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# Thermodynamics Analysis and Optimization of Abadan Combined Cycle Power Plant

**Mohammad Amin Javadi[1*] and Hossein Ghomashi[2]**

[1]Department of Mechanical Engineering, West Tehran Branch, Islamic Azad University, Tehran, Iran; benyaminjava@gmail.com
[2]School of Mechanical Engineering, Islamic Azad University, South Tehran Branch, Tehran, Iran; Ghomashi_h@azad.ac.ir

## Abstract

**Background/Objectives:** The welfare and comfort of people in the world is directly related to consuming energy and its economic supply, which depends on quantity of energy resources. This has been turned into an important challenge with respect to rapid growth at level of request for energy in the world. **Methods/Statistical analysis:** In this multi-objective optimization that has been carried out by Non-Dominated Sorting Genetic Algorithm (NSGA-II), two objective functions of exergy efficiency and produced power costs composing of the cost of injected fuel into combustion chamber and duct burner as well as exergy loss cost and investment cost have been studied. **Findings:** The efficiency of Abadan combined cycle power plant depends on design parameters including gas turbine input temperature, compressor pressure ratio, and pinch point temperature and any change occurring in these parameters may lead to noticeable change in objective functions, so that the efficiency of this power plant is increased after optimization up to 7.12 % and heat rate is correspondingly reduced from 7503 (kJ/kWh) to 7149 (kJ/kWh). Similarly, exergy destruction in total system shows 8.37 reduction. **Applications/Improvements:** In this research Abadan combined cycle power plant has been perfectly modeled in terms of thermodynamics and the given results were compared with output data from Thermo-Flow Software in order to ensure the validity of the modeling code.

**Keywords:** Abadan Combined Cycle Power Plant, Exergy Efficiency, Exergy Destruction, Optimization, NSGA-II

## 1. Introduction

The welfare and comfort of people in the world is directly related to consuming energy and its economic supply, which depends on quantity of energy resources. This has been turned into an important challenge with respect to rapid growth at level of request for energy in the world and therefore there is a necessity for optimization of energy production and energy consumption systems. Combination of Thermodynamic Second law with economic principle by taking logical approach, may provide strong tool for systematic design and optimization of complex energy systems[1].

Making effort to improve thermal efficiency of steam power plant with Rankine cycle and gas turbine power plant with Brighton cycle has led to implementation of wide corrections and modifications in these cycles. Using steam-gas combined cycle is one of the most prevalent modifications in such systems. Because of advantages of combined cycle power plants, their number has overtaken the number of other power plants. In many combined cycle power plants, Brighton cycle gas turbine is placed on top of Rankin cycle steam power plant. The combined cycle results in higher thermal efficiency and power, than each of cycles achieves separately[2]. Many researchers have employed economic exergy analysis in order to explore various types of thermal cycles. Have utilized this concept to evaluate a solar-thermal power plant[3]. Likewise, this concept has been employed for study on Proton Exchange Membrane Fuel Cell (PEMFC)[4], Ground source heat pumps[5], Combined cycle power plants[6], Heat exchange networks[7], Geothermal systems[8], Thermo-chemical cycles for hydrogen production[9], Multi-productive systems with gas-diesel engine[10],transportation sector[11]

* Author for correspondence

and CGAM co-generation system[12]. Kotas examined reduction of exergy for different thermal systems and approximated rate of exergy loss in different systems[13]. showed in their analysis on a semi-closed cycle of gas turbine, that the foremost source of wasting exergy of a system, is recovery and injection of water, so that 80 % of irreversibility of cycle is completely caused by that. At the same time, they explored the function of second law for a cycle, when no steam is injected into the given cycle, until the perfect injection is done and their results indicated, that the maximum rates of irreversibility occurred during steam perfect injection into the cycle[14]. explored thermo-economic optimization of operating parameters of heat recovery boiler, by means of reduced exergy loss. In this analysis, pinch point method has not been adapted and factor of gas performance has been assumed as basis in heat recovery boiler[15]. Have examined Rankine cycle re-heat steam power plant in order to study the efficiency of exergy and its energy under various operating conditions with change in temperature and pressure of boiler and the output work from the cycle[16].

Conducted analysis of exergy and energy for a combined cycle in Turkey. Similarly, they suggested some modifications to reduce exergy destruction in the combined cycle power plant. The results showed that combustion chamber, gas turbine, and Heat Recovery Steam Generator (HRSG) are considered as the main sources of irreversibility that 85 % of total exergy loss is caused by them[17]. carried out energy, exergy, and exergo-economic analysis on a steam power plant in Iran. Likewise, they considered changes in loadings and ambient temperature in order to find exergy destruction in any element of the cycle. The results indicated that energy losses are 67.63 MW for boiler. Nonetheless, the rate of irreversibility in the boiler is higher than the rate of irreversibility in other components of the system[18].

Sahoo conducted exergoeconomic analysis for thermal co-generation system with 50 MW power and optimized it using evolutionary algorithm. The resulting findings from his investigation at optimum case in exergo-economic analysis have shown 9.9 % reduction in cost versus system basic cost[19]. have assessed the environmental exergy for a gas turbine power plant in which the results show that rising exergy efficiency may reduce $CO_2$ emission[20]. There are numerous techniques and approaches in thermo-economic analysis, including theory of exergetic cost (TEC)[21], theory of exergetic distinct cost (TECD)[21, 22], thermo-economic functional analysis (TFA)[23], intelligent functional analysis (IFA)[24], principle of last-input first-out (LIFO)[25], specific exergy costing approach (SPECO)[26], and engineering functional analysis (EFA)[27,28]. The SPECO approach is utilized in the presented study. This study comprises of three major parts. In the first part of this study, the rate of exergy cost

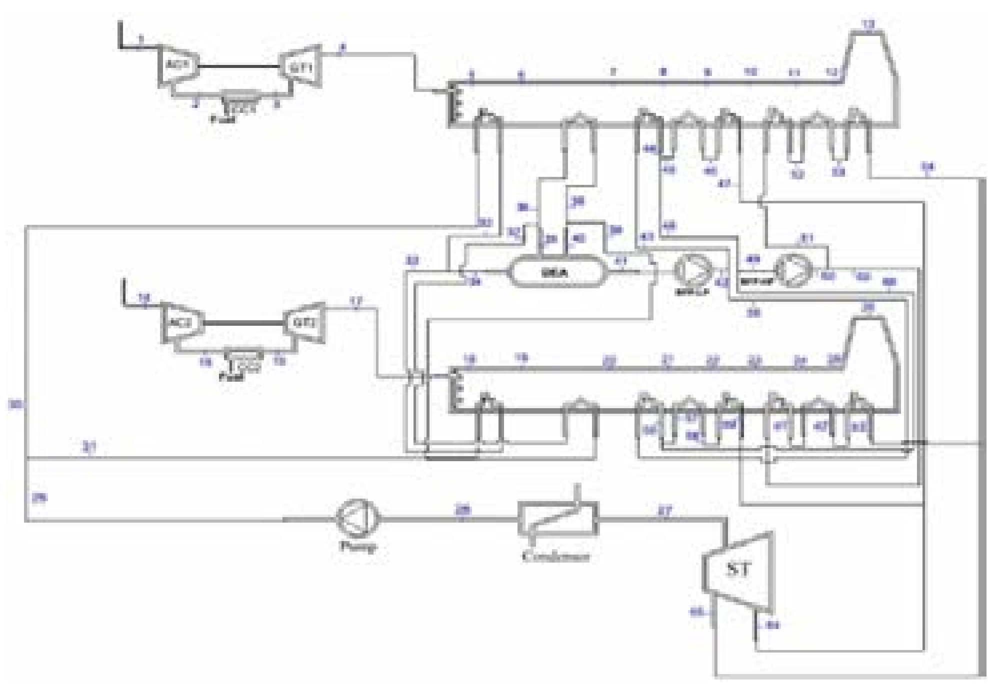


**Figure 1.** Schematic diagram of Abadan combined cycle power plant

**Table 1.**

| Nomenclature | | | |
|---|---|---|---|
| Greek letters | | Cost per exergy unit($/MJ) | c |
| Isentropic efficiency | η | Cost of fuel per energy unit($/MJ) | cf |
| Coefficient of fuel chemical exergy | ξ | Cost flow rate($/s) | Ć |
| Maintenance factor | Φ | Specific heat at constant pressure(kj/kg K) | $c_p$ |
| Subscripts | | Capital recovery factor | CRF |
| air | α | Exergy flow rate(MW) | Ėx |
| air compressor | AC | Exergy destruction rate(MW) | $\dot{E}_{xD}$ |
| Combustion chamber | CC | Specific exergy(kj/kg) | ex |
| chemical | ch | Annual interest rate (%) | i |
| condenser | Cond | Specific enthalpy(kj/kg) | h |
| gas turbine | GT | Specific enthalpy at environmental state(kj/kg) | $h_0$ |
| high pressure | HP | Lower heating value(kj/kg) | LHV |
| heat recovery steam generator | HRSG | Mass flow rate (kg/s) | m |
| Steam turbine | ST | Number of years | n |
| Low pressure | LP | Number of hours of plant operation per year | N |
| physical | ph | Pinch point | Pp |
| water | w | Heat transfer rate(kW) | Q |
| | | Compressor pressure ratio | rAC |
| | | Specific entropy (kj/kg K) | s |
| | | Specific entropy at environmental state (kj/kg K) | $S_0$ |
| | | Net power output (MW) | $W_{net}$ |
| | | Capital cost of a component ($) | Z |
| | | Capital cost rate ($/s) | Ž |

is calculated in each of stream using SPECO approach. The performance of this system is optimized according to cost function and exergy efficiency in second part of this survey and finally the impact of effective parameters on system performance is separately examined.

## 2. Characteristics of Combined Cycle Power Plant

In this study, a combined cycle power plant with power capacity of 450 MW has been explored according to Figure1. Subsystems of this combined power plant are composed of two gas units and one double- pressure steam unit. The model of gas turbine that is used in this power plant is Siemens V94.2 and it's steam turbine is Siemens E-Type.

## 3. Assumption

In order to model this combined cycle power plant, the following assumptions have been considered[29].

- All processes in this study have been considered as steady state and steady flow.
- The air and output gases from combustion chamber have been assumed as ideal gases.
- The kinetic and potential exergies have been ignored.
- The dead state in this study was considered and.
- The rate of pressure loss has been considered as equivalent to 0.03 in combustion chamber.
- Turbine, compressor, and pump were assumed as adiabatic.
- The ambient temperature and pressures were considered as input conditions in compressor.
- Methane was assumed as the consumed fuel in this modeling.

## 4. Thermodynamic Modeling

Exergo-economic analysis denotes the relevant cost for each of exergy streams. Therefore, in order to conduct exergoe-conomic analysis, the rate of exergy should be identified at different points in each of input and output streams. The rate of exergy stream is determined at different points of power plant with application of mass, energy, and exergy balance relations.

### 4.1 Mass, Energy and Exergy Balance Relations

Mass, energy and exergy balance relations could be calculated by considering a control volume for each component of the power plant by means of the following equations respectively: [15], [30]

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \quad (1)$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{out}\, h_{out} - \sum \dot{m}_{in}\, h_{in} \quad (2)$$

$$E\dot{x}_Q - \sum_{in} \dot{m}_{in}\, ex_{in} = \sum_{out} \dot{m}_{out}\, ex_{out} + E\dot{x}_w + E\dot{x}_D \quad (3)$$

Where, in Eq. (3), denotes the exergy destruction. Similarly, in this equation and are the exergy rates associated with work and heat transfer which at temperature $T$ is calculated by

$$E\dot{x}_W = \dot{W} \quad (4)$$

$$E\dot{x}_Q = \sum\left(1 - \frac{To}{T}\right)\dot{Q} \quad (5)$$

The exergy in each of the streams has been shown in Figure (1), are derived by following formulae [13,18,31-35]:

$$E\dot{x} = \dot{m}\, ex \quad (6)$$

$$E\dot{x} = E\dot{x}_{ph} + E\dot{x}_{ch} \quad (7)$$

$$E\dot{x}_{ph} = (h - h_0) - T_0(s - s_0) \quad (8)$$

$$E\dot{x}_{ph} = \left[\sum_{i=1}^{n} X_i ex^{ch_i} + RT_0 \sum_{i=1}^{n} X_i LnX_i\right] \quad (9)$$

To evaluate the fuel exergy, the above equation cannot be used. Therefore, fuel exergy is extracted from the following formula [13]:

$$\xi = \frac{ex_f}{LHV_f} \quad (10)$$

For the most of gaseous fuels, the ratio of chemical exergy to LHV is usually close to unit (1):

$$\xi_{CH_4} = 1.06 \quad (11)$$

For carbohydrate gaseous fuels(), the following empirical formula is used for calculation[36]:

$$\xi = 1.033 + 0.0169\frac{y}{x} = \frac{0.0698}{x} \quad (12)$$

In this study, in order to conduct exergy analysis in power plant, the rate of exergy for each of lines is computed and the changes in the exergy defined for the power plant's components.

### 4.2 Calculation of Capital Cost of Power Plant Components

The thermo-economic calculations of the system are done according to capital costs of its components. Numerous techniques have been proposed to determine equipment purchase cost based on terms of design parameters. Here, the cost function, which has been suggested is utilized. However, in order to achieve the regional conditions in Iran, some adjustments have been implemented. To convert investment cost into the cost per time unit, we have:

$$\dot{Z}_k = Z_k.CRF.\frac{\Phi}{(N \times 3600)} \quad (13)$$

Where, $Z_k$ is the purchase cost of kth equipment in US dollars. In this formula, the capital recovery Factor (CRF) depends on interest rate and approximated useful lifetime for components. CRF is calculated according to the formula:

$$CRF.\frac{i(1+i)^n}{(1+i)^n - 1} \quad (14)$$

Where, $i$denotes interest rate and $n$ expresses number of years for operating of the system[37].

In Eq. (13), $N$ is the number of hours for operation of power plant during a year, and $\Phi$ is maintenance factor that they have been set as 7446 and 1.06, respectively.

### 4.3 Cost Balance Relations based on SPECO

To calculate exergy costing of streams, cost balance realtions are separately written for each component of power plant. There are many thermo-economic approaches in this regard. The specific exergy costing (SPECO) technique is employed in this study[1,26]. This method is based on specific exergies and costs per exergy unit and auxiliary equations of cost for components of thermal systems. This technique includes three steps as follows: (1) Identifying exergy stream; (2) Definition of fuel and product for each of component of thermal system; and (3) Separate formulation of cost equation for each component of the power plant.

The related cost rates to exergy transfers by input and output streams and by rate of power and heat transfer is written respectively as follows:

$$\dot{C}_{in} = c_{in}\dot{E}x_{in} = c_{in}(\dot{m}_{in}\, ex_{in}) \quad (15)$$

$$\dot{C}_{our} = c_{out}\dot{E}x_{out} = c_{out}(\dot{m}_{out}\, ex_{out}) \quad (16)$$

$$\dot{C}_{W} = c_{w}\dot{W} \quad (17)$$

$$\dot{C}_{neat} = c_{neat}\dot{E}x_{neat} \quad (18)$$

Where, $c_{in}$, $c_{out}$, $c_w$, and $c_{heat}$ show mean costs per exergy unit.

Accordingly, the cost balance relation is written for kth component of power plant based on the following formula:

$$\sum(c_{in}\dot{E}x_{in})_k + c_{w,k}W_k = c_{neat,k}\dot{E}x_{neat} + \sum(c_{in}\dot{E}x_{in})_k + \dot{Z}_k \quad (19)$$

In the above formula, when a component receives the work (e.g. in compressor and/ or pump), the second term of relation on the left side would move to the right side of this relation[26].

Using cost balance formulas and auxiliary equations separately for each component, results in a system of linear equations that are led to their simultaneous solution and cost of each stream will be obtained. Therefore, cost balance and auxiliary equations are given in Table 2 based on SPECO method for different components in Abadan combined cycle power plant under the expressed conditions.

## 4.4 Calculation of Exergy Destruction Cost in the Cycle

In this section of analysis, two concepts of fuel and product are defined. Exergy product is defined with respect to the given consideration ofcomponents. Fuel is a source that may be used in generating this product. Product and fuel are expressed as some terms of exergy. The related cost ratesof fuel and product of different components of power plants are calculated with replacement of exergyrate .

In the cost balance equation (20) for a component there is no cost term directly associated with exergy destruction that is directly related to it. Accordingly, the related cost to exergy destruction in a comonent or process is ahidden cost,but a very important one, that can be appear only through a thermo-economic analysis[1].

The effect of exergy destruction will appear with combination of exergy balance equation (20) and economic balanceequation (21) together:

$$\dot{E}x_{F,k} = \dot{E}x_{p,k} + \dot{E}x_{L,k} + \dot{E}x_{D,k} \quad (20)$$

$$c_{p,k}\dot{E}x_{p,k} = c_{F,k}\dot{E}x_{F,k} + \dot{C}_{L,k} + \dot{Z}_k \quad (21)$$

to eliminatefrom the above equations the following formula

$$c_{p,k}\dot{E}x_{p,k} = c_{F,k}\dot{E}x_{p,k} + (c_{F,k}\dot{E}x_{L,k} = \dot{C}_{L,k}) + \dot{Z}_k + c_{F,k}\dot{E}x_{D,k} \quad (22)$$

and to eliminate the following formula will result in:

$$c_{P,k}\dot{E}x_{F,k} = c_{F,k}\dot{E}x_{F,k} + (c_{P,k}\dot{E}x_{L,k} = \dot{C}_{L,k}) + \dot{Z}_k + c_{P,k}\dot{E}x_{D,k} \quad (23)$$

The last term on the right side of Equations (22) and (23) involves the rate of exergy destruction. As it was already discussed, if rate of productexergy is assumed as fixed and the unit cost of fuelof the kth component is independent from the exergy destruction, cost of exergy destruction is defined by last term in Eq. (22)[1].

$$\dot{C}_{D,k} = c_{F,k}\dot{E}x_{D,k} \quad (24)$$

## 4.5 Definition of Objective Functions

In order to optimize variables in the combined cycle power plant in this study, multi-objective optimization technique is employed. Therefore, to achieve this goal, two different objective functions are defined. The first objective function is related to exergy efficiency *CCPP* that is derived by total net power output of the plant dividing by fuel exergy according to the following formula:

$$\eta_{exergy} = \frac{\sum \dot{W}_{net}}{\dot{E}x_f} \quad (25)$$

And the second objective function is composed of capital cost of component, cost of fuel consumed in combustion chamber and duct burner, and the cost associatedwith theexergy destruction.

$$\dot{C}_{Total} = C_f\, \dot{m}_f LHV + \sum \dot{Z}_i + \sum \dot{C}_{D,i} \quad (26)$$

**Table 2.** Cost balance and auxiliary equations for different components of the Abadan plant

| Component | Cost Balance Equation | Auxiliary equation |
|---|---|---|
| Air compressor 1 | $(c_2\dot{E}x_2 - c_1\dot{E}x_1) = c_{w,BC1}\dot{W}_{AC1} + \dot{Z}_{AC1}$ | $c_1 = 0$ |
| Combustion chamber 1 | $c_3\dot{E}x_3 = c_2\dot{E}x_2 + \dot{Z}_{CC1} + \dot{C}_{f-CC1}$ | No auxiliary equation |
| Gas turbine 1 | $c_{w,BC1}\dot{W}_{GT1} = (c_3\dot{E}x_3 - c_4\dot{E}x_4) + \dot{Z}_{GT1}$ | $c_3 = c_4$ |
| HRSG1 | | $\frac{c_{32}\dot{E}x_{32} - c_{30}\dot{E}x_{30}}{\dot{E}x_{32} - \dot{E}x_{30}} = \frac{c_{38}\dot{E}x_{38} - c_{36}\dot{E}x_{36}}{\dot{E}x_{38} - \dot{E}x_{36}}$<br>$\frac{c_{38}\dot{E}x_{38} - c_{36}\dot{E}x_{36}}{\dot{E}x_{38} - \dot{E}x_{36}} = \frac{c_{44}\dot{E}x_{44} - c_{43}\dot{E}x_{43}}{\dot{E}x_{44} - \dot{E}x_{43}}$<br>$\frac{c_{44}\dot{E}x_{44} - c_{43}\dot{E}x_{43}}{\dot{E}x_{44} - \dot{E}x_{43}} = \frac{c_{46}\dot{E}x_{46} - c_{45}\dot{E}x_{45}}{\dot{E}x_{46} - \dot{E}x_{45}}$<br>$\frac{c_{46}\dot{E}x_{46} - c_{45}\dot{E}x_{45}}{\dot{E}x_{46} - \dot{E}x_{45}} = \frac{c_{47}\dot{E}x_{47} - c_{46}\dot{E}x_{46}}{\dot{E}x_{47} - \dot{E}x_{46}}$<br>$\frac{c_{47}\dot{E}x_{47} - c_{46}\dot{E}x_{46}}{\dot{E}x_{47} - \dot{E}x_{46}} = \frac{c_{52}\dot{E}x_{52} - c_{51}\dot{E}x_{51}}{\dot{E}x_{51} - \dot{E}x_{51}}$<br>$\frac{c_{52}\dot{E}x_{52} - c_{51}\dot{E}x_{51}}{\dot{E}x_{52} - \dot{E}x_{51}} = \frac{c_{53}\dot{E}x_{53} - c_{52}\dot{E}x_{52}}{\dot{E}x_{53} - \dot{E}x_{52}}$<br>$\frac{c_{53}\dot{E}x_{53} - c_{52}\dot{E}x_{52}}{\dot{E}x_{53} - \dot{E}x_{52}} = \frac{c_{54}\dot{E}x_{54} - c_{53}\dot{E}x_{53}}{\dot{E}x_{54} - \dot{E}x_{53}}$<br>$c_{12} = c_4$ |
| Air compressor 2 | $(c_{15}\dot{E}x_{15} - c_{14}\dot{E}x_{14}) = c_{w,BC2}\dot{W}_{AC2} + \dot{Z}_{AC2}$ | $c_{14} = 0$ |
| Combustion chamber 2 | $c_{16}\dot{E}x_{16} = c_{15}\dot{E}x_{15} + \dot{Z}_{CC2} + \dot{C}_{f-CC2}$ | No auxiliary equation |
| Gas turbine 2 | $c_{w,BC2}\dot{W}_{GT2} = (c_{16}\dot{E}x_{16} - c_{17}\dot{E}x_{17}) + \dot{Z}_{GT2}$ | $c_{16} = c_{17}$ |
| HRSG2 | $c_{62}\dot{E}$: | $\frac{c_{38}\dot{E}x_{38} - c_{31}\dot{E}x_{31}}{\dot{E}x_{38} - \dot{E}x_{31}} = \frac{(c_{39}\dot{E}x_{39} - c_{37}X\dot{E}x_{37})}{\dot{E}x_{39} - \dot{E}x_{37}}$<br>$\frac{(c_{39}\dot{E}x_{39} - c_{37}\dot{E}x_{37})}{\dot{E}x_{39} - \dot{E}x_{37}} = \frac{c_{56}\dot{E}x_{56} - c_{55}\dot{E}x_{55}}{\dot{E}x_{56} - \dot{E}x_{55}}$<br>$\frac{c_{56}\dot{E}x_{56} - c_{55}\dot{E}x_{55}}{\dot{E}x_{56} - \dot{E}x_{55}} = \frac{c_{58}\dot{E}x_{58} - c_{57}\dot{E}x_{57}}{\dot{E}x_{58} - \dot{E}x_{57}}$<br>$\frac{c_{58}\dot{E}x_{58} - c_{57}\dot{E}x_{57}}{\dot{E}x_{58} - \dot{E}x_{57}} = \frac{c_{59}\dot{E}x_{59} - c_{58}\dot{E}x_{58}}{\dot{E}x_{59} - \dot{E}x_{58}}$<br>$\frac{c_{59}\dot{E}x_{59} - c_{58}\dot{E}x_{58}}{\dot{E}x_{59} - \dot{E}x_{58}} = \frac{c_{61}\dot{E}x_{61} - c_{60}\dot{E}x_{60}}{\dot{E}x_{61} - \dot{E}x_{60}}$<br>$\frac{c_{61}\dot{E}x_{61} - c_{60}\dot{E}x_{60}}{\dot{E}x_{61} - \dot{E}x_{60}} = \frac{c_{62}\dot{E}x_{62} - c_{61}\dot{E}x_{61}}{\dot{E}x_{62} - \dot{E}x_{61}}$<br>$\frac{c_{62}\dot{E}x_{62} - c_{61}\dot{E}x_{61}}{\dot{E}x_{62} - \dot{E}x_{61}} = \frac{c_{63}\dot{E}x_{63} - c_{62}\dot{E}x_{62}}{\dot{E}x_{63} - \dot{E}x_{62}}$<br>$c_{17} = c_{26}$ |

| | | |
|---|---|---|
| Steam Turbine | $c_{w,RC}\dot{W}_{ST} = (c_{65}\dot{E}x_{65} + c_{64}\dot{E}x_{64} - c_{27}\dot{E}x_{27}) + \dot{Z}_{ST}$ | $c_{65} = c_{64} = c_{27}$ |
| Steam Condenser | $c_{28}\dot{E}x_{28} = c_{27}\dot{E}x_{27} + \dot{Z}_{Cond}$ | No auxiliary equation |
| Condensate extraction pump | $(c_{29}\dot{E}x_{29} - c_{28}\dot{E}x_{28}) = c_{w,RC}\dot{W}_{CEP} + \dot{Z}_{CEP}$ | No auxiliary equation |
| Deaerator and BFP-LP | $c_{34}\dot{E}x_{34} + c_{40}\dot{E}x_{40} + \dot{Z}_{Dea,BFP-LP} = c_{35}\dot{E}x_{35} + c_{42}\dot{E}x_{42}$ | No auxiliary equation |
| High pressur pump | $(c_{50}\dot{E}x_{50} - c_{49}\dot{E}x_{49}) = c_{w,RC}\dot{W}_{BFP-HP} + \dot{Z}_{BFP-HP}$ | No auxiliary equation |
| Mixing of Streams 32 and 33 | $c_{34}\dot{E}x_{34} = c_{32}\dot{E}x_{32} + c_{33}\dot{E}x_{33}$ | No auxiliary equation |
| Mixing of Streams 38 and 39 | $c_{40}\dot{E}x_{40} = c_{38}\dot{E}x_{38} + c_{39}\dot{E}x_{39}$ | No auxiliary equation |
| Mixing of Streams 47 and 59 | $c_{64}\dot{E}x_{64} = c_{47}\dot{E}x_{47} + c_{59}\dot{E}x_{59}$ | No auxiliary equation |
| Mixing of Streams 63 and 54 | $c_{65}\dot{E}x_{65} = c_{54}\dot{E}x_{54} + c_{63}\dot{E}x_{63}$ | No auxiliary equation |
| Splitting of stream 29 | $c_{30}\dot{E}x_{30} + c_{31}\dot{E}x_{31} = c_{29}\dot{E}x_{29}$ | $c_{30} = c_{31}$ |
| Splitting of stream 35 | $c_{36}\dot{E}x_{36} + c_{37}\dot{E}x_{37} = c_{35}\dot{E}x_{35}$ | $c_{36} = c_{37}$ |
| Splitting of stream 42 | $c_{43}\dot{E}x_{43} + c_{55}\dot{E}x_{55} = c_{42}\dot{E}x_{42}$ | $c_{43} = c_{55}$ |
| Splitting of stream 44 | $c_{45}\dot{E}x_{45} + c_{48}\dot{E}x_{48} = c_{44}\dot{E}x_{44}$ | $c_{45} = c_{48}$ |
| Splitting of stream 48 | $c_{49}\dot{E}x_{49} + c_{66}\dot{E}x_{66} = c_{48}\dot{E}x_{48}$ | $c_{49} = c_{66}$ |
| Splitting of stream 66 | $c_{56}\dot{E}x_{56} + c_{57}\dot{E}x_{57} = c_{66}\dot{E}x_{66}$ | $c_{56} = c_{57}$ |

Maximization of exergy efficiency and minimization of rate of total cost are considered in multi-objective optimization. It is clear that as exergy efficiency is improved in cycle, total cost of cycle will be increased.

## 4.6 Decision Variables and Constraints

12 variables have been selected as decision variables for multi-objective optimization in the present research as follows: Compressor pressure ratio ($r_{AC}$), compressor isentropic efficiency ($\eta_{AC}$),gas turbine isentropic efficiency (($\eta_{GT}$),gas turbine inlet temperature (*GTIT*), high pressure pinch point temperature ($PP_{HP}$), low pressure pinch point temperature ($PP_{LP}$), condenser pressure ($P_{Cond}$), pump isentropic efficiency ($\eta_p$),steam turbine isentropic efficiency ($\eta_{ST}$), inlet temperature at high- pressure steam turbine ($HP_{Temp}$), inlet temperature at low pressure steam turbine ($LP_{Temp}$), and fuel flow ofduct burner ($\dot{m}_{db}$). Although decision variables may change for optimization, these changesshould be in a reasonable range. Some constraints are applied to thermodynamic variables at various points of power plant, which are listed in Table 3.

**Table 3.** Optimization constraints and their rationales[39]

| Constraints | Reason |
|---|---|
| $GTIT < 1550K$ | Material Temperature Limit |
| $r_{AC} < 22$ | Commercial Availability |
| $\eta_{AC} < 0.9$ | Commercial Availability |
| $\eta_{GT} < 0.9$ | Commercial Availability |
| $5°C < PP_{HP}$ $5°C < PP_{HP}, PP_{LP} < 30℃$ $PP_{LP} < 30℃$ | Heat Transfer Limit |
| $5\ bar < P_{Cond} < 15\ bar$ | Thermal Efficiency Limit |
| $0.75 < \eta_p < 0.9$ | Commercial Availability |
| $0.75 < \eta_{ST} < 0.9$ | Commercial Availability |
| $\dot{m}_{db} < 2\left({}^{kg}/_{s}\right)$ | Super Heater Temperature Limitation |
| $T_{12}, T_{26} \geq 120℃$ | To Avoid Formation Of Sulfuric Acid In Exhaust Gases |
| $P_{main} < 110\ bar$ | Commercial Availability |

## 4.7 Genetic Algorithm (GA)

Technique of genetic algorithm is employed for multi-objective optimization in this survey. This algorithm utilizes an iterative strategy and random searching for finding of an optimal response based on simple biologic principles. The first multi-objective GA algorithm, which was suggested by Schaffer, was called Vector evaluated GA. The algorithm that operated according to non- dominated sorting and proposed by Srinivas and Deb was called Non- Dominated Sorting Genetic Algorithm (NSGA). Then, Suggested a new algorithm with eliminating higher computational complexity, lack of elitism and requirement for specifying the sharing parameter that was called NSGA-II. This algorithm will be supposed as basis for optimization in this study[38].

# 5. Results and Discussion

## 5.1 Results of Optimization

The multi-objective optimization is done by applying the constraints listed in Table 2 and using the expressed formulas in the previous section for each power plant component.Figure 2shows the optimizations results from two objective functions of exergy efficiency and total cost of produced electricity in the combined cycle power plant on the Pareto curve. As it characterized in this Figure, the cost of produced electricity will be significantly increased by improving exergy efficiency from 47% to 49%. In fact, the maximum rate of exergy efficiency is at C-point (49%). At this point we will have the highest total cost for the produced electricity (7.98$/s).

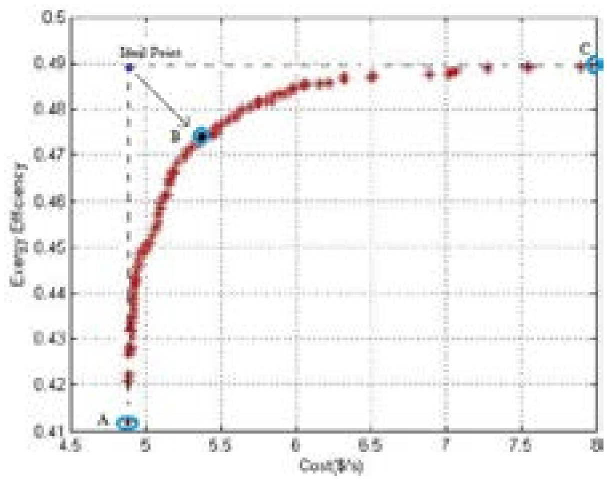


**Figure 2.** Pareto frontier: best tradeoff for the objective functions

The decision-making process is necessary in multi-objective optimization to select the final optimal solution among the availablesolutions. This is usually executed by the aid of equilibrium point in Figure 2 named as ideal point that both objectivehave their optimum values independent from the other objective functions. It is clear that synchronous access of two objective functions to their optimum values is impossible and as it characterized in Figure (2), the equilibrium point is not placed on Pareto frontier as well. The closest point to the equilibrium point in Pareto frontier can be supposed as the final solution. Nonetheless, the Pareto's optimal frontier possesses weak equilibrium; in other words, rate of produced electricity will highly change with a small variation in exergy efficiency. Thus, the equilibrium point in this problem may not be used for achieving decision-making process. In fact, each point can be used as optimization point in multi-objective optimization and Pareto solution. Therefore, depending on preferences and criteria taken by decision-maker, the optimum solution may be selected in different points. The decision variables of the combined cycle power plant can be achieved by considering objective functions and constraintsand applying thegenetic algorithm code of for this problem. The values of these decision variables after optimization are listed in Table 4.

**Table 4.** Optimized values for decision variables of the system for three points on the pareto frontier from multi-objective optimization

| Decision variable | Value |
|---|---|
| $r_{AC}$ | 10.96 |
| $\eta_{AC}$(%) | 0.912 |
| $\eta_{GT}$(%) | 0.83 |
| GTIT (K) | 1422.15 |
| $PP_{LP}$ (K) | 44 |
| $PP_{LP}$ (K) | 58 |
| $m_{db}$(kg/s) | 1.005 |
| $T_{HP}$ (K) | 793.15 |
| $T_{LP}$ (K) | 501.15 |
| $P_{cond}$ (Kpa) | 15.7 |
| $\eta_{pump}$ (%) | 0.84 |
| $\eta_{ST}$(%) | 0.85 |

## 5.2 Sensitivity Analysis

The sensitivity analysis is implemented for better understandingthe effect of variation of decision variables on the objective functions. Figure 3 shows the variations with compressor pressure ratio of combustion chamber mass flow rate. As it is seen in this Figure, the amount of consumed fuel has decreasedas the compressor pressure ratio increased. In fact, with risingthe compressor pressure ratio, the combustion chamber inlet temperature is increased and as a result, there is less mass flow rate of fuel in combustion chamber needed to increase outlet gas temperature in it. Figure 4 shows the variation with compressor pressure ratio of exergy efficiencies for the combined cycle power plant.

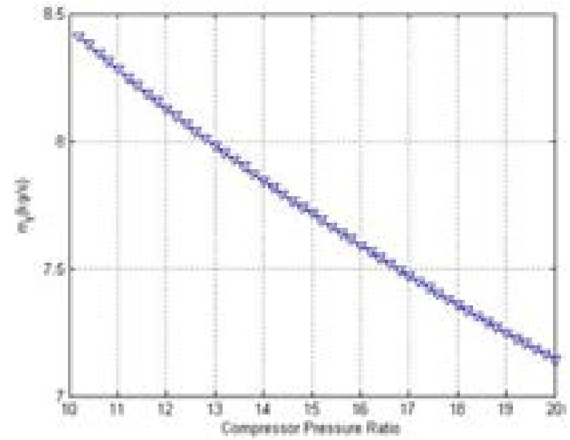


**Figure 3.** Variations with compressor pressure ratio of combustion chamber mass flow rate.

As it is evident in this Figurer, the exergy efficiency of the system is improved as compressor pressure ratio is increased and this is due to rising combustion chamber inlet temperature by increasing compressor pressure ratio and consequently reduced consuming fuel. In fact, the exergy efficiency can be improved with increased compressor pressure ratio; however, given that it needs to further work to compress this amount of air flow, total cost of produced electricity will be increased.

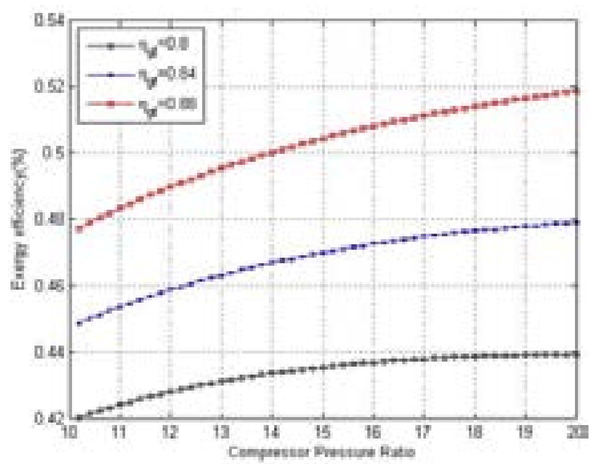


**Figure 4.** Variations with compressor pressure ratio of the combine cycle exergy efficiency for several gas turbine isentropic efficiencies.

Similarly, it is seen in Figure 4 that the exergy efficiency has a initial sharp increase. This is because at a lower pressure ratios, increasing the pressure ratio will led to an increase of the compressor outlet temperature and decrease of the mass flow rate of the fuel injected to combustion chamber. This will according to definition of the exergy efficiency, be the reason for increase in the exergyefficiency. From certain pressure ratio, as pressure ratio is improved, the rate of increase in rate of work received by compressor, becomes higher than rate of reduction of fuel and this causes reduction in fast growth rate of exergy efficiency. Figure 5shows variations with compressor pressure ratio of exergy efficiency and exergy destruction rate for the combined cycle power plant.

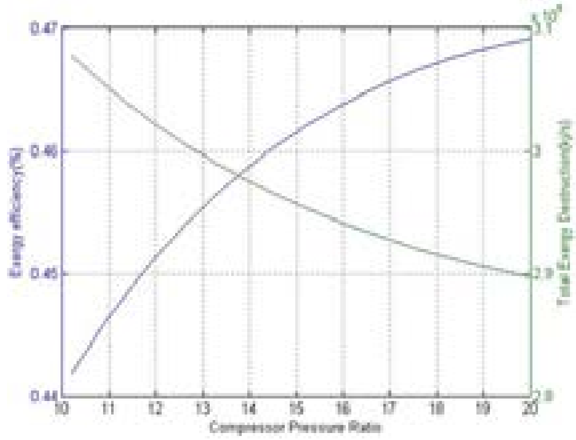


**Figure 5.** Variations with compressor pressure ratio of exergy efficiency and exergy destruction rate.

It is observed in this Figure that due to reduction in consuming fuel, as compressor pressure ratio is increased, the exergy efficiency is improved and consequently rate of exergydestruction is reduced.

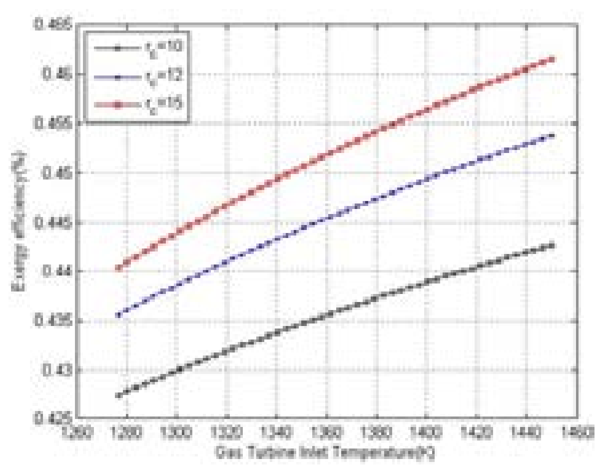


**Figure 6.** Variation of gas turbine inlet temperature and compressor ratio on exergy efficiency.

Figure 6 indicates variation of exergy efficiency subsequently after rising gas turbine inlet temperature and compressor pressure ratio. At the fixed compressor pressure ratio, rising gas turbine inlet temperature may be followed by improving of exergy efficiency due to increase in output power of that turbine. Similarly, as it has been already implied, in this Figure we can observe that the exergy efficiency is improved with the increase in pressure ratio. Figure 7 shows the variation oftotal cost of exergy destruction at power plant following to rising gas turbine inlet temperature.

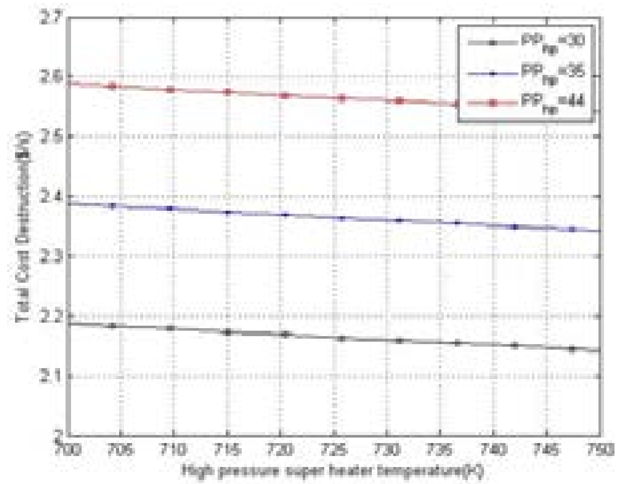


**Figure 7.** Variation of HRSG high pressure super heater and high temperature Pinch Point temperature on total cost of exergy destruction.

According to this Figure, rising gas turbine inlet temperature atfixed pressureratio will be followed by lower cost for loss of total exergy in system.In the following the effect of high-pressure super-heater temperature is examined on objective functions. Figure 7 indicates changes in total cost of exergy destruction during rising temperature at super-heater. Given that as temperature of high- pressure super-heater is increased, the output power is improved in Rankin cycle so that rate of exergy destruction will be reduced. Thus, according to this Figure, the cost of exergydestruction is reduced with rising this temperature.

Likewise, as it identified in Figure 7, if temperature of super-heater is supposed as fixed, as pinch point temperature is increased, the rate of total cost of exergy destruction in system will be increased.

## 6. Conclusion

The present research has provided for determining way of using well-known method of SPECO thermodynamic analysis to evaluate the combined cycle power plant. The cost balance formulae and auxiliary equations have been written for each components of power plant. The mean cost for eachexergy unit was computed for different points by solving the coordinates of equations in MATLAB software. Similarly, the values of economic exergy values have been calculated for each component. The multi-objective optimization was done for this power plant by means of NSGA-II algorithm so that by application of the resulting decision variables from this optimization, the rate of efficiency of this power plant has been improved up to 7.12%. The effect of decision variables such as compressor pressure ratio, gas turbine inlet temperature, pinch point temperature, and temperature of high pressure super-heater was examined on two objective functions. The following results are obtained from these studies:

- With rising compressor pressure ratio, the rate of exergy efficiency is increased in the combined cycle and this rising trend firstly become further with higher gradient and this slop is gradually decreased. Of course, it is noteworthy that the rate of total loss of exergy will be reduced in this cycle as exergy efficiency is improved.
- The exergy efficiency of the combined cycle power plant will be improved with rising isentropic efficiency of gas turbine.
- The exergy efficiency of the combined cycle power plant will be improved as gas turbine inlet temperature is increased.
- The total cost of exergy destruction of system is increased with rising of pinch point temperature.
- The total cost of exergy destruction of system will be reduced as temperature is increased in high pressure super-heater.

Similarly with respect to the aforesaid issues, it can be concluded that SPECO economic exergy analytical technique will be deemed as an efficient tool to identify and evaluate defects and ineffectiveness in terms of cost and efficiency and the methods and formulae used in this study are not exclusively limited to thermal systems, but also they can be used in other systems.

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