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EFFECTS OF DUCT BURNER ON BOTTOMING CYCLE IN A COMBINED CYCLE POWER PLANT

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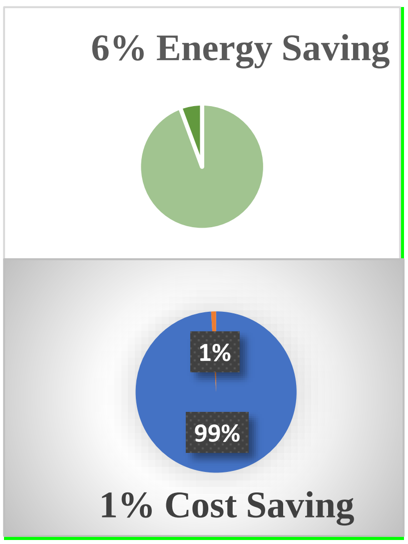
Graphical abstract



Abstract

In this study, thermodynamic modeling and exergoeconomic assessment of a Combined Cycle Power Plant (CCPP) with a Duct Burner (DB) was performed. Obtaining an optimum condition for the performance of a CCPP, using a DB after gas turbine was investigated by various researchers. DB is installed between gas turbine cycle and Rankine cycle of a CCPP to connect the gas turbine outlet to the Heat Recovery Steam Generator (HRSG) in order to produce steam for bottoming cycle. To find the irreversibility effect in each component of the bottoming cycle, a comprehensive parametric study is performed. In this regard, the effect of DB fuel flow rate on cost efficiency and economic of the bottoming cycle are investigated. To obtain a reasonable result, all the design parameters are kept constant while the DB fuel flow rate is varied. The results indicate that by increasing DB fuel flow rate, the investment cost and the efficiency of CCPP are increased. T-S diagram reveals that by using a DB, higher pressures steam in heat recovery steam generator has higher temperature while the low pressure is decreased. In addition, the exergy of flow gases in heat recovery steam generator increases. So, the exergy efficiency of the whole cycle was increased to around 6 percent, while the cost of the plant reduced by one percent.

Keywords: Exergy efficiency and destruction, combined cycle power plant, duct burner, bottoming cycle

1.0 INTRODUCTION

The high prices of energy and depletion of fossil fuel resources highlighted the importance of optimum application in energy conversion management and energy consumption. Today Combined Cycle Power Plant (CCPP) is known as one of the best choices to produce electricity due to the high efficiency and less carbon emission comparing to gas cycle power plant [1].

One way to improve an energy system efficiency is to find the irreversibility of each component in the system and try to reduce them. To find these irreversibility, an exergy analysis is needed to determine the types and magnitudes of inefficiencies [2, 3]. Recently thermoeconomic analysis also have been utilized to the energy systems to find the cost of the thermodynamic inefficiencies in the total product [4-6]. To improve the efficiency of various systems, multigeneration energy systems were proposed. Many relevant researches and studies have been performed on the CCPP power generation and multigeneration cycles up to now.

Economy of the energy systems are as important as their efficiency and environmental performance. Various studies analysed biogas fuelled economy with different approaches and measures. Kang *et al*. [7], analysed the economy of a CHP plant based on gas turbine using co-firing natural gas and biogas considering a complete plant including a biogas generator. They concluded that economic of the plant is affected greatly by changes in fuel combination. Pipatmanomai, *et al*. [8] analysed a power generation scenario using pig farm wastes to produce biogas including all processes like H_2S removal and evaluated payback period as an economic measurement. The prime mover in their study was an internal combustion engine for power generation. It was concluded that price of electricity affects the payback period of the plant significantly and governmental subsidy is important as well. Budzianowski and Budzianowska [9] compared different digestion systems in terms of operating pressure for power and heat generation. They considered, pressurized and atmospheric digestion system for biogas production. In addition, biogas upgrading options and cost evaluation of different cases were taken into account. They concluded that pressurized digestion system can be more economic. It was also noted that under the current policy, conventional CHP systems are more attractive economically. Basrawi *et al*. [10] investigated the optimal sizing of a cogeneration plant using economic and thermodynamic modelling. They analysed three sizes of gas turbine including 30, 50 and 200 kW and concluded that 200 kW is the most economic option in terms of NPV.

Kang *et al*. [11] compared CHP and Combined Cycle based on a 5 MW biogas fuelled gas turbine, economically. They considered different economic measures and heat demand patterns and selling price. They modelled the system hourly and considered a constant composition of 65% methane and 35% CO_2 for biogas. They concluded that especially for cases with heat demand and high selling price of heat, CHP has economical advantage over the single gas turbine and combined cycle system.

In addition to energy and economy analysis, off design and operation optimization of power generation systems were carried out by some researchers. Yağlı, *et al*. [12] provided a parametric optimization of an ORC for waste heat recovery from exhaust gases of a CHP plant. They used parametric study to find the optimum design of bottoming ORC cycle and cost analysis was not considered. Optimal subcritical and supercritical systems were calculated 27.2% and 27.76 % exergy efficiency respectively.

Chanel Ann Gibson *et al*. [13] investigated the economy of a gas turbine CHP system working with a mixture of natural gas and biogas as fuel. The main parameter in their study was carbon pricing in Australia. They used a partial load model for gas turbine and heat recovery system. They considered a gasification for wood pellets and concluded that with current cost of biomass in Australia, biogas is not a competitive option in comparison to natural gas.

Yechiel and Shevah [14] analysed flare to power option from a municipal solid waste landfill. A linear programming method was used by the authors to optimize the economic output of the plant operation. They considered two internal combustion engines each capable of producing 1 MW. The authors stipulated that plant economic performance will be 20% higher if biogas storage is used and power generation limited to the network peak power.

Exergy has been proven to be a powerful tool for evaluating the performance of the energy systems. Exergy in fact provides the second law analysing tool and provides an inside through the cycle losses and destructions and it has been used to assess biogas fuelled system as well. Farhad, *et al*. [15] compared three different system configurations for a SOFC based micro CHP for a residential building. They used exergy analysis to compare the systems and measured the net electrical efficiency. The average biogas composition which produced in Ontario waste water treatment plant was considered. Hosseini *et al*. [16] investigated a micro-power generation cycle based on the energy and exergy analyses for various biogas compositions. The system was a combination of a gas turbine with preheater and ORC as bottoming cycle. They carried out a parametric study on various parameters including fuel composition variations.

Some researchers, investigated the economy of the plant and considered the exergy analysis for the system performance evaluation and separated exergy and economy. Ozdil and Tantekin [17] analysed an onsite electricity generation for a wastewater treatment based on exergy economic analysis. A gas engine coupled with gas turbine was used for power generation and the waste heat was recovered and used in waste treatment plant. Their results showed that the cost of fuel in terms of exergy before feeding to the fuel compressor was about 4.88US/GJ. Wu *et al*. [18] designed three systems using biogas energy, biogas upgrading, conventional CHP and SOFC power generation. Exergetic-environmental and economic analyses was employed by the authors to design and compare the results. They concluded that biogas upgrading option is more economically and technically viable than other systems. They examined a general block box thermodynamic modelling and not a detailed exergy analysis was carried out and their focus were more on the economy of the plant. To find the irreversibility and exergy analysis of the CCPP the mathematical model of this CCPP is developed in MTLAB software. A parametric study is performed to determine the effects of DB fuel flow rate on the efficiency, economic and environmental of bottoming cycle.

2.0 METHODOLOGY

2.1 Description of Case Study

The Neka CCPP which is located near the Caspian Sea in Iran has two compressors, two gas turbines, two

HRSGs, two deaerators, one steam turbine and one surface condenser with a cooling system. A DB is placed between the gas turbine system and heat recovery steam generator to increase the flue gas entering the HRSG. The output power of this power plant is about 415.1 MW which is illustrated in Figure 1.

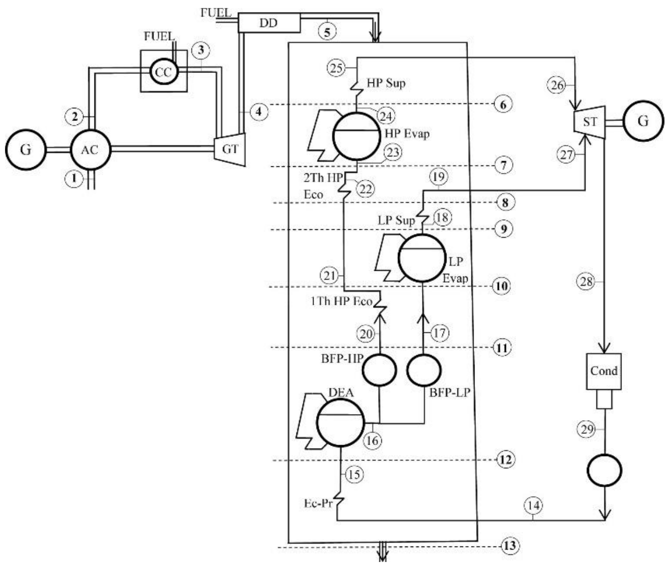


Figure 1 Thermal schematic diagram of a CCPP

Following assumptions are applied to assess the thermodynamic modelling [19]:

- All the processes considered as steady-state model.
- The ideal gas mixture is assumed for combustion products.
- Methane (CH_4) is the fuel which is injected to the combustion chamber and DB.
- Three present heat loss is considered from the chamber.
- Rest of components are considered adiabatic.
- The dead state is T0=293.15 K and P0=1.01 bar.

2.2 Energy, Exergy, and Exergo-Economic Analysis

By applying the energy balance equation to each component of this CCPP (Figure 1), the thermodynamic properties of each point of it is specified. To find these thermodynamic properties, a simulation program was developed in Matlab software. We need these properties to do the exergy and economic analysis. The energy balance equations for various parts of the CCPP (Figure 1) are as follows:

Air compressor (AC)

$$T_2 = T_1\left\{1+\frac{1}{\eta_{AC}}\left[r_c^{\frac{\gamma_a-1}{\gamma_a}}-1\right]\right\} \tag{1}$$

$$\dot W_{AC} = \dot m_a c_{pa}(T_2 - T_1) \tag{2}$$

In these equations T represent temperature, η is efficiency, *a* is air, *r* is specific heat, γ is specific heat ratio, W is the work, m is mass flow rate, c_p is specific heat at constant pressure.

Combustion chamber (CC)

$$\dot m_a h_3 + \dot m_{f-CC}\, LHV = \dot m_g h_4 + \dots (1-\eta_{CC})\dot m_{f-CC}\, LHV \tag{3}$$

LHV represent lower heating value of methane, h is specific heat, f is fuel.

Gas Turbine (GT)

$$\dot W_{GT} = \dot m_g . c_{pg}(T_3 - T_4) \tag{4}$$

$$\frac{P_4}{P_3} = (1-\Delta p_{cc}) \tag{5}$$

P represent the pressure, g is gas and W is the work.

Duct Burner (DB)

$$\dot m_g h_4 + \dot m_{f-DB}\, LHV = \left(\dot m_g + \dot m_{f-DB}\right) h_5 + \dots \quad (1-\eta_{DB})\dot m_{f-DC}\, LHV \tag{6}$$

The complete thermodynamic modeling of this power plant can be found in reference [1].

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.

The value of heat release in near ambient temperature at condensers (at a power plant cycle) is about two-thirds of total energy input but it does not worth two-thirds of the exergy source (fuel). The reason is that this amount of energy cannot produce useful work except a little amount. So, there must be a measure for flow values according to their ability to produce useful work which is called exergy.

The exergy balance equation is extracted from the first and second laws of thermodynamic [20].

$$\dot{E}x_Q + \sum_i \dot m_i ex_i = \sum_e \dot m_e ex_e + \dot{E}x_W + \dot{E}x_D \tag{7}$$

In this equation subscripts e and i denote the inlet and outlet of a control volume, Ex is the exergy, D is the destruction and [8]:

$$\dot{E}x_Q = \left(1-\frac{T_\circ}{T_i}\right)\dot Q_i \qquad \dot{E}x_W = \dot W \tag{8}$$

$$ex = ex_{ph} + ex_{ch} \tag{9}$$

$$ex_{ph} = (h - h_\circ) - T_\circ(s - s_\circ) \tag{10}$$

$$ex_{mix}^{ch} = \sum_{i=1}^{n} X_i ex^{ch_i} + RT_0 \sum_{i=1}^{n} X_i LnX_i \tag{11}$$

Here, $\dot{E}x_Q$ and $\dot{E}x_W$ 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 ^{0}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:

$$\dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k = \sum_e \dot{C}_{e,k} + \dot{C}_{w,k} \quad (12)$$

$$\dot{C}_j = c_j \dot{E}x_j \quad (13)$$

$$\dot{Z}_k = \frac{Z_k.\mathrm{CRF}.\phi}{N \times 3600} \quad (14)$$

Here Z_k is the purchase of the kth component, N is the annual number of operation hours for the unit, φ 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

$$CRF = \frac{i \times (1+i)^n}{(1+i)^n - 1} \quad (15)$$

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 [9]. [A][C] = [B]

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 follow:

$$\dot{C}_{Total} = \dot{C}_F + \sum_k \dot{Z}_k + \dot{C}_{Env} + \dot{C}_D \quad (16)$$

$$\dot{C}_{env} = c_{CO}\dot{m}_{CO} + c_{NO_x}\dot{m}_{NO_x} + c_{CO_2}\dot{m}_{CO_2} \quad (17)$$

$$\dot{C}_f = c_f \dot{m}_f \times LHV \quad (18)$$

$$\dot{C}_D = c_F \dot{E}x_D$$

Here $\dot{C}_F$ and $\dot{C}_D$ are fuel cost and cost of exergy destruction and $\dot{Z}_k$ is the purchase cost of each component. Further information about the cost balance equation is given in [21].

3.0 RESULTS AND DISCUSSION

The gas flow from the gas turbine outlet in a CCPP has a restricted energy and temperature. So, a DB is added to increase the overall output power of the cycle and also ensure the superheat degree at turbine exit at high steam flow rates. The effects of this DB on the efficiency, economic and environmental of CCPP are investigated and discussed.

As it is obvious, DB fuel flow rate has no effect on the gas cycle performance. Effect of DB fuel flow rate on the HRSG purchase cost and total purchase cost of the plant is illustrated in Figure 2. By increasing the fuel flow rate in DB, the HRSG cost and total purchase cost increases due to the higher temperature of bottoming cycle.

Figure 3 shows the effect of DB mass flow rate on both exergy efficiency and exergy destruction rate of the bottoming cycle. Combustion process is the main source of entropy generation due to the high temperature difference. Thus more fuel causes more entropy generation rate which eventually leads to an increase in exergy destruction rate of the cycle while overall exergy efficiency at bottoming cycle decreases as the denominator of the definition of overall exergy efficiency increases. It should be noted that although the steam turbine power production increases, the rate of increase in mass flow rate is dominated which results in reduction of overall exergy efficiency of the bottoming cycle.

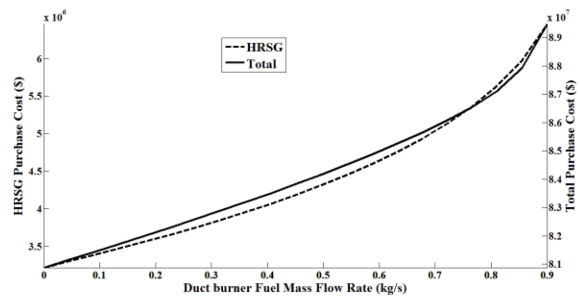


Figure 2 Effect of DB fuel flow rate on the HRSG purchase cost and total purchase cost

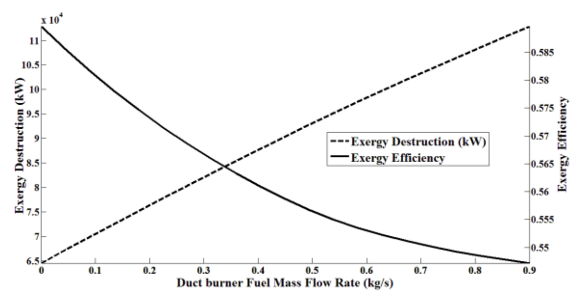


Figure 3 Effect of DB fuel flow rate on the exergy efficiency and exergy destruction rate of the bottoming cycle

Figure 4 shows the effect of DB fuel flow rate on exergy efficiency and exergy destruction rate of the steam cycle. By increasing the fuel flow rate of DB, the exergy efficiency and exergy destruction of the steam cycle are increased as well. These increments are due to the high temperature in HRSG inlet gas while the mass flow rate of steam is constant.

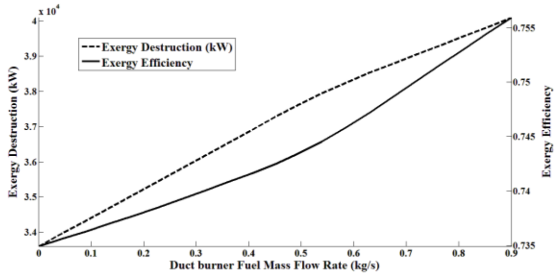


Figure 4 Effect of DB fuel flow rate on exergy efficiency and exergy destruction of the steam cycle

By increasing the fuel flow rate of DB, exergy destruction rate of total plant increases due to the increment of destruction in both combustion and steam cycle. But the trend of exergy efficiency is different. Exergy efficiency of the plant decreases firstly and then increases. The reduction of exergy efficiency in total plant is due to adding more fuel and combustion destruction to the cycle. Further, the increment of exergy efficiency in total plant is due to higher efficiency and temperature at steam cycle which is shown in Figure 5.

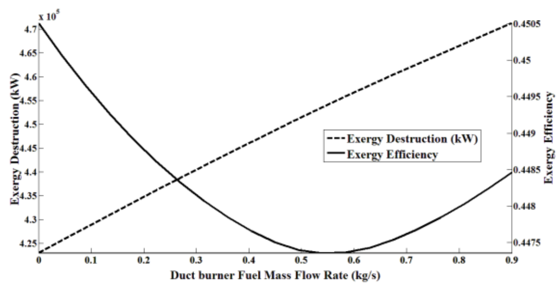


Figure 5 Effect of DB fuel flow rate on exergy efficiency and exergy destruction of plant

Figure 6 shows the effect of DB fuel flow rate on CO_2 emission of the plant. By increasing the fuel flow rate of DB, more net CO_2 emission is produced. In the other hand, although more net CO_2 emission is produced, but efficiency is reduced firstly and then increases. Therefore, the trend of Figure 6 which follows the efficiency tells the net emission of the cycle reaches a maximum inversely related to efficiency variations.

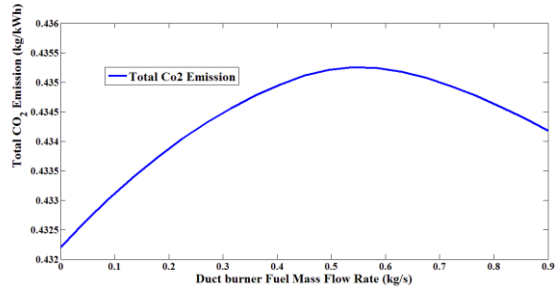


Figure 6 Effect of DB fuel flow rate on CO_2 emission of the plant

T-S diagram is a tool to demonstrate the flow diagram of each cycle. The T-S diagram of design cycle and the cycle with DB for both gas cycle and steam cycle is shown in Figure 7. By using DB, HP in steam line increases while the pressure of LP in steam line is decreased as it is shown in Figure 7.

The exergy efficiency and exergy destruction rate of combined cycle power plant components and also the whole gas cycle are also considered in another paper published by[1]. The comparison of the results of this paper with our results confirms that both results have the same tendency. Moreover, the exergy efficiency of the combustion chamber and duct burner is much lower than that of other plant components which is due to the high irreversibility rate in the combustion chamber resulting from the high temperature difference between flame temperature and working fluid.

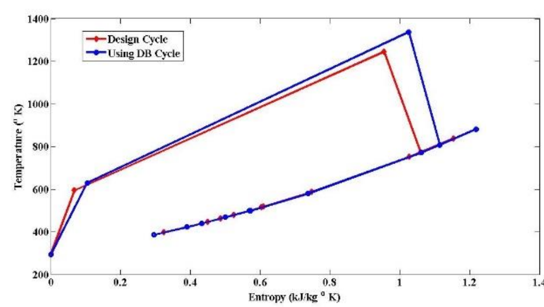


Figure 7 Comparing gas turbine cycle for both cycles in T-S plane

4.0 CONCLUSION

Thermodynamic, economic and environmental analyses of a CCPP are performed and the results confirmed that by adding a DB between gas cycle and steam cycle, the efficiency of the CCPP is increased. Moreover, exergy, and exergo-economic and environmental of the system has been conducted. The effects of DB fuel flow rate 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 fuel flow rate of DB the overall cost of the plant increased while the CO_2 emission of the plant is reduced.

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