrical output of bottoming cycle for the same demand. Results of such an analysis may also be helpful in determining the sale prices of products of a combined cycle cogeneration plant. Similar systems may be analyzed and compared with the methodology described in this paper.

<table>
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<th>( \dot{C} ) (S/h)</th>
<th>( c ) (S/GJ)</th>
<th>State (Case2)</th>
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Table 3: Exergy rate, cost rate, cost per unit exergy of states of cases (1000$ steam turbine for case-1, 3000$ steam turbine for case-2 and case-3) for 6 kg/s steam demand.
Figure 1: Schematic drawing of case-1

Figure 2: Schematic drawing of case-2
REFERENCES


