Effects of gas turbine inlet temperature on exergy efficiency, cost and environment of a combined cycle power plant

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Abstract: Thermal systems design and analysis involve principles of many fields of mechanical engineering like thermodynamics, heat transfer, fluid mechanics, manufacturing and mechanical design. In this work, the thermodynamics aspect of the design is handled. So, in this paper, the effects of gas turbine inlet temperature (GTIT) on design parameters of CCPP were performed. In this regard, the energy and exergy of the plant was analyzed to show the irreversibility in each component of the system. In addition, a parametric study has been performed to investigate the effects of GTIT on energy, exergy, and efficiencies, exergo-economic and environmental of the system. Results show that an increase in GTIT, the exergy efficiency is increased and exergy destruction rate decreases for both gas turbine cycle and steam cycle. Consequence, the CO2 emission of both gas cycle and steam cycle decreases.

Key-Words: Energy, Exergy, Efficiency, Economic, Environment.

1 Introduction

Energy is one of the top10 crucial items in our daily life for almost everything and its production, from heating and cooling applications to nuclear power plants [1, 2]. Exergy is a measure of the utility or value or quality of energy form. Technically, exergy means using thermodynamics principles for the maximum amount of work that can be generated by a system or a flow of matter or energy as it comes to equilibrium with environment [3, 4]. The exergy method of thermal plant analyses by Tadeusz J Kotas [5] explained an introduction of exergy method in thermal system and optimization. Therefore, the advantages of the use of exergy analysis were demonstrated by pointing out and quantifying thermodynamic losses of various plant components and plant configurations. Moreover, this book helps students to have a deeper understanding of the nature of irreversibilities of various kinds and their effect on plant performance. Lior [6] proposed a concept about future power generation systems and the role of exergy analysis in their development. He illustrated some thoughts to meet the power demands under the constraints of increased population and land use when holding the environmental impact to a tolerable one. Next he focused on exergy analysis which would be essential in the conception and development of such processes. Finally he discussed about which kind of development is essential for the power generation units in the coming century. The results indicated that exergy analysis will help the designers to come up with a good decision on how to improve the system performance. Economic issue is important in the evaluation of energy technologies, energy conversion devices and costs of energy system. Some researchers [7-9] have suggested some methods which show costs are better shared among outputs based on exergy. Therefore, these methods have been developed on performing economic analyses based on exergy, which are called as second law costing, or thermoeconomics and exergoeconomics [8-10]. In this regard, Enadi et al. [11] investigated the second law based optimization of a micro gas turbine (MGT) cycle for residential application using a genetic algorithm. They conducted a comprehensive thermodynamic modeling of a combined heat and power (CHP) system. Then the optimal design parameters obtained by introducing an objective function, a set of design parameters, and the thermo-physical constraints of the CHP system. They compared their results from thermodynamic modeling and optimization procedure with the results reported in CGAM problem. They also studied the variations of optimal design parameters when the power and steam process demands change. Finally by changing in capital investment the sensitivity of the optimal design parameters were studied.
One of the most important concerns for human is to reduce environmental impact of energy system and make use of sustainable energy technologies to mitigate global warming. Moreover, by reducing CO2 emission can increase efficiency. In this regard Ganjeh Kaviri [12] investigated the exergo-environmental optimization of a heat recovery steam generators in combined cycle power plant. Barzegaravval et al. [13] studied the thermoeconomic environmental multi objective optimization of a gas turbine power plant using evolutionary algorithm. They compared their simulation code with the data obtained from an actual gas turbine power plant located near the Yazd city, one of the middle provinces in Iran to ensure the accuracy of the developed code. They considered three different objective functions and used multi-objective optimization in order to optimize the system for the better performance assessment. The results from optimization showed that the overall exergoeconomic factor of the system increases from 32.79 to 62.24% compared with the actual power plant.

In this paper, a mathematical model is developed for a combined cycle power plant. Energy, exergy, and exergoeconomic analyses of this power plant are conducted. In addition, a parametric study is performed to investigate the effects of varying gas turbine inlet temperature on the cost, energy and exergy, efficiencies, economic and environmental of the system.

2 Description of case study
The thermodynamic modelling of the combined cycle power plant system considered here (Fig. 1) is divided into three main system: topping cycle (Gas cycle), bottoming cycle (Steam cycle) and a dual pressure heat recovery steam generator (HRSG). The output power of this power plant is about 415.1 MW.

3 Assumption
The thermodynamic model is developed based on some basic assumptions which are as follows [1].
1. All processes in this case are assumed as steady state.
2. The air and the gases resulting from combustion are considered as ideal gases.
3. The energy variation and the kinetic and potential exergies are assumed negligible.
4. The natural gas is injected to the combustion chamber and duct burner as natural gas.
5. Three percent of the fuel lower heating value (LHV) is considered as heat loss from combustion chamber, and other components are assumed to be adiabatic.
6. The dead state for this case state is considered as P0=1.01 bar and T0=293.15oK.
7. 0.03 pressure drops is considered along the heat recovery steam generator and also 0.05 pressure drops is considered in combustion chamber.

![Fig. 1 Thermal schematic diagram of a combined cycle with dual pressure HRSG](image)

4. Energy, exergy, exergo-economic and exergo-environmental analysis
We model a topping cycle, which consists of a gas turbine cycle using the first law of thermodynamics. As seen in Fig. 1, air at ambient conditions enters the air compressor at point 1 and, after compression (point 2), is supplied to the combustion chamber (CC). Fuel is injected in the combustion chamber and hot combustion gases exit (point 3) pass through a gas turbine to produce shaft power. The hot gas expands in the gas turbine to point 4, after which it passes through the duct burner to warm air. These hot gases leave the duct burner at point 5 and enter the HRSG linked to the bottoming cycle.
To determine the exergy at different points of the power plant, the temperature profile, input, output, enthalpy of each line in the combined cycle power plant have been conducted.

**Air compressor (AC)**

\[ T_i = T_1 + \frac{1}{\eta_{AC}} \left[ T_2 - T_1 \right] \]

\[ W_{AC} = \dot{m}_c c_p (T_2 - T_1) \]  
(1)

**Combustion chamber (CC)**

\[ \dot{m}_c h_c + \dot{m}_{f-CC} LHV = \dot{m}_c h_c + (1 - \eta_{CC}) \dot{m}_{f-CC} LHV \]  
(2)

**Gas Turbine (GT)**

\[ W_{GT} = \dot{m}_c c_p (T_3 - T_1) \]

\[ \frac{P_2}{P_1} = 1 - \Delta P_{co} \]  
(3)
m\textsubscript{q}\textsubscript{t} + m\textsubscript{y}\textsubscript{t}LHV = (m\textsubscript{y} + m\textsubscript{t}LHV)h\textsubscript{y} + (1 - \eta\textsubscript{t})m\textsubscript{t}LHV \tag{5}

High-Pressure super heater (HP SH)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{15} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{HP}(h\textsubscript{25} - h\textsubscript{24}) \tag{6} \]

High-Pressure evaporator (HP Evap)
\[ T\textsubscript{e} = T\textsubscript{24} + H\textsubscript{HP}\textsubscript{patch}\textsubscript{point} \quad T\textsubscript{w} = T\textsubscript{24} - H\textsubscript{HP}\textsubscript{Approachpoint} \]
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{24} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{HP}(h\textsubscript{24} - h\textsubscript{23}) \tag{7} \]

Second High-Pressure economizer (2Th HP Eco)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{24} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{Eco}(h\textsubscript{22} - h\textsubscript{21}) \tag{8} \]

Low-Pressure super heater (LP SH)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{15} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{LP}(h\textsubscript{19} - h\textsubscript{18}) \tag{9} \]

Low-Pressure evaporator (LP Evap)
\[ T\textsubscript{e} = T\textsubscript{18} + L\textsubscript{LP}\textsubscript{patch}\textsubscript{point} \quad T\textsubscript{w} = T\textsubscript{18} - L\textsubscript{LP}\textsubscript{Approachpoint} \]
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{18} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{LP}(h\textsubscript{18} - h\textsubscript{17}) \tag{10} \]

First High-Pressure economizer (1St HP Eco)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{18} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{Eco}(h\textsubscript{21} - h\textsubscript{20}) \tag{11} \]

Dearator (DEA)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{15} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{DEA}(h\textsubscript{16} - h\textsubscript{15}) \tag{12} \]

Economizer Preheated (Ec-Pr)
\[ m\textsubscript{g}\textsubscript{pg}(T\textsubscript{12} - T\textsubscript{e}) = m\textsubscript{h}\textsubscript{Ec}(h\textsubscript{14} - h\textsubscript{13}) \tag{13} \]

Steam Turbine (ST)
\[ m\textsubscript{h}\textsubscript{26} + m\textsubscript{c}\textsubscript{h}\textsubscript{28} = W_{ST} + m\textsubscript{c}\textsubscript{h}\textsubscript{28} \tag{14} \]
\[ \eta_{ST} = \frac{W_{ST,crit}}{W_{ST,cr}} \]

Condenser (Cond)
\[ m\textsubscript{h}\textsubscript{28} + m\textsubscript{c}\textsubscript{h}\textsubscript{29} = m\textsubscript{h}\textsubscript{29} + m\textsubscript{c}\textsubscript{h}\textsubscript{31} \tag{15} \]

Condenser Pump (CP)
\[ m\textsubscript{h}\textsubscript{29} = W_{p} + m\textsubscript{h}\textsubscript{28} \tag{16} \]
\[ \eta_{pump} = \frac{h_{4(0)} - h\textsubscript{29}}{h_{4(0)} - h\textsubscript{28}} \]

To evaluate the properties of gases and water Refpro 8.0 software has been coupled with MATLAB software through DLL (Dynamic Link Library) and MEX programming.

Exergy is a measure of the maximum capacity of a system to perform useful work, as it proceeds to a specified final state in equilibrium with its surrounding.

Applying the first and second laws of thermodynamics, the following exergy balance can be obtained.
\[ \dot{E}_{x,y} + \sum_{i} m_{i}e_{i} = \sum_{i} m_{i}e_{i} + \dot{E}_{x,y} + \dot{E}_{y} \tag{17} \]

where the subscripts e and i denote the inlet and outlet of a control volume \( \dot{E}_{x,y} \) is the exergy destruction and other terms are as follows [5]:
\[ \dot{E}_{x,y} = \left[ 1 - \frac{T_{y}}{T_{0}} \right] Q_{i} + \dot{E}_{w} = W \quad \dot{e}_{x,ph} = (h - h_{s}) - T_{s} (s - s_{s}) \]
\[ \dot{e}_{x,ch} = - \sum_{i} X_{i} \dot{e}_{x,i} + RT_{0} \sum_{i} X_{i} \ln X_{i} \tag{18} \]

Here, \( \dot{E}_{x,y} \) and \( \dot{E}_{x,y} \) are the corresponding exergy rates, associated with heat transfer and work across the boundary of a control volume, respectively, and \( T \) is the absolute temperature and the subscript 0 refers to the reference environment conditions. The reference environment considered here are \( T_{0} = 20 \degree C \) and \( P_{0} = 1.01 \) bar.

For each flow in the system, a parameter called flow cost rate was defined, and the cost balance was written for each component as follows:
\[ \dot{C}_{k} = \dot{C}_{k} \dot{E}_{x,k} + Z_{k} \frac{C_{k} \dot{E}_{x,k}}{N \times 3600} \tag{19} \]

Where \( \dot{C}_{k} \) is the purchase of the kth component, \( N \) is the annual number of operation hours for the unit, \( \dot{C}_{k} \) is the maintenance factor which is usually 1.06 and CRF is the capital recovery factor. Capital recovery factor depends on equipment life time and the interest rate which was determined as follows
\[ CRF = \frac{i \times (1+i)^{n}}{(1+i)^{i}-1} \tag{20} \]

where i and n are the interest rate and the total operating period of the system in years respectively. By using cost balance equations for each component separately, a set of linear equations are produced which their simultaneous solution results in streams cost. \([A][C]=[B]\) \tag{21}

Where C matrix is a vector of unknown stream cost values corresponding to each state point in figure 1. A is the vector of unknown cost values in cost equations. B is the vector of constant values in cost equations and its elements are combinations of fuel cost rates.

The total cost rate of the plant is the summation of purchase cost of each component, fuel cost, cost of exergy destruction and the environmental cost which is as follows:
\[ C_{total} = \dot{C}_{f} + \sum_{i} Z_{i} + C_{env} + C_{D} \quad \dot{C}_{f} = c_{f} \dot{m}_{f} \times LHV \]
\[ C_{env} = c_{CO2} m_{CO2} + c_{NOx} m_{NOx} + c_{CO} m_{CO} \tag{22} \]

Where \( \dot{C}_{D} \) are fuel cost and cost of exergy
destruction, respectively. Further information about the cost balance equation is given in [3, 5].

NO\textsubscript{x} formation and CO emission are more accurately related to the primary combustion parameters. Therefore, adiabatic flame temperature and also resident time in this region affect the NO\textsubscript{x} and CO emission. High resident time increases NO\textsubscript{x} but reduces the CO value. Thus, the analytical prediction of gas turbine combustion emission is provided.

The amount of CO and NO\textsubscript{x} produced in a combustion chamber and combustion reaction is dependent on the adiabatic flame temperature [5, 13]. The pollutant emissions (in grams per kilogram of fuel) can be identified as follows [14]:

\[
m_{NO_x} = \frac{0.156E16a^{0.5}e^{-(71100/T_p)}}{p_{3}^{0.55}(\Delta p_p/p_{3})^{0.5}}
\]

(24)

\[
m_{CO} = \frac{0.179E7e^{(7800/T_p)}}{p_{2}^{0.5}(\Delta p_p/p_{3})^{0.8}}
\]

(25)

Where \(\tau\) is the residence time in the combustion zone (assumed constant and equal to 0.002 s) [15], \(T_P\) is the primary zone at combustion, \(P_3\) is the combustor inlet pressure, and \(\Delta p_p/p_{3}\) is the non-dimensional pressure drop in the combustion chamber. These equations which are based on experimental values provide estimation for NO\textsubscript{x} and CO emissions and are used in environmental assessment.

Source of carbon emission in power plants or all energy conversion come from their fuels. In this regard, fossil fuels including carbon based like coal and natural gas are sources of CO\textsubscript{2} emission. This gas is one of the main products of combustion and there is no method to reduce its emission except reducing the fuel flow or capturing methods. Reducing the fuel flow means higher efficiencies in the cycle.

If the combustion is considered as a complete combustion, the value of CO\textsubscript{2} emission can be calculated by using the following simple equation:

\[
\dot{m}_{CO_2} = 44.01x \left( \frac{m_{fuel}}{M_{fuel}} \right)
\]

(26)

where \(x\) is the mole fraction of carbon in the fuel and \(M_{fuel}\) is the molar weight of the fuel. This simple equation estimated the CO\textsubscript{2} emission accurately for complete combustion. In this case the total carbon dioxide is the sum of carbon dioxide produced in duct burner and gas turbine combustion.

4 Results and Discussion

The design variable which is selected to be consider here is gas turbine inlet temperature. Higher temperature requires better components to be able to utilize the energy. Therefore, by increasing the gas turbine inlet temperature, both HRSG purchase cost and overall cost will be sharply increase as it is shown in Fig 2.

![Fig.2 Effect of GTIT on HRSG purchase cost and investment cost of overall plant](image1)

By increasing gas turbine inlet temperature, efficiency of the cycle increases. Thus, the environmental cost and fuel cost are reduced. Reduction in environmental cost is due to lower carbon dioxide and CO emissions. These trends are shown in Fig 3.

![Fig.3 Effect of GTIT on Fuel cost and environmental cost](image2)

![Fig.4 Effect of GTIT on exergy efficiency and exergy destruction of gas cycle](image3)
GTIT has a major effect on the gas turbine performance; these effects are presented in Fig 4. The hot source temperature for gas cycle is in GTIT. Higher temperature in GTIT leads to the higher equivalent Carnot cycle efficiency for gas turbine cycle. Therefore, exergy destruction decreases in gas turbine cycle while exergy efficiency increases. By increasing GTIT the gas turbine specific work will be increased while the compressor specific work is constant. The effect of GTIT on exergy efficiency and exergy destruction rate of steam cycle is shown in Figure 5. As it is presented, by increasing the GTIT, higher temperature will be applied to the bottoming cycle. Higher steam temperature at the steam turbine inlet will have higher efficiency in bottoming cycle. Higher temperature in steam cycle has a same effect as gas cycle. Higher specific work in steam turbine with constant mass flow rate means higher net power produced.

Figure 6 shows the effect of GTIT on exergy efficiency and exergy destruction rate of combined cycle power plant. As presented, an increase in GTIT, the exergy efficiency is increased and exergy destruction rate decreases for both gas turbine cycle and steam cycle. Since the overall plant follows this trend, thus the exergy efficiency is increased and exergy destruction rate decreases in the combined cycle power plant accordingly. In this regard, for each 37.5 oK increases in GTIT, the efficiency of the plant increases for about 1% which is a great amount. This shows the importance of the GTIT which is the hot source temperature of the cycle or highest temperature.

By increasing GTIT, the exergy efficiency increases. This means higher power per unit of fuel consumption is produced. Therefore, the CO2 emission of both gas cycle and steam cycle decreases sharply as it is shown in Fig 7.

NOx is a direct function of combustion chamber flame temperature. Therefore, an increase in GTIT, results in increase in NOx formation while this leads to decrease in CO formation consequently. The details about the trend are provided in Fig 8.
Figure 9 shows the effect of GTIT on the total carbon dioxide emission. As it is presented an increase in GTIT results in exergy efficiency rises while emission for both cycles and also total CO2 emission of the plant is reduced. On the other hand, for each 1 Kelvin degree increment in turbine inlet temperature, carbon emission of the plant is reduced by 23152 ton per year.

5 Conclusions
Energy, exergy, and exergo-economic of a CCPP has been conducted. Then the effects of varying gas turbine inlet temperature on the cost, energy and exergy, efficiencies and emission of the system was performed. In addition the parametric study is conducted to show how a major design parameter would influence the system performance. Results show, by increasing the GTIT, the exergy efficiency is increased and exergy destruction rate decreases for both gas turbine cycle and steam cycle. Since the overall plant follows this trend, thus the exergy efficiency is increased and exergy destruction rate decreases in the combined cycle power plant accordingly. And also an increase in GTIT, the NOx of the plant will increase while this leads to decrease in CO of the plant.

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