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OPTIMIZATION OF A MICRO GAS TURBINE USING GENETIC ALGORITHM

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ABSTRACT

Distributed generation is an attractive way of producing energy, minimizing transport losses and enhancing energy efficiency. Micro gas turbines in distributed generation systems add other advantages such as low emissions and fuel flexibility. In the present work, a 100 kW micro gas turbine is considered. The optimization procedure is done by Genetic Algorithm method which is a new method in optimizing problems. The plant is comprised of an air compressor, recuperator, combustion chamber and gas turbine. The design Parameters of the plant, were chosen as: compressor pressure ratio, compressor isentropic efficiency, gas turbine isentropic efficiency, combustion chamber inlet temperature and the temperature of the combustion gas at the gas turbine inlet. In order to find the design parameters optimally, a thermo-economic approach has been followed. An objective function, representing the total cost of the plant in terms of dollar per second, was defined as the sum of the operating cost, related to the fuel consumption, the capital investment which stands for equipment purchase and maintenance cost. Subsequently, different parts of the objective function have been expressed in terms of design variables. Finally, the optimal values of design variables were obtained by minimizing the objective function using Genetic Algorithm code that is developed in Matlab software programming.

INTRODUCTION

Due to strict power quality, new approaches to power generation have evolved. One of these is the microturbine that is typically single-shaft engine with no gearbox. Micro Gas Turbines (MGTs) burn gaseous and liquid fuels to create high speed rotation that turns an electrical generator. They are expected to have steady growth in future energy service especially in distributed generation, for instance in hospitals, supermarkets, and in small industry. Distributed generation is one of the most promising alternatives for generating and delivering electric power [1-3].

The size range for microturbines available and in development is from 30 to 500 kilowatts (kW), while conventional gas turbine sizes range from 500 kW to 250 megawatts (MW). Microturbines run at high speeds and, like larger gas turbines, can be used in power-only generation or in combined heat and power (CHP) systems. They are able to operate on a variety of fuels, including natural gas, sour gases (high sulfur, low Btu content), and liquid fuels such as gasoline, kerosene, and diesel fuel/distillate heating oil. In resource recovery applications, they burn waste gases that would otherwise be flared or released directly into the atmosphere [1-8].

Single-shaft models generally operate at speeds over 60,000 revolutions per minute (rpm) and generate electrical power of high frequency, and of variable frequency (alternating current --AC). This power is rectified to direct current (DC) and then inverted to 60 hertz (Hz) for commercial use. In the two-shaft version, the power turbine connects via a gearbox to a generator that produces power at 60 Hz [3,6].

To accurately analyze and assess the energy saving potential and the economic feasibility of microturbine for distributed power generation and combined production use, optimize operation modes of cogeneration or trigeneration system, a simple but enough accurate performance model of microturbine is desired. Many researchers have engaged on developing mathematic model for gas turbine and many

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models were developed. A good review about that was given by Jurado [9]. But among these existing models, some of them are detailed first principle models based upon fundamental mass, momentum and energy balances, and thus are very complicated and time-consumed in computation. These models are not suitable for hourly energy consumption analysis of equipment operation though they can be used for design of gas turbine. To simulate the dynamic characteristics of microturbine and design control system for it, some non-linear models were developed by Jurado [9-10]. However, these models are mainly interested in electric-mechanical behavior and care few about energy conversion and utilization process. Ehyaei [11] studied the selection of micro gas turbines considered to meet the electrical, domestic hot water, heating and cooling energy needs of a residential building located in Tehran, Ahvaz and Hamedan, 3 cities in Iran and in reference [12] studied the economic analysis of a microturbine for CHP production.

Much useful research work has been done by Zaltash in developing models for building cooling, heating and power system [13-14]. Based on experimental data of a commercially available microturbine, a semi-empirical model was developed by Labinov [14]. In his model, the efficiencies of turbine, compressor and recuperator were regards as constants and thermophysical properties of air and flue gas were assumed to be not change, which is not the case in practice.

As is seen, most of the studies in the above cited literature have been conducted using the first law of thermodynamics or energy balance approach. However, energy analysis does not characterize the irreversibility of processes within the system. In contrast, exergy analysis will characterize the work potential of a system. Exergy analysis helps determine the real thermodynamic inefficiencies in a thermal system and their causes and locations, and improve the overall system and its components. Exergy analysis is based on the second law of thermodynamics. [15-17]

Huang [18] applied the second law method for the thermodynamic analysis of combustion gas turbine cogeneration system, and observed the effects of pinch point temperature and process steam pressure on the energetic and exergetic performance of the system. Energy and exergy analysis are performed for a combined-cycle power plant to analyze and identify the potential for improving efficiency of the system Cihan et al. [19]. They found that combustion chambers, gas turbines and heat recovery steam generators (HRSG) are the main sources of irreversibilities representing more than 85% of the overall exergy losses. Regulagadda et al. [20] carried out a thermodynamic analysis of a subcritical boiler–turbine generator is performed for a 32 MW coal-fired power plant. They conducted a parametric study for the plant under various operating conditions, including different operating pressures, temperatures and flow rates, in order to determine the parameters that maximize plant performance. Their results show that boiler and turbine irreversibilities yield the highest exergy losses in the power plant.

In the present paper, the thermodynamic modeling of a microturbine plant system is performed. The new objective function including total cost rate of product and cost rate of environmental impact, is considered. The design parameters are considered as compressor pressure ratio (r_{AC}), compressor isentropic efficiency (η_{AC}), gas turbine isentropic efficiency (η_{GT}), combustion chamber inlet temperature (T_3), and turbine inlet temperature (T_4). The optimal values of decision variables were obtained by minimizing the objective function using Genetic Algorithm method. In summary, the following are the specific contributions of this paper to the subject area:

- A complete thermodynamic modeling of a MGT system is performed.
- A new objective function, including the cost of environmental impacts (particularly for NOx and CO), is considered.
- The genetic algorithm method is used for optimization.
- The exergy analysis is applied to the MGT system.

ENERGY ANALYSIS

As MGT systems are commonly used for many applications, the optimization of such systems is so important in both thermodynamic and economic point of view. In addition, exergoeconomic analysis helps designers to find ways to improve the performance of a system in a cost-effective way. Most of the conventional exergoeconomic optimization methods are iterative in nature and require the interpretation of the designer at each iteration.

To find the optimum physical and thermal design parameters of the system, a simulation program is developed in Matlab software. Thus, the temperature profile in MGT plant, and the input and output exergy of each line in the plant were estimated to study the optimization of the plant. The energy balance equations for various parts of the plant (Fig. 1) are as follows:

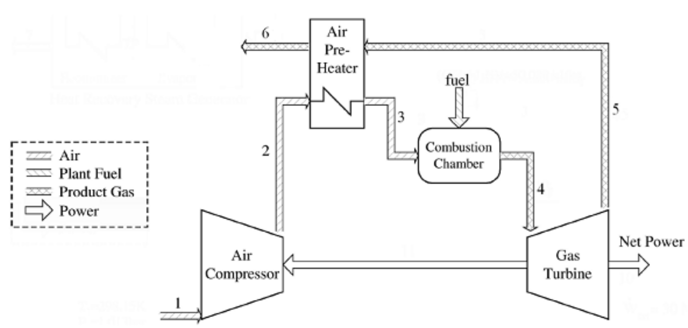

Figure 1. SCHEMATIC DIAGRAM OF THE MGT PLANT

- Air compressor:

$$T_2 = T_1\left\{1+\frac{1}{\eta_{AC}}\left[r_{AC}^{(\gamma_a-1)/\gamma_a}-1\right]\right\} \quad (1)$$

$$\dot{W}_{AC} = \dot{m}_a C_{P,a}(T_2 - T_1) \quad (2)$$

where r_{AC} is the compressor pressure ratio (=P_2/P_1) and according to inlet air, γ_a=1.4 and $C_{P,a}$=1.004 kJ/kg.K.

- Recuperator:

$$P_3 = P_2(1-\Delta P_{a,rec}) \quad (3)$$

$$P_6 = P_5(1-\Delta P_{g,rec}) \quad (4)$$

$$\dot{m}_a C_{P,a}(T_3 - T_2) = \dot{m}_g C_{P,g}(T_5 - T_6)\eta_{rec} \quad (5)$$

- Combustion chamber:

$$\dot{m}_a h_3 + \dot{m}_f \mathrm{LHV} = \dot{m}_g h_4 + \dot{m}_f \mathrm{LHV}(1-\eta_{CC}) \quad (6)$$

$$P_4 = P_3(1-\Delta P_{CC}) \quad (7)$$

where $h_3 = C_{P,a}(T_3\text{-}T_0)$, $h_4 = C_{P,g}(T_4\text{-}T_0)$, in which T_0 is 298.15 K; Also, η_{CC} = 0.98 is the efficiency of the combustion chamber. Moreover, The reference state is P_0=1.013 bar and T_0=298.15 K.

In this paper, the lower heating value (LHV) is used, which does not include the heat of condensation of the water vapor in the combustion products and it is 50,000 kJ/kg for methane.

With the following combustion equation:

$$\begin{aligned}&\lambda CH_4 + (x_{O_2}O_2 + x_{N_2}N_2 + x_{H_2O}H_2O + x_{CO_2}CO_2)\\ &\quad\rightarrow y_{CO_2}CO_2 + y_{N_2}N_2 + y_{O_2}O_2 + y_{H_2O}H_2O\\ &\quad+ y_{NO}NO + y_{CO}CO\end{aligned} \quad (8)$$

where (M_f = 16.043 and M_a = 28.254 kg/kmol):

$$\lambda = \frac{\dot{n}_f}{\dot{n}_a} = \frac{\dot{m}_f}{\dot{m}_a}\frac{M_a}{M_f} \quad (9)$$

- Gas turbine:

$$T_5 = T_4\left\{1-\eta_{GT}\left[1-\left(\frac{P_4}{P_5}\right)^{(1-\gamma_g)/\gamma_g}\right]\right\} \quad (10)$$

$$\dot{W}_{GT} = \dot{m}_g C_{P,g}(T_4 - T_5) \quad (11)$$

where γ_g=1.33 is the specific heat ratio for combustion gases and $C_{P,g}$=1.17 kJ/kg.K.

Using the fact that $\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC}$, the fuel mass rate is estimated from:

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} = (\dot{m}_a + \dot{m}_f)C_{P,g}(T_4 - T_5) - \dot{m}_a C_{P,a}(T_2 - T_1) \quad (12)$$

These combinations of energy- and mass-balance equations were numerically solved and the temperature and pressure of each line of the plant were predicted.

It should be noted that the utilized thermodynamic mode is developed based on the following basic assumptions:

1) All the processes in our study are considered based on the steady-state model.
2) The principle of ideal gas mixture is applied for the air and combustion products.
3) The fuel injected to the combustion chamber is assumed to be methane.
4) Kinetic and potential energy and exergy are assumed to be negligible as the elevation and speed have negligible changes. Also bearing losses are considered negligible.
5) Heat loss from the combustion chamber is considered to be 3% of the fuel lower heating value.
6) In the recuperator, 3% pressure drop is considered in gas side and 5% in air side. In addition, 5% pressure drop is considered in the combustion chamber.

EXERGY ANALYSIS

Exergy can be divided into four distinct components. The two important ones are the physical exergy and chemical exergy. In this study, the two other components which are kinetic exergy and potential exergy are considered negligible as the elevation and speed have negligible changes [21,22]. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state [23]. The chemical exergy is associated with the departure of the chemical composition of a system from its chemical equilibrium. The chemical exergy is an important part of exergy in combustion process. Therefore, the following exergy balance equation is written:

$$\dot{E}x_Q + \sum_{in}\dot{m}_{in}ex_{in} = \dot{E}x_W + \sum_{out}\dot{m}_{out}ex_{out} + \dot{E}x_D \quad (13)$$

where $\dot{E}x_D$ is the exergy destruction, other terms in this equation are as follows:

$$\dot{E}x_Q = \left(1-\frac{T_0}{T_i}\right)\dot{Q}_i \quad (14)$$

$$\dot{E}x_W = \dot{W} \quad (15)$$

$$ex = ex_{ph} + ex_{ch} \quad (16)$$

where $\dot{E}x_Q$ and $\dot{E}x_W$ are the corresponding exergy of heat transfer and work which cross the boundaries of the control volume, T is the absolute temperature (K) and (0) refers to the ambient conditions respectively. In equation (13), term $\dot{E}x$ is defined as $\dot{E}x = \dot{m}ex$.

The physical exergy and the chemical exergy of the mixture is defined as follows [22,23]:

$$ex_{ph} = (h - h_0) - T_0(S - S_0) \tag{17}$$

$$ex_{ch}^{mix} = \sum_{i=1}^{n} x_i ex_{ch_i} + RT_0 \sum_{i=1}^{n} x_i \mathrm{Ln} x_i + G^E \tag{18}$$

The last term, G^E, which is the excess free Gibbs energy, is negligible at low pressure at a gas mixture.

Here, for the exergy analysis of the plant, the exergy of each line is calculated at all states and the changes in the exergy are determined for each major component. The source of exergy destruction (or irreversibility) in combustion chamber is mainly combustion (chemical reaction) and thermal losses in the flow path respectively. However, the exergy destruction in the heat exchanger of the system i.e. recuperator is due to the large temperature difference between the hot and cold fluid. The exergy destruction rate and the exergy efficiency for each component for the whole system (Fig. 1) are shown in Table 1.

Table 1. THE EXERGY DESTRUCTION RATE AND EXERGY EFFICIENCY EQUATIONS FOR PLANT COMPONENTS

System component	Exergy destruction	Exergy efficiency
Air compressor	$\dot{E}x_{D,AC} = \dot{E}x_1 - \dot{E}x_2 + \dot{W}_{AC}$	$\eta_{ex,AC} = \frac{\dot{E}x_2 - \dot{E}x_1}{\dot{W}_{AC}}$
Combustion chamber	$\dot{E}x_{D,CC} = \dot{E}x_3 + \dot{E}x_f - \dot{E}x_4$	$\eta_{ex,CC} = \frac{\dot{E}x_4}{\dot{E}x_3 + \dot{E}x_f}$
Gas turbine	$\dot{E}x_{D,GT} = \dot{E}x_4 - \dot{E}x_5 - \dot{W}_{GT}$	$\eta_{ex,GT} = \frac{\dot{W}_{GT}}{\dot{E}x_4 - \dot{E}x_5}$
Recuperator	$\dot{E}x_{D,\mathrm{Rec}} = \sum_{i,\mathrm{Rec}} \dot{E}x - \sum_{e,\mathrm{Rec}} \dot{E}x$	$\eta_{ex,\mathrm{Rec}} = 1 - \frac{\dot{E}x_{D,\mathrm{Rec}}}{\sum_{i,\mathrm{Rec}} \dot{E}x}$

where, $\dot{E}x_f$ is the exergy rate of fuel injected to combustion chamber ($\dot{E}x_f = \dot{m}_f ex_f$) in which ex_f = 51850 kJ/kg [22].

ECONOMIC ANALYSIS

The total cost rate of operation for the installation is obtained from:

$$\dot{C}_T = \dot{C}_F + \sum_k \dot{Z}_k \tag{19}$$

Where $\dot{C}_T$ is the total cost rate of fuel and equipment ($/s) and $\dot{Z}_k$ is the capital cost rate ($/s) of the kth equipment item.

$$\dot{Z}_k = Z_k \mathrm{CRF}\varphi / N \tag{20}$$

Where Z_k, CRF, N (8000 h) and φ (1.06) are the purchase cost of kth component in dollar, the capital recovery factor, the annual number of the operation hours of the unit and the maintenance factor [22,24].

$$CRF = \frac{i(i+1)^n}{(i+1)^n - 1} \tag{21}$$

It is worth mentioning that the capital recovery factor is an economical parameter which depends on the interest rate (i) as well as estimated equipment life time (n). Considering the values of i and n to be 16% and 15 years respectively, CRF would be 18.2%.

The cost rate associated with fuel is obtained from:

$$\dot{C}_\mathrm{F} = c_\mathrm{f} \dot{m}_\mathrm{f} \times \mathrm{LHV} \tag{22}$$

where c_f = 0.004 $/MJ is the regional cost of fuel per unit of energy (on LHV basis) [22,23], $\dot{m}_f$ is the fuel mass flow rate, and LHV is the lower heating value of methane.

Table 2. EQUATIONS FOR CALCULATING THE PURCHASE COSTS (Z) FOR THE SYSTEM COMPONENTS [24]

System component	Capital or investment cost functions
Air compressor	$Z_{AC} = \left(\frac{C_{11}\dot{m}_a}{C_{12} - \eta_{AC}}\right)\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right)$
Combustion chamber	$Z_{CC} = \left(\frac{C_{21}\dot{m}_a}{C_{22} - \frac{P_4}{P_3}}\right)\left[1 + e^{(C_{23}T_4 - C_{24})}\right]$
Gas turbine	$Z_{GT} = \left(\frac{C_{31}\dot{m}_g}{C_{32} - \eta_{GT}}\right)\ln\left(\frac{P_4}{P_5}\right)\left[1 + e^{(C_{33}T_4 - C_{34})}\right]$
Recuperator	$Z_{REC} = C_{41}\left(\frac{\dot{m}_g C_{p,g}(T_5 - T_6)}{U\left[\frac{(T_6 - T_2) - (T_5 - T_3)}{\log((T_6 - T_2)/(T_5 - T_3))}\right]}\right)^{0.6}$

where the constants (C) are obtained from Refs. [22,24].

ENVIRONMENTAL ANALYSIS

In order to minimize the environmental impacts, a primary target is to increase the efficiency of energy conversion processes and, thus, decrease the amount of fuel and the related overall environmental impacts, especially the release of carbon dioxide as a major greenhouse gas. Therefore, optimization of thermal systems based on this fact has been an important subject in recent years. Although there are many papers in the literature, dealing with optimization of CHP plants, they consider no environmental impacts. For this reason, one of the major goals of the present work is to take the environmental impacts as producing the CO and NOx into account. As it is discussed in Ref. [25], the adiabatic flame temperature in the primary zone of the combustion chamber is derived as follow:

$$T_{pz} = A\sigma^{\alpha} \exp\left(\beta(\sigma+\lambda)^2\right)\pi^x\theta^y\psi^z \tag{23}$$

Where π is a dimensionless pressure p/p_{ref} (p being the combustion pressure p_3, and p_{ref} = 101300 Pa); θ is a dimensionless temperature T/T_{ref} (T being the inlet temperature T_3, and T_{ref} = 300 K); ψ is the H/C atomic ratio (ψ = 4, the fuel being pure methane); $\sigma = \phi$ for being the fuel to air equivalence ratio (ϕ is assumed constant); It is considered 0.64 for the combustion equation with the fuel of methane.

x, y and z are quadratic functions of σ; A, α, β and λ are constants (different sets of constants are used for different ranges of ϕ and θ).

$$x = a_1 + b_1\sigma + c_1\sigma^2 \tag{24}$$

$$y = a_2 + b_2\sigma + c_2\sigma^2 \tag{25}$$

$$z = a_3 + b_3\sigma + c_3\sigma^2 \tag{26}$$

The adiabatic flame temperature is used in the semi analytical correlations proposed by Rizk and Mongia [26] to determine the pollutant emissions in grams per kilogram of fuel:

$$\dot{m}_{NO_x} = \frac{0.15\times10^{16}\tau^{0.5}\exp\left(-71100/T_{pz}\right)}{P_3^{0.05}\left(\Delta P_3/P_3\right)^{0.5}} \tag{27}$$

$$\dot{m}_{CO} = \frac{0.179\times10^{9}\exp\left(7800/T_{pz}\right)}{P_3^{2}\tau\left(\Delta P_3/P_3\right)^{0.5}} \tag{28}$$

Where τ is the residence time in the combustion zone (τ is assumed constant and is equal to 0.002 s); T_{pz} is the primary zone combustion temperature; p_3 is the combustor inlet pressure; $\Delta p_3/p_3$ is the non-dimensional pressure drop in the combustor.

OBJECTIVE FUNCTION

The objective function in this paper is defined as the sum of three parts; the operational cost rate, which is related to the fuel expense, the rate of capital cost which stands for the capital investment and maintenance expenses, and the cost of environmental impacts (NOx and CO). Therefore, the objective function represents total cost rate of the plant in terms of dollar per unit of time.

$$OF = \dot{C}_T + \dot{C}_{env} = \dot{C}_F + \sum_k \dot{Z}_k + \dot{C}_{env} \tag{29}$$

where:

$$\dot{C}_{env} = \dot{m}_{CO}C_{CO} + \dot{m}_{NO_x}C_{NO_x} \tag{30}$$

$$C_{CO} = 0.02086\ \$/\text{kgCO} \tag{31}$$

$$C_{NO_x} = 6.853\ \$/\text{kgNO}_x \tag{32}$$

The last part of the objective function (OF) expresses the environmental impact as the total pollution damage ($/s) due to CO and NOx emission by multiplying their respective flow rates by their corresponding unit damage cost (C_{CO}, C_{NOx} are equal to 0.02086 $\$/kg_{CO}$ and 6.853 $\$/kg_{NOx}$) [23].

In the present work, the cost of pollution damage is assumed to be added directly to the expenditures that must be paid. Since the amount of ultimate products (net power) is fixed, the objective function is to be minimized so that the values of optimal design parameters would be obtained.

Decision variables:

The decision variables (design parameters), which are considered in this study, are as follows: compressor pressure ratio (r_c), compressor isentropic efficiency (η_{AC}), gas turbine isentropic efficiency (η_{GT}), combustion chamber inlet temperature (T_3), and turbine inlet temperature. Even though the decision variables may be varied in the optimization procedure, each decision variables is normally required to be within a reasonable range. The list of these constraints and the reasons of their applications are briefed based on Refs. [39,40] and listed in Table 3.

Table 3. THE LIST OF CONSTRAINTS

Constraints	Reason
$T_4 < 1600$	Material limitation
$r_c < 6$	Commercial availability
$\eta_{AC} < 0.9$	Commercial availability
$\eta_{GT} < 0.93$	Commercial availability

Based on Fig. 1, the following constraints should be satisfied in heat exchanger (recuperator):

$$T_3 > T_2; T_6 > T_2; T_5 > T_3 \tag{33}$$

GENETIC ALGORITHM

In recent years, optimization algorithms have received increasing attention by the research community as well as the industry. Evolutionary algorithms (EAs) are highly relevant for industrial applications, because they are capable of handling problems with non-linear constraints and multiple objectives.

Genetic algorithms (GAs) are an optimization technique based on natural genetics. GAs were developed by Holland [30] in an attempt to simulate growth and decay of living organisms in a natural environment. Even though originally designed as simulators, GAs proved to be a robust optimization technique. The term robust denotes the ability of the GAs for finding the global optimum, or a near-optimal point, for any optimization problem. The basic idea behind GAs could be described in brief as follows. A set of points inside the optimization space is created by random selection of points. Then, this set of points is transformed into a new one. Moreover, this new set will contain more points that are closer to the global optimum. The transformation procedure is based only on the information of how optimal each point is in the set, consists of very simple string manipulations, and is repeated several times. This simplicity in application and the fact that the only information necessary is a measure of how optimal each point is in the optimization space, make GAs attractive as optimizers. Nevertheless, the major advantages of the GAs are the following:

- Constraints of any type can be easily implemented.
- GAs usually find more than one near-optimal point in the optimization space, thus permitting the use of the most applicable solution for the optimization problem at hand.

Therefore, GAs generate solutions to optimization problems using techniques inspired by natural evolution, such as selection, crossover and mutation. In genetic algorithm, crossover combines two individuals, or parents, to form a new individual, or child, for the next generation. Mutation functions make small random changes in the individuals in the population, which provide genetic diversity and enable the genetic algorithm to search a broader space.

In this paper, a population size of 300 has been chosen in all runs with crossover probability P_c and mutation probability P_m as 0.9 and 0.02, respectively.

RESULTS AND DISCUSSION (OPTIMIZATION)

For the required power output of the MGT plant (100kW), the obtained numerical values of the optimum design parameters are reported in Table 4. Furthermore, the corresponding numerical values of selected dependent variables are listed in Table 5 and 6.

Table 4. OPTIMUM VALUES OF THE MGT PLANT

Decision variable	Optimum design values using GA
r_{AC}	4.502
η_{AC}	0.801
T_3 (K)	906.48
T_4 (K)	1213.99
η_{GT}	0.839
Objective function ($/h)	7.3491

Table 5. VALUES OF THE TEMPERATURE AND PRESSURE FOR THE STREAM IN THE OPTIMAL DESIGN OF PLANT

Flow	Temperature (K)	Pressure (bar)
1	298.15	1.013
2	498.04	4.561
3	906.48	4.333
4	1213.99	4.116
5	929.52	1.099
6	582.37	1.066

Table 6. NUMERICAL VALUES OF SELECTED DEPENDENT VARIABLES IN THE OPTIMAL DESIGN

$\dot{m}_a$ (kg/s)	0.7388
$\dot{m}_f$ (kg/s)	0.007
$\dot{W}_{AC}$ (kW)	148.27
$\dot{W}_{GT}$ (kW)	248.27

According to equations (20, 22 and 30), Table 7 shows the optimal cost values in the plant.

Table 7. OPTIMAL COST RATES (OPERATING, INVESTMENT AND POLLUTION)

	Cost rate ($/h)	Percentage (%)
Air compressor	0.6481	8.82
Combustion chamber	0.0703	0.96
Gas turbine	0.7200	9.82
Recuperator	0.6512	8.86
Fuel cost rate	5.04	68.58
Environmental pollutions	0.2194	2.96
Overall plant	7.3491	100

As it is shown in Table 7, fuel cost rate is comprised almost 69% of total cost rate. Therefore, it is the most important part in the economic objects.

EXERGY RESULTS

Exergy-based performance analysis for the MGT has been carried out in this study. It is known that exergy is an excellent tool to analyze the cause of performance deterioration in MGT components, by investigating behaviors of the exergy-related parameters, such as exergetic efficiency and exergy destruction. The performance deterioration of the compressor is related to the increase of exergy destruction. In spite of its positive contribution to the performance enhancement of the whole micro gas turbine, the blade cooling air in the turbine plays an important role in exergy destruction. It was confirmed that the exergy efficiency and exergy destruction in the combustion chamber are mainly affected by this parameter. Considerable exergy destruction occurs in the combustion chamber only, and therefore, both the exergy efficiency and the exergy destruction in the plant are affected mostly by the turbine inlet temperature.

Table 8 shows the physical and chemical exergies of MGT system.

Table 8. EXERGY DATA FOR THE MGT PLANT OF FIGURE. 1.

state	Exergy Rates (kW)		
	$\dot{E}x_{ph}$	$\dot{E}x_{ch}$	$\dot{E}x_{T}$
1	0	0	0
2	129.9148	0	129.9148
3	297.1919	0	297.1919
4	524.3508	3.3010	527.6518
5	260.4202	3.3010	263.7212
6	77.1485	3.3010	80.4495

The exergy destruction is summarized in Table 9, clearly identifies the combustion chamber as the major site of thermodynamic inefficiency. Roughly equal contributions to inefficiency are made by the gas turbine and recuperator. Air compressor is an only slightly smaller contributor.

Table 9. EXERGY DESTRUCTION DATA FOR THE PLANT

Component	Exergy Destruction	
	Rate (kW)	Percentage
Air compressor	15.3570	8.30
Combustion chamber	137.8179	74.56
Recuperator	15.9946	8.65
Gas turbine	15.6668	8.49
Overall plant	187.8363	100

The values of exergetic efficiency of combustion chamber is lower than that of other components, and can be increased by increasing the combustion inlet temperature (T_3) and turbine inlet temperature (T_4). However, it should be noted that due to physical constraints and capital cost limitations, these temperatures can be changed only within allowable extents. This means that the improvement of the exergetic efficiency by increasing T_3 and T_4 may move the design point from the optimum situation to a new situation at which, the objective function is not minimum.

Figure 2 shows the exergetic efficiency values of each components of the system. The exergetic efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy. Overall plant exergy efficiency is 29.7% whereas the energy efficiency for total cycle is 28.57%.

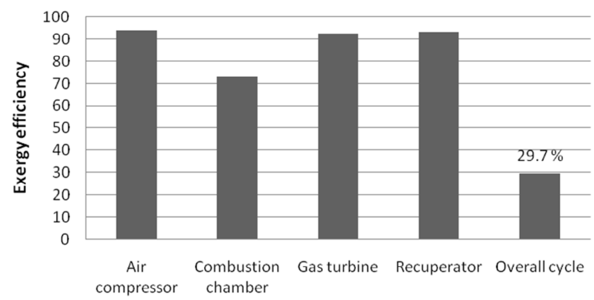

Figure 2. Exergy efficiency of each component of MGS.

CONCLUSION

In the present research, the exergoenvironmental analysis and optimization of a typical 100kW MGT plant are carried out using Genetic Algorithm. At the first part of the paper, the thermodynamic modeling of a MGT plant is done. Moreover, the optimization of MGT plant is performed to find the optimal design parameters of the cycle. The new objective function, including total cost of the plant as well as cost of environmental impacts, is considered. The exergy balance applied to a process or the whole plant tells us how much of the usable work potential, or exergy supplied as the input to the system under consideration has been consumed by the process.

NOMENCLATURE

c_f	cost of fuel per energy unit [$/MJ]
C	cost flow rate [$/s]
C_p	specific heat at constant pressure [kJ/kg K]
CRF	capital recovery factor
$\dot{E}x$	exergy flow rate [kW]

$\dot{E}x_D$	exergy destruction rate [kW]
h	enthalpy (kJ/kg)
LHV	lower heating value [kJ/kg]
$\dot{m}$	mass flow rate [kg/s]
r_{AC}	compressor pressure ratio
R	gas constant (kJ/kg K)
S	entropy (kJ/kg K)
$\dot{W}_{net}$	net power output [kW]
Z	capital cost of a component [$]
$\dot{Z}$	capital cost rate [$/s]
ΔP	pressure loss
η_{AC}	compressor isentropic efficiency
η_{CC}	combustion chamber first law efficiency
η_{GT}	gas turbine isentropic efficiency
γ	specific heat ratio
x_i	molar fraction

Subscripts

a	Air
AC	air compressor
CC	combustion chamber
f	Fuel
g	combustion gases
GT	gas turbine
REC	Recuperator

REFERENCES

[1] Hamilton, S.L., 2003, The Handbook of Microturbine Generators, PennWell Corporation, Tulsa.
[2] Opdyke, D.E. and Franus, D.J., 2004, “Gas turbine industry set to rebound”, *Turbomachinery International Handbook* 44.
[3] Technology characterization–microturbine, 2002, Energy Nexus Group, USA Environmental Protection Agency.
[4] Sun, Z., and Xie, N., 2010, “Experimental studying of a small combined cold and power systems driven by a micro gas turbine”, *Int. J. Applied Thermal Engineering*.
[5] Sanaye, S., Aghaei Meybodi, M. and Shokrollahi, S., 2008, "Selecting the prime movers and nominal powers in combined heat and power systems", *International Journal of Applied Thermal Engineering*, Vol. 28, pp. 1177-1188.
[6] Hinnells, M., 2008, "Combined heat and power in industry and buildings", *Int. J. Energy Policy*, Vol. 36, pp. 4522-4526.
[7] Staunton, R.H. and Ozpineci, B., 2003, "Microturbine power conversion technology review", Oak Ridge National Laboratory, Washington DC, Tech. Rep. ORNL/TM-2003/74.
[8] Pilavachi, P.A., 2002, "Mini- and micro- turbines for combined heat and power", *Int. J. Applied Thermal Engineering*, Vol. 22, pp. 2003-2014.
[9] Jurado, F., 2005, "Modeling Micro-Turbines Using Hammerstein Models", *Int. J. Energy Research*, Vol. 29, pp. 841-855.
[10] Jurado, F., 2005, "Non-Linear Modeling of Micro-Turbines Using NARX Structures on the Distribution Feeder", *Int. J. Energy Conversion and Management*, Vol. 46, pp. 385-401.
[11] Ehyaei, M.A. and Bahadori, M.N., 2007, "Selection of micro turbines to meet electrical and thermal energy needs of residential buildings in Iran", *Int. J. Energy and Buildings*, Vol. 39, pp. 1227-1234.
[12] Ehyaei, M.A. and Mozafari, A., 2009, "Energy, economic and environmental (3E) analysis of a micro gas turbine employed for on-site combined heat and power production", *Int. J. Energy and Buildings*, pp. 1-6.
[13] Zaltash, A., Petrov, A.Y., Rizy, D.T., La Vineyard, E.A. and Linkous, RL., 2003, “Laboratory R&D on Integrated Energy Systems”, *Proceedings of International Congress of Refrigeration*, pp.1-8.
[14] Labinov, S.D., Zaltash, A., Rizy, D.T., Fairchild,P.D., Devault, R.C. and Vineyard, E.A., 2002, "Predictive Algorithms for microturbine performance for BCHP systems", *J. ASHRAE Transactions*, Vol. 108, pp. 670-681.
[15] Aphornratana, S., Eames, I.W., 1995. Thermodynamic analysis of absorption refrigeration cycles using the second law of thermodynamics method. International Journal of Refrigeration 18, 244–252.
[16] Bonnet, S., Alaphilippe, M. and Stouffs, P., 2005, “Energy, exergy and cost analysis of a micro-cogeneration system based on an Ericsson engine”, *Int. Journal of Thermal Sciences*, 44: 1161–1168.
[17] Rivero, R. and Garfias, M., 2006, “Standard chemical exergy of elements updated”, *Energy*, 31: 3310–3326.
[18] Huang, F.F., 1990. “Performance evaluation of selected combustion gas cogeneration systems based on first and second law analysis”, *ASME Transactions Journal of Engineering for Gas Turbines and Power 112*, 117–121.
[19] Cihan A, Hacıhafizoglu O and Kahveci K., “Energyeexergy analysis and modernization suggestions for a combined-cycle power plant”, Int. *Journal of Energy Research* 2006; 30:115-26.
[20] P. Regulagadda, I. Dincer, G.F. Naterer, 2010, “Exergy analysis of a thermal power plant with measured boiler and turbine losses”, *Applied Thermal Engineering 30*, 970–976.
[21] Kotas Tj. The exergy method of thermal plant analysis. London: Butterworths; 1985.
[22] Bejan A, Tsatsaronis G, Moran M. Thermal design and optimization. New York: Wiley; 1996.
[23] Dincer I, Rosen MA. Exergy: energy, environment and sustainable development. Elsevier; 2007.

[24] Valero, A., Lozano, M. and Serra, L., 1994, CGAM problem: definition and conventional solution, Int. J. Energy 19(3):279–86.
[25] Gülder OL., 1986, "Flame temperature estimation of conventional and future jet fuels", *Journal of Engineering for Gas Turbine and Power*, Vol. 108 (2): 376–80.
[26] Rizk NK, Mongia HC., 1993, "Semi analytical correlations for NOx, CO and UHC emissions", *Journal of Engineering for Gas Turbine and Power*, Vol. 115 (3):612-9.
[30] Holland JH., Adaptation in natural and artificial systems. Ann Arbor: The University of Michigan Press; 1975.