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Exergoeconomic and Sustainability Analysis of Reheat Gas Turbine Engine

Abdulrahman Almutairi[1,*], Pericles Pilidis[1], Nawaf Al-Mutawa[2]

[1]Power and Propulsion Department, School of Aerospace, Transport and Manufacturing Cranfield University, Cranfield, Bedfordshire MK43 0AL, UK
[2]Mechanical Engineering Department, College of Engineering and Petroleum, Kuwait University, Al Asimah, P.O. Box 5969 Safat 13060, Khalidiya 72301, Kuwait
*Corresponding author: a.s.almutairi@cranfield.ac.uk

Abstract Exergoeconomic and sustainability analyses have been performed for a heavy duty industrial reheat gas turbine engine. The proposed system was inspired by a GT26, Alstom advance-class gas turbine with a unique design modification based on the reheat principle using two sequential combustion chambers. The IPSEpro software package was used for validating the process and results tested against the manufacturer's published data. Energy system performance is usually evaluated through energetic or exergetic criteria. The latter has the advantage of determining energy degradation and quantifying the deficiencies within a system as well as recognizing loss sources and types. The cost-effectiveness of using this gas turbine engine has been evaluated using exergoeconomic approach: the Specific Exergy Costing [SPECO] method. The sustainability of the proposed model was estimated using a generic combustor model, HEPHAESTUS, to appraise the emissions impact. The performance of gas turbine engines has been investigated for different load demand and climatic conditions using two configurations. The first system, Case-I, was a simple gas turbine (SCGT) engine, and the second, Case-II, a reheat gas turbine (RHGT) system. The reheat system boosted power output in RGHT, at the same time, reducing exergetic efficiency because of greater fuel consumption. Operating both systems at low ambient temperature is preferable and full load reduces waste exergy. The production cost on an exergy basis demonstrates that the RHGT has a lower value at 7.58 US$/GJ while the SCGT produces energy at 7.77 US$/GJ. From a sustainability perspective, the SCGT shows lower emission levels and has lower environmental impact than the RHGT.

Keywords: *reheat, exergoeconomic, sustainability, SPECO, irreversibilities*

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1. Introduction

Today the gas turbine engine is the first choice for power applications or as primary movers for multi-generation systems. Adequate power generation is essential for sustainable economic growth and the demand for electricity is increasing rapidly due to population growth, economical activities and changing life styles. Gas turbine power plants have the advantages of a short installation period, high power to space ratio and relatively low environmental impact. For the longer term, the depletion of a non-renewable resource is a motivation to improve the energy efficiency of the gas turbine.

Exergy analysis is powerful tools that can be used to evaluate energy systems with the advantages of determining energy loss locations, types and magnitudes. Exergy is defined as the maximum amount of useful work that can be extracted from a system as it moves to an equilibrium state [1]. The exergoeconomic approach combines the concepts of exergy and economics to assess the energy system's cost-effectiveness. The deficiencies of a system can be measured, appraised and, therefore, reduced either during the design or operation stage. The term exergoeconomic was initially suggested by Tsatsaronis [2]. Many researchers have conducted exergoeconomic analyses of different thermal applications [3,4,5,6]. Wang et al. [7] introduced an exergoeconomic analysis to an ultra-supercritical power plant located in China using the SPECO method and proposed improvements to the plant's cost-effectiveness. Ameri et al. [8] conduct an exergoeconomic analysis of the Hamedan steam power plant. According to their study, the highest cost of exergy destruction took place in the boiler followed by the turbine. Altuntas et al. [9] analyzed a piston-prop aircraft engines using exergoeconomic analysis. They found that the highest irreversibilities occurred during the take-off phase due to high fuel consumption. The taxi phase has the highest cost of exergy destruction because of its relatively long duration with respect to others.

Kwak et al. [10] developed a computer program to assess the power output unit costs of a combined-cycle power plant (CCPP) based on exergy-costing. Kwon and Kwak [11] investigated a gas turbine based cogeneration system using two different exergoeconomic methods. This is the well known CGAM problem, initially introduced by Bejan et al. [12]. The results showed that the unit product cost is highly affected by component annualized costs. Turan and Aydin [13] carried out an exergoeconomic analysis of an aero-derivative gas turbine engine (LM6000). The levelized cost method was applied to calculate the exergy cost rate and unit cost rate for all streams. Their study found that the exergoeconomic factor for all components was low which means the sources of cost are associated with exergy destruction and exergy losses. Hence further modifications were necessary to improve the entire system's cost effectiveness. Fagbenle et al. [14] applied exergy, exergoeconomic, and environmental impact analysis to several gas turbine power plants in Nigeria. The results show that as the turbine inlet temperature (TIT) increased the exergetic efficiency improved while the cost of exergy destruction decreased as did the environmental impact. However, the applicability of exergoeconomic analysis extends beyond power plants to other energy applications such as drying plant, heat pump, refrigeration systems and sugar factories as reported by [15,16,17,18] respectively.

Environmental impact has a significant affect on sustainability, and the relation between them is inversely proportional. Many reports in the literature have analyzed the thermal process using a exergoenvironmental approach to minimize environmental impact by improving energy conversion efficiency [19]. Ahmadi et al. [20] performed exergy, exergoeconomic, environmental impact analysis and optimization to a CCPP with supplementary firing. Three objective functions were used for optimization purposes; exergy efficiency, total cost rate of the plant and carbon dioxide (CO_2) emissions. The results demonstrate that the cost of exergy destruction decreased as the TIT increased, with the combustion chamber the major source of irreversibilities. The CO_2 emission can be minimized by improving components efficiency and lowering fuel flow rate. Petrakopoulou et al. [21] conducted exergoeconomic and exergoenvironmental analyses, including CO_2 capture, for three oxy-fuel plants. A conflict was found between the economic and environmental assessments, making the optimal selection of plant subject to the specific concerns of the decision-maker.

The aim of this study is to perform exergy, exergoeconomic and sustainability analyses of an industrial reheat gas turbine engine integrated with two sequential combustions. The proposed system has not previously been considered in the literature using exergetic principles and a real set of data. This article reports the investigation of two systems; a reheat gas turbine engine and hypothetical model without reheat modification. The study aims to explore the effects of reheating on exergy destruction for all components and investigate how load variation and ambient conditions affect the proposed systems' performance. Further, the exergoeconomic and exergoenvironmental analyses are probed for an advanced energy system.

2. System Description

An advanced industrial RHGT and hypothetical simple cycle SCGT engines were investigated in the current study. In the RHGT system, a second combustion chamber was available for reheat whereas the SCGT has a single combustor. Reheating has several advantage for gas turbine engines such as augmenting power output, providing greater operational flexibility and improving the performance at full and part load. Furthermore, the temperature of the flow exhaust is maintained at a level compatible with combined cycle or any multi-generation system. However, the proposed systems consist of two shafts, see Figure 1. The first shaft is connected to the high-pressure components; and the second to the power turbine. The cold section comprises from a single axial compressor has a high pressure ratio equal to 35. The compressed air is delivered directly to the main annular combustor at high pressure and temperature and mixes with the fuel to yield hot gases. Both cases are consistent in the cold section; the difference between the RHGT and SCGT is in the hot section, the addition of the second, reheat combustor (RH) after high-pressure turbine (HPT) to reheat the gases before entering to the power turbine (PT). The useful work produce by the PT will enhance engine output power as a result of reheating. In the SCGT case, the exhaust gases flow from the HPT directly to the PT at a lower temperature. Hence, less useful power is produced by the PT and the temperature of the exhaust gases at the discharge is lower. The performance data for the proposed systems at International Standards Organization [ISO] ambient conditions are listed in Table 1.

Table 1. Performance data of proposed systems at ISO condition

Description	System		Unit
	SCGT	RHGT	
GT Power output	286.3	326.0	MW
Thermal efficiency	44.9	41.4	%
Heat rate	8,014.0	8,705.5	kJ/kWh
Pressure ratio	35.0	35.0	----
Exhaust Mass flow	677.6	677.6	Kg/s
Exhaust Temperature	469.8 (742.9)	602.8 (883.6)	oC (K)
Rotating speed	3,000	3,000	rpm
Frequency	50.0	50.0	Hz

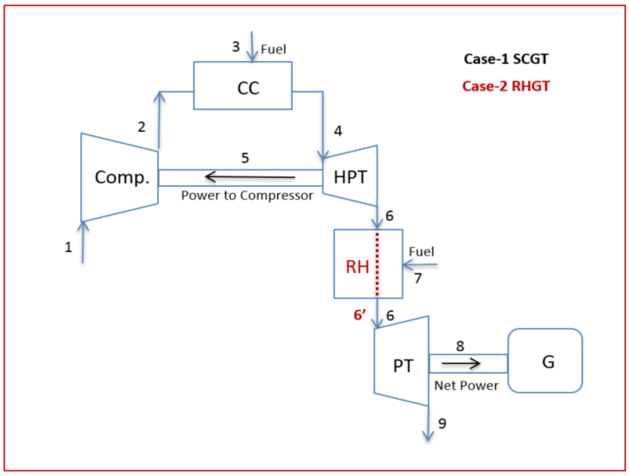


Figure 1. Schematic diagram of the SCGT and RHGT systems

3. Methodology

Exergoeconomic and sustainability analyses were performed for the two gas turbine systems at design and off-design conditions by changing ambient temperatures and load demands. The reference state was selected as ambient pressure and a temperature of 288 K (ISO conditions). The following assumptions were made for the models for the sake of simplicity.

- Steady state operating conditions.
- The air and combustion products behave as ideal gases.
- Potential and kinetic exergies may be omitted for all streams.
- The system fuel is natural gas, and nitrogen gas was inert.
- The combustion reaction is completed with 2% heat loss.
- The rotating parts are assumed to be adiabatic.

3.1. Energy Analysis

The first law of thermodynamics describes the exchange of energy of any system. The sum of all energies in the specific domain is constant based on the principle of the conservation of energy, neglecting internal losses, and assuming all type of energies are fully convertible. The energy equation for a steady-state condition can be expressed as:

$$\dot{Q} - \dot{W} = \Delta H + \Delta KE + \Delta PE \tag{1}$$

The left-hand side of Equation (1) represent the heat transfer rate and work whereas the right-hand side represents the differences in enthalpy, and kinetic, and potential energies, respectively. Energy analysis is a useful tool to estimate thermal efficiency and power output but it cannot quantify the inefficiencies of any energy system. The energy rate balance for gas turbine components are described in the following sections.

3.1.1. Axial Compressor

For a wide range of loads, the axial compressor delivers the air from an initial state under different climatic conditions, to the combustor chamber. The number of stages in the compressor is 22, and the overall pressure ratio is 35. The compressed air temperature is a function of pressure ratio (P_r), isentropic efficiency of the compressor (η_{AC}) and specific heat ratio of the air (γ_a) as given by the following expression:

$$T_e = T_i\left[1 + \frac{1}{\eta_{AC}}\left(P_r^{\frac{\gamma a-1}{\gamma a}} - 1\right)\right] \tag{2}$$

Where T_e is the exit temperature, and T_i is the inlet temperature of air stream.

The compressor consumption work can be calculated as follows:

$$\dot{W}_{AC} = \dot{m}_a C_{pa}(T_e - T_i) = \dot{m}_a(h_e - h_i) \tag{3}$$

Where $\dot{m}_a$ refer to air mass flow rate, h_e is the exit. enthalpy, h_i is the inlet enthalpy, and C_{pa} represents the heat capacity of the air stream which is significantly affected by temperature and can be expressed as [22]:

$$C_{pa}(T) = 1.04841 - 0.000383719T + \left(\frac{945378T^2}{10^7}\right) - \left(\frac{5.49031T^3}{10^{10}}\right) + \left(\frac{7.92981T^4}{10^{14}}\right) \tag{4}$$

3.1.2. Combustion Chambers

The fuel is provided to the combustion chamber at high pressure and different mass flow rates ($\dot{m}_f$), depending on compressed air temperature and load variations. The air stream mass flow rate varies, as well, in the cold section. The combustion reaction can be expressed in terms of mole fractions of the gas species as follows:

$$\bar{\lambda}\begin{bmatrix} 0.9334\ CH_4 \\ 0.00211C_2H_6 \\ 0.00029C_3H_8 \\ 0.0642\ N_2 \end{bmatrix} + \begin{bmatrix} 0.7748\ N_2 \\ 0.2059\ O_2 \\ 0.019\ H_2O \\ 0.0003CO_2 \end{bmatrix} \rightarrow (1+\bar{\lambda})\begin{bmatrix} y_{N_2}N_2 \\ y_{O_2}O_2 \\ y_{H_2O}H_2O \\ y_{CO_2}CO_2 \end{bmatrix} \tag{5}$$

Where $\bar{\lambda}$ and y are the fuel-to-air ratio and mole fraction respectively.

The molar analysis was performed to determine the fuel-to-air ratio and number of moles of products and reactants. The procedure is discussed in detail in [12], [23]. Once $\bar{\lambda}$ has been calculated, the composition of the combustion products can also be calculated as can the energy equation across the combustion chamber:

$$\dot{m}_a h_i + \eta_{CC}\dot{m}_f LHV = \dot{m}_g h_e \tag{6}$$

here the subscripts a , f and g represent the air, fuel and combustion product respectively. η_{CC} refer to combustion efficiency while LHV is the lower heating value of the fuel.

3.1.3. Gas Turbines

The RHGT unit comprises a high-pressure turbine (HPT) and a low-pressure turbine (PT). The second combustor lies between the two and is responsible for reheating, see Figure 1. The PT consists of four stages with highly efficient blade cooling and low leakage. The HPT is situated between two combustors and has only a single stage. When the initial expansion takes place, all the exhaust gases move to the second combustor [24]. The gas turbine exit stream temperature for exhaust gases is a function of the isentropic efficiency of the compressor (η_{GT}), expansion ratio (P_g) and specific heat ratio (γ_g), and can be expressed as follows:

$$T_e = T_i\left[1 + \eta_{GT}\left(1 - P_g^{\frac{1-\gamma g}{\gamma g}}\right)\right] \tag{7}$$

The energy rate balance for HPT and PT are given by:

$$\dot{W}_{GT} = \dot{m}_g C_{pg}(T_i - T_e) = \dot{m}_g(h_i - h_e) \tag{8}$$

Where C_{pg} refer to the heat capacity of the exhaust gases stream and be expressed as function of temperature as follows [18]:

$$C_{pg}(T) = 1.0397 + \left(\frac{2.429T}{10^5}\right) + \left(\frac{1.63T^1}{10^7}\right) - \left(\frac{6.966T^4}{10^{11}}\right) \tag{9}$$

3.2. Exergy Analysis

Exergy is the maximum possible work that can be extracted from the system under reversible operating condition. More specifically, the exergy represents the upper limit of useful work that can be produced by the system which is different from actual work. The exergy analysis can quantify the inefficiencies of the energy system based on the second law of thermodynamics. The exergy can transfer across the boundary of energy systems in a manner similar to energy but is not conserved in any real case due to entropy generation. The physical, kinetic, potential and chemical exergies constitute the major components of exergy and the total exergy value can be calculated by:

$$\dot{E}_x = \dot{E}_{ph} + \dot{E}_{ke} + \dot{E}_{pe} + \dot{E}_{ch} \quad (10)$$

The kinetic and potential exergies are omitted due to their insignificant effects in the proposed system. The physical exergy is defined as maximum useful work that can be obtained when the system moves from a specified state (T_s, P_s) to the reference state (T_o, P_o) due to differences in pressure and temperature. The physical exergy is comprised of thermal and mechanical exergy, and takes the form:

$$\dot{E}_{ph} = \dot{m}[(h_s - h_o) - T_o(s_s - s_o)] \quad (11)$$

Chemical exergy is associated with mass flows from the reference state to a dead state due to differences in concentration and molecular structure. The maximum useful energy that can be extracted while the flow moves to the chemical equilibrium state represents chemical exergy. The chemical exergy of the fuel and gas mixture streams can be calculated by:

$$\dot{E}_{ch} = \dot{m}_f LHV \quad (12)$$

$$\dot{E}_{ch} = \dot{n}[\sum y_k \bar{e}_k^{ch} + \bar{R}T_o \sum y_k \ln y_k] \quad (13)$$

Where $\dot{n}$ is the mole rate and $\bar{e}_k^{ch}$ is the molar chemical exergy for component k in the mixture, and can be found from standard chemical exergies table, as presented in [12].

The general exergy balance of a system can be written as:

$$\dot{E}_{xq} + \sum_i \dot{E}_{xi} = \sum_e \dot{E}_{xe} + \dot{E}_{xw} + \dot{E}_{xd} \quad (14)$$

Where the subscripts i and e denote the inlet and outlet condition of any component (k). The terms $\dot{E}_{xi}$, $\dot{E}_{xe}$ $\dot{E}_{xq}$ and $\dot{E}_{xw}$ refer to inlet and outlet exergy rates, heat transfer exergy rate and exergy due to work, respectively.

The $\dot{E}_{xd}$ term represent the exergy destruction during a process in a component or the system, and is a measure of the irreversibilities present. The chemical reaction, mixing, friction, heat transfer driven by finite temperature difference, and non-quasi-equilibrium compression or expansion constitute the main sources of irreversibilities. However, the rate of outlet exergy is less than the rate at the inlet for any component due to exergy destruction and exergy loss. In steady state conditions, these quantities can be related as:

$$\dot{E}_{xi} = \dot{E}_{xe} + \dot{E}_{xd} + \dot{E}_{xl} \quad (15)$$

The exergy destruction ($\dot{E}_{xd}$) and exergy loss ($\dot{E}_{xl}$) represent the waste exergy, and the difference between them is a measure of the losses inside the system. while the latter measures amount of energy emitted to the environment at the end of the process [25]. The system inefficiencies and exergetic efficiency for all components can be measured from exergy destruction and exergy loss, as illustrated in Table 2.

The exergetic efficiency is the ratio of extracted product exergy to the fuel exergy supplied to the component or entire system. Exergetic efficiency shows how much energy can be obtained from the maximum available work. Defining product exergy ($\dot{E}_{xp}$) and fuel exergy ($\dot{E}_{xf}$) is an essential step to evaluating exergetic efficiency.

Table 2. Exergy destruction rates and exergetic efficiencies for the different component at steady state conditions

Component	Exergy Destruction ($\dot{E}_{xd}$)	Exergetic efficiency (η_{ex})
Axial Compressor	$\dot{E}_{xd,AC} = \dot{E}_{xi} - \dot{E}_{xe} + \dot{W}_{AC}$	$\eta_{ex} = \frac{\dot{E}_{xi} - \dot{E}_{xe}}{\dot{W}_{AC}}$
Combustion Chambers	$\dot{E}_{xd,CC} = \dot{E}_{xi} - \dot{E}_{xe} + \dot{E}_{xf}$	$\eta_{ex} = \frac{\dot{E}_{xe}}{\dot{E}_{xi} + \dot{E}_{xf}}$
Gas Turbines	$\dot{E}_{xd,GT} = \dot{E}_{xi} - \dot{E}_{xe} - \dot{W}_{GT}$	$\eta_{ex} = \frac{\dot{W}_{GT}}{\dot{E}_{xi} - \dot{E}_{xe}}$
The Cycle	$\dot{E}_{xd} = \sum_k \dot{E}_{xd,k}$	$\eta_{ex} = \frac{\dot{E}_{xp}}{\dot{E}_{xf}}$ =1- $\frac{\dot{E}_{xd}+\dot{E}_{xl}}{\dot{E}_{xf}}$

3.3. Exergoeconomic Analysis

The exergoeconomic analysis is a useful tool aimed at minimizing both the cost of inefficiencies and production cost. The exergetic and economic analyses together provide decisive information about the proposed system that cannot be obtained by separate investigations. The SPECO method is required to appraise the cost of streams and exergoeconomic parameters, and is used to determine cost effectiveness enhancement potentials [3,12]. Appendix-A contains further explanation about SPECO method.

3.3.1. Economic Analysis

The fuel cost, annual levelized equipment purchase cost (EPC) with operation and maintenance (O&M) cost for each component are the most important outcomes of the economic analysis. Table 3 shows some of the economic constraints and assumptions used in the proposed economic model. The general cost balance equation for each component is:

$$\dot{C}_{q,k} + \sum_i \dot{C}_{i,k} + \dot{Z}_k = \sum_e \dot{C}_{e,k} + \dot{C}_{w,k} \quad (16)$$

Where $\dot{C}$ is the flow cost rate in \$/hr and $\dot{Z}_k$ refer to the sum of capital investment costs (EPC) and operation and maintenance (O&M) expenses, and takes the form:

$$\dot{Z}_k = \dot{Z}_k^{CI} + \dot{Z}_k^{OM} \quad (17)$$

The input values must be levelized before the model is used in the exergoeconomic analysis.

Table 3. Economic constants and assumptions for the proposed system

Parameter	Unit	Value
Annual operation hours (τa)	h/year	8,000
Engine life time (n)	year	20
Nominal escalation rate (r_n)	%	5.0
Discount rate (i_{eff})	%	6.0
Fuel price (FP)	$/GJ	5.0
Lower heating value (LHV)	kJ/kg	46,802

Cost levelization is used to relate the expenditure at the beginning of the project with an equivalent annuity to avoid a non-uniform cash flow. The present worth (PW), also called net present value (NPV), represents an equivalent value for a series of costs or benefits over time with respect to present day values. The salvage value (S_v) is the estimated value of the asset at the end of its useful life [12,13].

$$PW = CIC - S_v . PWF \quad (18)$$

$$S_v = j.CIC \quad (19)$$

Where *CIC*, j and *PWF* are capital investment cost, salvage rate (%) and present worth factor respectively. *PWF* is used in a number of calculations estimating values to be received in the future, and is written as:

$$PWF = \frac{1}{(1+i_{eff})^n} \quad (20)$$

Where i_{eff} denote the discount rate and n the relevant number of years.

The capital recovery factor (CRF) converts a present value to a series of equal annual payments over a specified time, at a specified discount rate, so as to recover an initial investment.

$$CRF = \frac{i_{eff}\left(1+i_{eff}{}^{n}\right)}{\left(1+i_{eff}\right)^n+1} \quad (21)$$

The annual capital cost (ACIC) is given by:

$$ACIC = CRF.PWF(i,n) \quad (22)$$

The hourly levelized cost of plant and its k^th component are given by:

$$\dot{Z}^T = \frac{ACIC}{\tau} \quad (23)$$

$$\dot{Z}_k = \dot{Z}^T \frac{EPC_k}{\sum EPC_k} \quad (24)$$

Where τ represent the plant availability in terms of operational hours per year.

The energy systems must consider a number of entering and exiting streams for each component, work done and heat generated, and how both of these interact with the environment at steady state operating conditions. In exergy costing, all streams have a cost associated with exergy values and can be expressed in general form as:

$$\dot{C}_j = \dot{c}_j \dot{E}_{x,j} \quad (24)$$

Where j could denote an entering stream, exiting stream, heat transfer and work done. The term $\dot{C}_j$ represents the livelized cost per unit exergy for the j^{th} stream. Combining the cost rate expression of equation (24) with the general cost equation (16) leads to the following:

$$\dot{c}_{q,k}\dot{E}_{q,k} + \sum_i (\dot{c}_i\dot{E}_i)_k + \dot{Z}_k = \sum_e (\dot{c}_e\dot{E}_e)_k + \dot{c}_{w,k}\dot{W}_k \quad (25)$$

For exergoeconomic analysis to evaluate the exiting streams it is assumed that the cost per unit exergy for the entering streams are known for all components. The EPC and O&M costs are extracted from economic analysis. Sometimes there are a number of inlet and outlet streams for the components and that can lead to the situation where the number of unknowns are greater than the number of cost balance equations. In such circumstances auxiliary equations are developed using fuel and product rules [2,12,25].

3.3.2. Exergoeconomic Balance Equations

The exergy destruction and stream costs can be estimated by solving the cost balance equations with auxiliaries. This will give a linear system of equations, as:

$$[\dot{E}_k] X [c_k] = [\dot{Z}_k] \quad (26)$$

Where $[\dot{E}_k]$ is matrix of an exergy-rate that obtained from exergy analysis, $[\dot{Z}_k]$ is the vector of total cost, which is obtained from economic analysis, and $[c_k]$ is a vector representing the exergetic cost.

The essential cost balance and auxiliary equations for all components in the RHGT are presented below:

- Axial Compressor

$$\dot{C}_1 + \dot{C}_5 + \dot{Z}_{AC} = \dot{C}_2 \quad (27)$$

$$\dot{C}_1 = 0 \; [\text{Assumption at reference state}] \quad (28)$$

- Combustion Chamber [CC]

$$\dot{C}_2 + \dot{C}_3 + \dot{Z}_{CC} = \dot{C}_4 \quad (29)$$

$$\dot{C}_3 = Fuel\ price = Constant \quad (30)$$

- High-Pressure Turbine [HPT]

$$\dot{C}_4 + \dot{Z}_{HPT} = \dot{C}_5 + \dot{C}_6 \quad (31)$$

$$\frac{\dot{C}_4}{\dot{E}_4} = \frac{\dot{C}_6}{\dot{E}_6} [\text{F rule}] \quad (32)$$

- Reheater [RH]

$$\dot{C}_6 + \dot{C}_7 + \dot{Z}_{RH} = \dot{C}_{6\prime} \quad (29)$$

$$\dot{C}_7 = Fuel\ price = Constant \quad (30)$$

- Power Turbine [PT]

$$\dot{C}_{6\prime} + \dot{Z}_{PT} = \dot{C}_8 + \dot{C}_9 \quad (31)$$

$$\frac{\dot{C}_{6'}}{\dot{E}_{6'}} = \frac{\dot{C}_9}{\dot{E}_9} [\text{F rule}] \quad (32)$$

Fuel cost in $/h can be calculated using the following expression.

$$Fuel\ price = 3600.\tau.FP.LHV.\dot{m}_f \quad (33)$$

The variable of cost per exergy unit ($\dot{C}_1 - \dot{C}_9$) is solved by using equations (27) to (32).

3.3.3. Exergoeconomic Factor

The exergoeconomic factor, f_k, is an important exergoeconomic variable used to evaluate and optimize thermal systems. The cost of the k^{th} component can be attributed to two sources, the first is associated with exergy related costs such as exergy losses which may be considered as inefficiencies, and the second related to non-exergy quantities such as capital investment and O&M costs. Identifying the sources of costs is considered useful information that helps to determine the relative weight of the two sources which can improve system cost effectiveness. The exergoeconomic factor f_k can be formulated as:

$$f_k = \frac{\dot{Z}_k}{\dot{Z}_k + c_{f,k}\left[\dot{E}_{d,k+}\dot{E}_{L\ k}\right]} \quad (34)$$

3.4. Exergoenvironmental Analysis

An exergoenvironmental analysis evaluates environmental impact from an exergy perspective [26]. Dincer and Rosen [27] reported three relationships between exergy and environmental damage as illustrated in Figure 2. This damages strongly affects sustainability, and takes the form of waste exergy emission, the order of destruction, and resource degradation.

Figure 2. Exergy analysis of three forms of environmental damage

As the exergetic efficiency increase there is a substantial improvement in several measures, including CO_2, and the environmental impact of the energy systems is reduced. In the present study, CO_2, carbon monoxide (CO) and nitrogen oxide (NOx) were selected to represent exhaust gases, and found to be highly effected by engine efficiency. The amount of emissions per unit of product (kg/kWh) is evaluated using HEPHAESTUS, generic combustor model, developed by Cranfield University. The cost of emissions can be calculated by multiplying their flow rates by unit cost, which is equal to 0.024 $/kg for CO_2, 0.02086 $/kg for CO and 6.853 $/kg for NOx. The total cost of environmental damage can be added directly to the fuel cost, to integrate environmental and economical aspect.

4. Results and Discussion

The inputs data of the exergy analysis study was derived from IPSEpro software while the economic input data from the end user and previous literature. Table 4 illustrates thermodynamic and exergetic data for RHGT at ISO ambient condition.

Table 4: Exergetic data for RHGT at ISO ambient condition

State	Substance	Mass flow [kg/s]	Temp. [K]	Pressure [bar]	Exergy $\dot{E}_{ph}$ [MW]	Exergy $\dot{E}_{ch}$ [MW]	Exergy $\dot{E}_x$ [MW]
1	Air	677.7	288.0	1.010	0.00	0.91	0.91
2	Air	677.7	830.5	35.45	371.2	0.91	372.1
3	Fuel	13.62	320.0	41.35	6.890	637.5	644.3
4	Exhaust gases	691.3	1550.0	33.68	847.5	6.84	854.4
5	Power to Compressor				391.1	----	391.1
6	Exhaust gases	691.3	1096.6	5.97	438.6	6.84	445.5
6’	Exhaust gases	3.220	288.0	6.96	0.880	150.8	151.7
7	Fuel	694.5	1269.2	5.67	547.4	9.74	557.1
8	Net Power				326.0	----	326.0
9	Exhaust gases	694.5	877.7	1.010	207.1	9.74	216.9

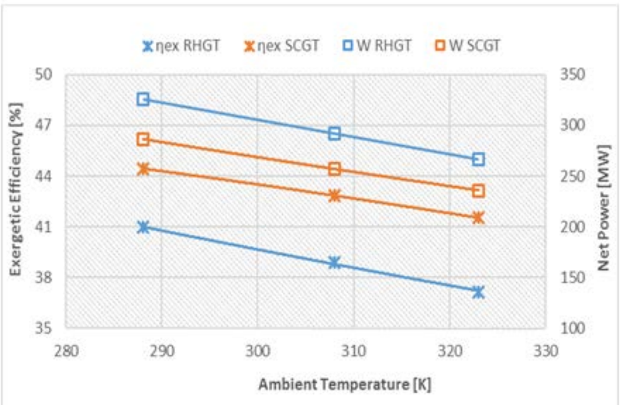


Figure 3 Relationship between ambient temperature, exergetic efficiency and net power output

Change in ambient temperature reflects climatic conditions around the gas turbine location. Figure 3 presents the relationship between ambient temperature, exergetic efficiency and net power output. Increase in ambient temperature leads to a decrease in both exergetic efficiency and net power output because of the increased power consumption of the compressor as a consequence of decreased air density. The compressed air inlet temperature of the combustion chamber increases as the ambient temperature increases and that causes a reduction in fuel consumption. This further clarifies the reason for the clearly discernible drop in exergetic efficiency and net power output with increase in ambient temperature. The fuel consumption in the combustion chamber is reduced but the compressor consumes more energy, reducing exergetic efficiency and net power output. Furthermore, the quality of electrical energy is higher than chemical energy because of the level of convertibility to useful work.

The exergetic efficiency and net power output of RHGT and SCGT were reduced by 1.7 MW, 1.4 MW and 0.1%, 0.08% per degree respectively. The RHGT is more sensitive to ambient temperature variation due to coupling between the high-pressure rotational parts which control the reheater inlet condition. The exergetic efficiency of the SCGT is higher than that of the RHGT system while the net power output for the latter is substantially higher due to the reheat effect. However, the effect of changes in ambient temperature can be largely eliminated by cooling the gas turbine intake area using a commercially available cooling system.

Figure 4 presents the exergetic efficiency of RHGT and SCGT as a function of load. It is well known that the electrical grid is subject to load variation due to changes in end user demand. Thus load variations are an essential aspect of grid operation but greatly effect engine life due to the thermal stress generated on, e.g. startup. However, compared to other fossil fuel energy systems, the gas turbine engine has the advantages of quick startup and operational flexibility. The SCGT shows higher exergetic efficiency at all load conditions investigated due to combustor effects. These are a major cause of irreversibilities, and the RHGT has two sequential combustors.

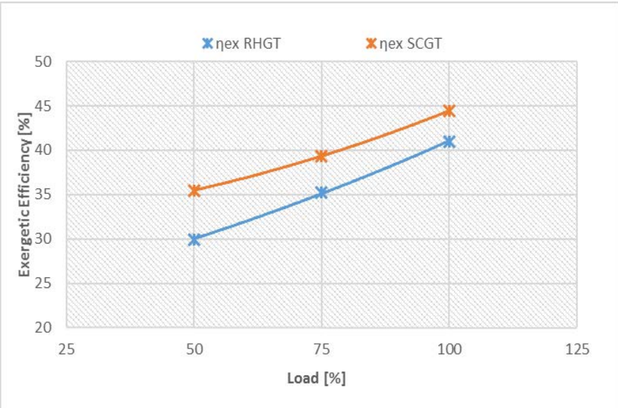


Figure 4. Exergetic efficiency versus load variations

The high-pressure TIT as well as fuel consumption is reduced as the load reduces. The relative reduction in power consumption in the compressor is less than the fuel consumption in the system and that may be a consideration in the drop in efficiency at part load.

Figure 5 illustrates exergy destruction in the RHGT engine for each component as a percentage of total exergy destruction. The combustion chamber and reheat system, combined, represent 79.6 % of the total exergy destruction. The air compressor has a relatively higher percentage (7.8%) than other rotating parts, which may be attributed to aerodynamic and heat transfer loss. The irreversibilities in the high-pressure turbine (7.0%) are considered to be due to high fuel exergy. The power turbine (5.6%) contributes least irreversibilities because of increasing TIT as a result of reheat.

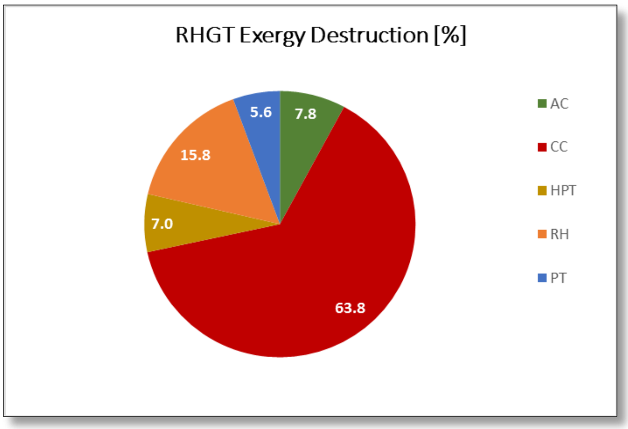


Figure 5. Exergy destruction for RHGT system

The sum of exergy destruction cost and component cost ($\dot{C}_d + \dot{Z}_k^T$) for all the parts in proposed systems are shown in Figure 6. The combination between expenses and the cost of inefficiencies provides useful information on relative priorities for replacement or refurbishment of components. Exergoeconomic principles recommend paying more attention to the component that has the highest value of ($\dot{C}_d + \dot{Z}_k^T$) to maximise improvements to the cost effectiveness of the entire system and for both systems being considered it is obviously the combustion chamber that has the highest exergy destruction. For both the SCGT and RHGT the combustion chamber is followed by the air compressor (14.2% and 12.6%, respectively) and high pressure turbine (13.6% and 12.4%), then the reheat for the RHGT (10.9%) and bottom of the list for both is the power turbine (11.0%, 9.8%) but, as can be seen, the differences between the components was very small, a range of 3.2% at most.

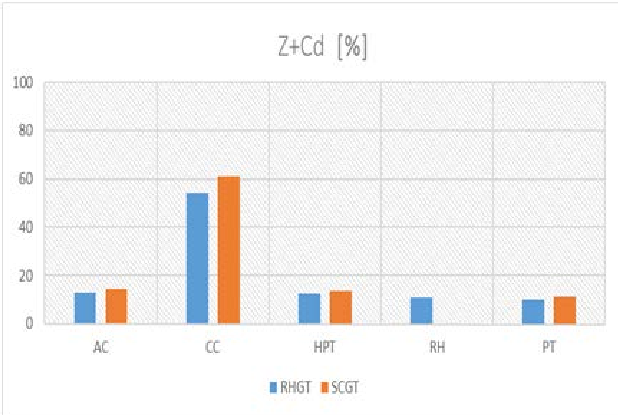


Figure 6. The relative percentages of exergy destruction and component cost ($\dot{C}_d + \dot{Z}_k^T$) for RHGT and SCGT

The impact of improvement in each component is constrained by location and source of cost. The former is obvious while the latter need tools to determine the source of cost, to determine what is related to exergy destruction and what to capital investment and O&M. The exergoeconomic factor is the most useful variable for shedding light on the cost source, see Figure 7. Values of exergoeconomic factor greater than 50% means that the component capital investment and O&M cost should be reduced even at the expense of a loss in efficiency. On the other hand, low values of the exergoeconomic factor such as we have for both SCGT and RHGT air compressor, high pressure turbine and the power turbine signifies that, for example, capital would be well spent to improve the performance of these components. Where there are contradictions between evaluated results for the components, the priority should always be given to the higher $\dot{C}_d + \dot{Z}_k^T$ value.

It is expected that the cost effectiveness of the two systems will improve after making the following changes:

- Improve combustion chamber cost effectiveness by increasing TIT and compressed air temperature.
- It is highly recommended to operate the compressor at a low ambient temperature, below the ISO ambient condition.
- Adapt the reheat system to reduce the expansion ratio of the HPT and increase the inlet temperature of reheater and reduce fuel flow rate.

- Evaluate the operation of all rotating parts with a view to increase the pressure ratio (and enhance the expansion ratio) and so increase isentropic efficiencies.

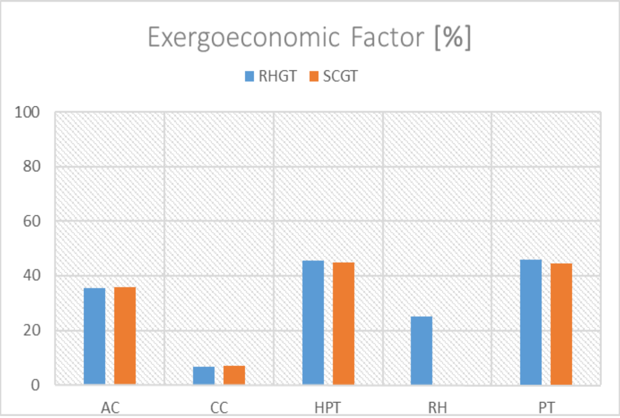


Figure 7. Exergoeconomic factors for engine components for both RHGT and SCGT

The product cost of any energy system is considered an important factor in its evaluation. The RHGT produces electricity at a rate of 7.58 $/GJ whereas the SCGT has a rate of 7.77 $/GJ. The better figure for RHGT is attributed to an improvement in performance as a result of reheating. The RHGT production cost is substantially lower than aero-derivative gas turbine engine as found in [13].

Figure 8 and Figure 9 shows the environmental impact of RHGT and SCGT at different ambient temperatures, and show that the emissions levels increased monotonically as the ambient temperature increased. The RHGT demonstrates a higher level of environmental impact because of its greater fuel consumption. CO_2 emission is strongly related to engine efficiency. Generally, for a given rate of fuel consumption, the amount of carbon dioxide is greater the higher the engine efficiency, and the SCGT achieved the lower value of CO_2 emissions per MWh. CO emissions result mainly from incomplete combustion and poor mixing of air and fuel that result in a low combustion efficiency. The amount of CO is reduced substantially as the air-to-fuel ratio increases, but this will increase NOx emissions. A long residence time at high temperature is directly proportional to NOx formation, conversely the NOx emission of the combustor substantially reduce as the residence time reduces. Thus there is conflict in the conditions required minimise the individual polluting emissions.

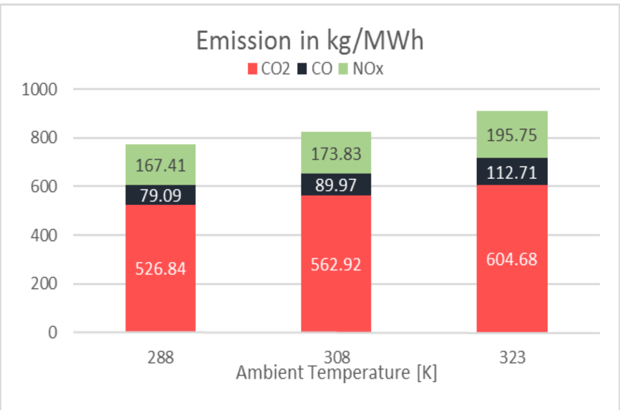


Figure 8. The CO_2 and pollutant emissions for RHGT per MWh of electrical energy generated for three ambient temperatures

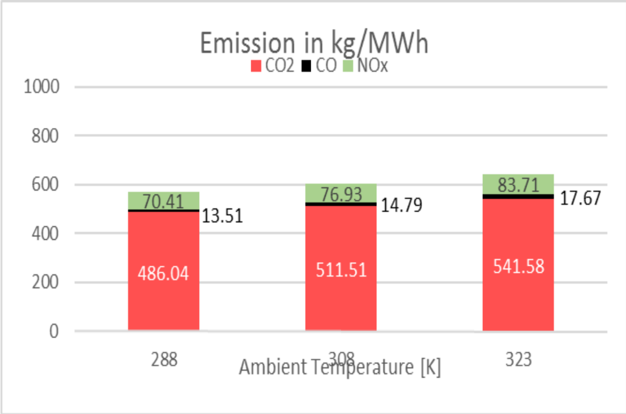


Figure 9. The CO2 and pollutant emissions for SCGT per MWh of electrical energy generated for three ambient temperatures

It can be seen from the figures that the SCGT produces lower emissions for each of the three pollutants considered at each ambient temperature and is thus considered the better choice from an environmental perspective.

In the light of the previous discussion, there is an important issue associated with energy system performance that needs clarifying. The efficiency is sometimes used as equivalent to the performance but that not always true. The performance of an energy system must be related mainly to its design objectives, which may be in contradiction with maximum efficiency. For instance, the RHGT produced more power and, today, is considered the best primary mover for multi-generation systems. The main reason for the environmental superiority of the SCGT is the hypothetical high-pressure ratio which does not exist at any industrial gas turbine on the market except the RHGT.

5. Conclusion

In this paper, exergy, exergoeconomic and sustainability analyses have been conducted for two advanced industrial gas turbine engines at different operating conditions. Variation of load and ambient temperature was investigated to simulate their impacts on electrical grid stability and engine performance. The thermodynamic and exergetic data was extracted from the gas turbine models and verified with the manufacturer's published data. The main conclusions drawn from this study are summarized below:

- The main source of irreversibilities occurs in the combustion chambers due to chemical reaction, mixing, temperature difference and friction.
- An increase in ambient temperature leads to an adverse effect on the exergetic efficiency and net power output due to an increase in power consumed by the compressor. The RHGT and SCGT exergetic efficiency and net power output were reduced by 1.7 MW, 1.4 MW and 0.1%, 0.08% respectively per degree increase in ambient temperature.
- The SCGT shows higher exergetic efficiencies at different load conditions due to fewer components that are compatible with thermal design criteria.
- The RGHT shows a lower cost of electricity production due to an improvement in power turbine performance as a result of reheating.

- The RHGT demonstrates a higher level of environmental impact compared to the SCGT because of sequential combustion which consumes more fuel.

Nomenclature

$\dot{C}$	Cost rate
c	Average unit cost
C_p	Heat capacity
$\dot{E}$	Exergy rate
$\bar{e}_k^{ch}$	Molar chemical exergy
f_k	Exergoeconomic factor
h	Enthalpy
$\bar{h}$	Molar enthalpy
i_{eff}	Discount rate
j	Salvage rate
$\overline{LHV}$	Low heating value in molar basis
M	Molecular weight
$\dot{m}$	Mass flow rate
n	Number of mole
P	Pressure
P_r	Pressure ratio
P_g	Expansion ratio
$\dot{Q}$	Heat transfer rate
R	Gas constant
$\bar{R}$	Universal gas constant
r_k	Relative cost difference
r_n	Nominal escalation rate
S	Entropy
S_v	Salvage value
$\bar{s}$	Molar entropy
$\dot{S}_{gen}$	Entropy generation
T	Temperature
$\dot{W}$	Work rate
y	Mole fraction
$\dot{Z}$	Purchase cost rate
Greek symbols	
γ	Specific heat ratio
$\bar{\lambda}$	Fuel-to-air ratio
η_{ex}	Exergetic efficiency
μ_k	chemical potential
Subscripts	
a	Air
CI	Capital investment
ch	Chemical
d	destruction
e	Exit
F	Fuel
g	Combustion product
i	Inlet
k	Component
ke	Kinetic energy
L	Loss
o	Reference state
p	Product
ph	Physical
pe	Potentials
S	Specified state
t	Turbine
T	Total
x	Total
Abbreviations	
ACIC	Annual capital cost
CC	Combustion chamber
CIC	Capital cost
CRF	Capital recovery factor
FP	Fuel price
GT	Gas turbine
AC	Axial compressor
HPT	High pressure turbine
IPT	Intermediate pressure turbine
LHV	Lower heating value
ISO	International Standards Organization
O&M	Operation and maintenance
PEC	Purchased equipment cost
PRO	Pressure retarded osmosis
PT	Power turbine
PW	Present worth
PWFQ	Present worth factor
RH	Reheater
RHGT	Reheat gas turbine
SCGT	Simple cycle gas turbine
SPECO	Specific exergy costing method
TIT	Turbine inlet temperature

References

[1] P. Ahmadi, I. Dincer, and M. Rosen, "Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants," *Energy*, vol. 36, no. 10, pp. 5886-5898, 2011.

[2] G. Tsatsaronis, "Combination of Exergetic and Economic Analysis in Energy-Conversion Processes," 1985, pp. 151-157.

[3] G. Tsatsaronis and M. Winhold, "Exergoeconomic analysis and evaluation of energy-conversion plants—I. A new general methodology," *Energy*, vol. 10, no. 1, pp. 69-80, Jan. 1985.

[4] L. Meyer, R. Castillo, J. Buchgeister, and G. Tsatsaronis, "Application of exergoeconomic and exergoenvironmental analysis to an SOFC system with an allothermal biomass gasifier," *Int. J. Thermodyn.*, vol. 12, no. 4, pp. 177-186, 2009.

[5] L. G. Alves and S. A. Nebra, "Exergoeconomic study of hydrogen production from steam reforming of natural gas," in *ECOS 2005 - Proc. 18th Int. Conf. on Efficiency, Cost, Optimization, Simulation, and Environmental Impact of Energy Systems*, 2005, pp. 1123-1130.

[6] S. E. Yalçin and T. Derbentl, "Exergoeconomic analysis of boilers," in *ECOS 2006 - Proc. 19th Int. Conf. on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems*, 2006, pp. 459-463.

[7] L. Wang, Y. Yang, C. Dong, Z. Yang, G. Xu, and L. Wu, "Exergoeconomic Evaluation of a Modern Ultra-Supercritical Power Plant," *Energies*, vol. 5, no. 12, pp. 3381-3397, 2012.

[8] M. Ameri, P. Ahmadi, and A. Hamidi, "Energy, exergy and exergoeconomic analysis of a steam power plant: A case study," *Int. J. Energy Res.*, vol. 33, no. 5, pp. 499-512, Apr. 2009.

[9] O. Altuntas, T. H. Karakoc, and A. Hepbasli, "Exergetic, exergoeconomic and sustainability assessments of piston-prop aircraft engines," *Isi Bilim. Ve Tek. Dergisi/ J. Therm. Sci. Technol.*, vol. 32, no. 2, pp. 133-143, 2012.

[10] H. Kwak, D. Kim, and J. Jeon, "Exergetic and thermoeconomic analyses of power plants," *Energy*, vol. 28, no. 4, pp. 343-360, 2003.

[11] Y.-H. Kwon, H.-Y. Kwak, and S.-D. Oh, "Exergoeconomic analysis of gas turbine cogeneration systems," *Exergy, An Int. J.*, vol. 1, no. 1, pp. 31-40, 2001.

[12] A. Bejan, G. Tsatseronis and M. Moran, *Thermal design and optimization*. John Wiley & Sons, 1996.

[13] O. Turan and H. Aydin, "Exergetic and exergo-economic analyses of an aero-derivative gas turbine engine," *Energy*, vol. 74, pp. 638-650, 2014.

[14] R. O. Fagbenle, S. S. Adefila, S. Oyedepo, and M. Odunfa, "Exergy, Exergoeconomic and Exergoenvironomic Analyses of Selected Gas Turbine Power Plants in Nigeria," pp. 1-13, 2014.

[15] T. Taner, "Optimisation processes of energy efficiency for a drying plant: A case of study for Turkey," Appl. Therm. Eng., vol. 80, pp. 247-260, 2015.
[16] H. Esen, M. Inalli, M. Esen, and K. Pihtili, "Energy and exergy analysis of a ground-coupled heat pump system with two horizontal ground heat exchangers," Build. Environ., vol. 42, no. 10, pp. 3606-3615, 2007.
[17] G. Chiummo, A. Di Nardo, G. Langella, and C. Noviello, "Exergoeconomic analysis of absorption systems for turbine air inlet cooling in trigeneration plants," in ECOS 2005 - Proceedings of the 18th International Conference on Efficiency, Cost, Optimization, Simulation, and Environmental Impact of Energy Systems, 2005, pp. 471-476.
[18] T. Taner and M. Sivrioglu, "Energy-exergy analysis and optimisation of a model sugar factory in Turkey," Energy, vol. 93, pp. 641-654, 2015.
[19] M. Rosen and I. Dincer, "On exergy and environmental impact," *Int. J. Energy Res.*, vol. 21, no. January 1996, pp. 643-654, 1997.
[20] P. Ahmadi, I. Dincer, and M. A. Rosen, "Exergy, exergoeconomic and environmental analyses and evolutionary algorithm based multi-objective optimization of combined cycle power plants," *Energy*, vol. 36, no. 10, pp. 5886-5898, 2011.
[21] F. Petrakopoulou, A. Boyano, M. Cabrera, and G. Tsatsaronis, "Exergoeconomic and exergoenvironmental analyses of a combined cycle power plant with chemical looping technology," *Int. J. Greenh. Gas Control*, vol. 5, no. 3, pp. 475-482, 2011.
[22] M. Moran, H. Schapiro, D. Boettner, M. Bailey, *Fundamentals of Engineering Thermodynamics*. John Wiley & Sons, 2004.
[23] A. Almutairi, P. Pilidis, and N. Al-Mutawa, "Energetic and Exergetic Analysis of Combined Cycle Power Plant: Part-1 Operation and Performance," *Energies*, vol. 8, no. 12, pp. 14118-14135, Dec. 2015.
[24] A. Almutairi, "Gas Turbine Technologies." UK, 2014.
[25] G. Tsatsaronis, "Thermoeconomic analysis and optimization of energy systems," *Prog. Energy Combust. Sci.*, vol. 19, no. 3, pp. 227-257, 1993.
[26] M. H. Khoshgoftar Manesh, P. Navid, M. Baghestani, S. Khamis Abadi, M. A. Rosen, A. M. Blanco, and M. Amidpour, "Exergoeconomic and exergoenvironmental evaluation of the coupling of a gas fired steam power plant with a total site utility system," *Energy Convers. Manag.*, vol. 77, pp. 469-483, 2014.
[27] I. Dincer and M. A. Rosen, *Exergy*. Elsevier Ltd, 2013.

Appendix A – Exergoeconomic Method

SPECO approach was selected among various exergoeconomic methods in the present study to evaluate the proposed model. The methodology of SPECO consists of three sequential steps, which are identify all exergy streams in the system, define component fuel and product, and producing the cost equations.

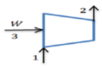
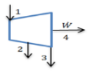
Step 1: Identification of Exergy Streams

Conduct an exergetic analysis on the proposed system will identify the value of all streams that crossing the boundaries of the components .In addition, work or heat interactions with surroundings must be identified.

Step 2: Definition of Fuel and Product

Selection the fuel and product of the component significantly affect on exergetic analysis results, which changes according to component type and purpose. The fuel value depends on the exergy difference between inlet and outlet streams for a specified component. The product value is equal to the adding exergy to a specified stream within a component. However, Table A-1 summarizes the cost-rate definition according to the fuel and product principles for different components of proposed system.

Table A 1. Fuel and Product Cost Rates for Different Components at steady-state operation

Component	Schematic	Cost of fuel rate	Cost rate of Product
Compressor		$\dot{C}_w$	$\dot{C}_1 - \dot{C}_2$
Turbine		$\dot{C}_1 - \dot{C}_2 - \dot{C}_3$	$\dot{C}_w$
Combustion Chamber		$\dot{C}_1 + \dot{C}_2$	$\dot{C}_3$

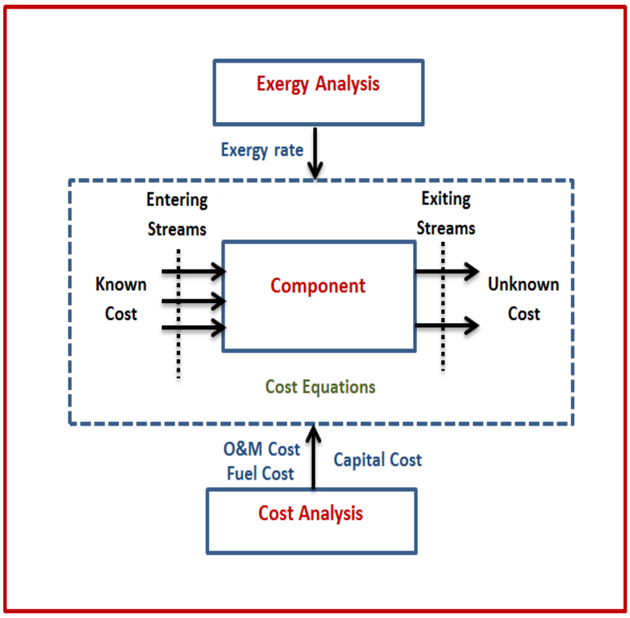


Figure A 1. Block Diagram of SPECO Method

Step 3: Cost Equations

The general cost balance equations are evaluated separately for all components in the proposed system. The entering and exiting streams costs , plus the capital cost and the operation and maintenance cost, are introduced in the balance equation (equation (16)). However, in the case of more than one stream exiting from a specified component such as turbine ,auxiliary equations is required, to aid in solving the cost-rate balance equations. Figure A-1 illustrates SPECO method.