THERMO-ECONOMIC-ENVIRONMENTAL OPTIMIZATION OF A MICROTURBINE USING GENETIC ALGORITHM

by

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Thermo-economic-environmental optimization of a 100 kW microturbine has been numerically investigated by genetic algorithm optimization method. An objective function is defined as the sum of the total cost of the plant and the costs of environmental pollutant effects (the emission of NO_x and CO gases due to fuel combustion), while the design parameters are the common important parameters of an industrial power plant cycle. The objective function is then formulated in the design parameters. Finally, the optimum values of the parameters are computed by minimizing the objective function using the genetic algorithm.

Key words: *microturbine, optimization, genetic algorithm, exergy, CGAM*

Introduction

Due to strict power quality, new approaches to power generation have evolved. One of these approaches is a micro gas turbine that is typically single-shaft engine with no gearbox. Micro gas turbines (MGT) burn gaseous and liquid fuels to create high speed rotation that turns an electrical generator. They are expected to have steady growth in the 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 350 kW, while conventional gas turbine sizes range from 500 kW to 250 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 90,000 rpm and generate electrical power of high and variable frequency (alternating current – AC). This power is rectified to direct current (DC) and then inverted to 60 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 and also

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optimize operation modes of co-generation or tri-generation system, a simple but enough accurate performance model of microturbine is desired. Many researchers have engaged on developing mathematic model for gas turbine. So many models were developed. A good review on the mathematical models was given by Jurado [9]. But among the 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 the 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. Pourhasanzadeh [11] studied the performance analysis of a microturbine for CHP production. Ehyaei and Bahadori [12] 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, three cities in Iran.

Much useful research work in developing models for building the cooling, heating, and power systems has been done by Zaltash [13]. Based on experimental data of a commercially available microturbine, a semi-empirical model was developed by Labinov [14]. In his model, the efficiencies of the turbine, the compressor, and the recuperator were regarded as constants and the thermophysical properties of the air and the flue gas were assumed to be not changed, 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 co-generation system. He observed the effects of pinch point temperature and process steam pressure on the energetic and exergetic performance of the system. Cihan *et al*. [19] performed energy and exergy analysis for a combined-cycle power plant to analyze and identify the potential for improving efficiency of the system. 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 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 1990, a group of concerned specialists in the field of thermoeconomics (C. Frangopoulos, G. Tsatsaronis, A, Valero, and M. von Spakovsky) decided to compare their methodologies by solving a predefined and simple problem of optimization: the CGAM problem which was named after the first initials of the participating investigators. They described and defined the CGAM problem and presented a conventional solution to its optimization problem [21]. The CGAM problem refers to a co-generation plant which delivers 30 MW of electricity and 14 kg/s of saturated steam at 20 bar. The models used in the CGAM problem are realistic but incomplete from an engineering point of view since the object of this study is to present distinct models of thermo-economic optimization. Therefore, it would be unreasonable to use an excessively complicated mathematical model to describe the performance of the plant.

In the present study, a new methodology based on the CGAM problem is developed to optimize the objective function by implementation of genetic algorithm (GA). GA is an optimization technique based on natural genetics. For the verification of this model, the results are compared against those of CGAM problem. In the next step, the GA code is used for the optimization of a 100 kW microturbine plant system. The new objective function which includes the total cost rate of the product and the cost rate of the environmental impact is considered. The design parameters are considered as the compressor pressure ratio (r_{AC}), the compressor isentropic efficiency (η_{AC}), the gas turbine isentropic efficiency (η_{GT}), the combustion chamber inlet temperature (T_3), and the turbine inlet temperature (T_4), as shown in fig. 1. The optimal values of decision variables are obtained by minimizing the objective function using GA. In summary, the following are the specific contributions of this paper to the subject area:

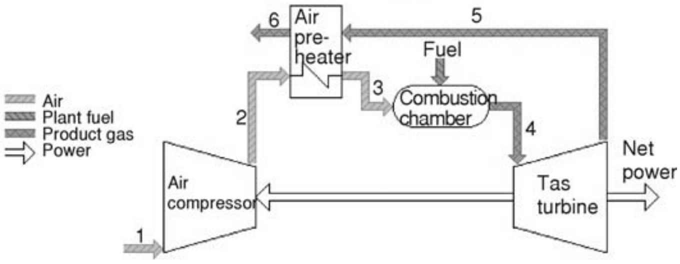

Figure 1. Schematic diagram of the MGT plant (subsequent numbers starting from 1 to 6 are thermodynamic states of the cycle)

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

Energy analysis

Since 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, exergo-economic analysis helps designers to find ways to improve the performance of a system in a cost-effective way. Most of the conventional exergo-economic 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 are estimated to study the optimization of the plant. The energy balance equations for various parts of the plant (fig. 1) are:

– air compressor

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

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

– combustion chamber

$$\dot{m}_a h_3 + \dot{m}_f LHV = \dot{m}_g h_4 + \dot{m}_f LHV (1 - \eta_{CC}) \tag{3}$$

$$P_4 = P_3(1 - \Delta P_{CC}) \tag{4}$$

where $h_3 = C_{P,a}(T_3 - T_0)$ and $h_4 = C_{P,g}(T_4 - T_0)$, in which T_0 is 298.15 K and the efficiency of the combustion chamber (η_{CC}) is 0.98. The combustion equation is:

$$\begin{aligned} \lambda CH_4 + (x_{O_2}O_2 + x_{N_2}N_2 + x_{H_2O}H_2O + x_{CO_2}CO_2) \rightarrow y_{CO_2}CO_2 + \\ + y_{N_2}N_2 + y_{O_2}O_2 + y_{H_2O}H_2O + y_{NO}NO + y_{CO}CO \end{aligned} \tag{5}$$

$$\lambda = \frac{n_f}{n_a} = \frac{\dot{m}_f}{\dot{m}_a}\frac{M_a}{M_f} \tag{6}$$

where CH_4, is the injected fuel. x and y are the molar fraction of the reactants and combustion products; respectively. λ is the ratio of fuel to air on molar basis, [kmol/kmol]:

– gas turbine

$$T_5 = T_4\left\{1 - \eta_{GT}\left[1 - \left(\frac{P_4}{P_5}\right)^{(1-\gamma_g)/\gamma_g}\right]\right\} \tag{7}$$

$$\dot{W}_{GT} = \dot{m}_g C_{P,g}(T_4 - T_5) \tag{8}$$

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

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC} = (\dot{m}_a + \dot{m}_f)C_{P,g}(T_4 - T_5) - \dot{m}_a C_{P,a}(T_2 - T_1) \tag{9}$$

– recuperator

$$\dot{m}_a C_{P,a}(T_3 - T_2) = \dot{m}_g C_{P,g}(T_5 - T_6)\eta_{Rec} \tag{10}$$

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

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

– all the processes in this study are considered based on the steady-state model,
– the principle of ideal gas mixture is applied for the air and combustion products,
– the fuel injected to the combustion chamber is assumed to be methane,
– heat loss from the combustion chamber is considered to be 3% of the fuel lower heating value; moreover, all the other components are considered adiabatic,
– the reference state is $P_0 = 1.013$ bar and $T_0 = 298.15$ K, and
– in the recuperator, 3% pressure drop is considered in the gas side and 5% in the air side. In addition, 5% pressure drop is considered in the combustion chamber.

It is worth mentioning that all turbine and engine manufactures quote heat rates in terms of the lower heating value (*LHV*) of the fuel. On the other hand, the usable energy content of fuels and their prices are typically measured on a higher heating value (*HHV*) basis. In this paper, the equations are based on *LHV*, so an *LHV/HHV* factor is included (about 0.9 for methane) [3].

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 [22, 23]. The physical exergy is defined as the maximum theoretical useful work obtained as a system interacts with an equilibrium state [24]. 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:

$$\dot{E}x_{\mathrm{Q}} + \sum_{in} \dot{m}_{\mathrm{m}} ex_{\mathrm{in}} = \dot{E}x_{\mathrm{W}} + \sum_{out} \dot{m}_{\mathrm{out}} ex_{\mathrm{out}} + \dot{E}x_{\mathrm{D}} \tag{11}$$

where $\dot{E}x_{\mathrm{D}}$ is the exergy destruction rate. The other terms in this equation are:

$$\dot{E}x_{\mathrm{Q}} = \left(1 - \frac{T_0}{T_i}\right)\dot{Q}_i \tag{12}$$

$$\dot{E}x_{\mathrm{W}} = \dot{W} \tag{13}$$

$$ex = ex_{\mathrm{ph}} + ex_{\mathrm{ch}} \tag{14}$$

where $\dot{E}x_{\mathrm{Q}}$ and $\dot{E}x_{\mathrm{W}}$ are the corresponding exergy of heat transfer and work which cross the boundaries of the control volume, T [K] – the absolute temperature and 0 refers to the ambient conditions, respectively. In eq. (11), the term $\dot{m} \times ex$ is defined as $\dot{E}x$.

The physical and the chemical exergy of the mixture are defined as [23, 24]:

$$ex_{\mathrm{ph}} = (h - h_0) - T_0(S - S_0) \tag{15}$$

$$ex_{ch}^{mix} = \sum_{i=1}^{n} x_i ex_{\mathrm{ch}_i} + \mathrm{R}T_0 \sum_{i=1}^{n} x_i \ln x_i + G^E \tag{16}$$

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 tab. 1.

Table 1. The exergy destruction rate and exergy efficiency equations for plant components

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

Economic analysis

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

$$C_{\text{tot}} = \dot{C}_{\text{f}} + \sum_{k} \dot{Z}_{\text{k}} \tag{17}$$

where $\dot{C}_{\text{tot}}$ is the total cost rate of fuel and equipment in \$/s and $\dot{Z}_{\text{k}}$ is the cost rate in \$/s associated with capital investment and the maintenance costs for the k^{th} equipment item:

$$\dot{Z}_{\text{k}} = \frac{Z_{\text{k}}\text{CRF}\varphi}{N} \tag{18}$$

where Z_{k} is the purchase cost of the k^{th} component in dollar, which is presented in tab. 2. Moreover, CRF, N (= 8000 hours) and φ (= 1.06) are the capital recovery factor, the annual number of the operation hours of the unit and the maintenance factor, respectively [21, 23]:

$$\text{CRF} = \frac{i(i+1)''}{(i+1)'' - 1} \tag{19}$$

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:

$$\dot{C}_{\text{f}} = \dot{m}_{\text{f}} c_{\text{f}} LHV \tag{20}$$

where c_{f} = 0.004 \$/MJ is the regional cost of fuel per unit of energy (on an *LHV* basis) [23, 24], $\dot{m}_{\text{f}}$ is the fuel mass flow rate and LHV = 50000 kJ/kg for methane.

Table 2. Equations for calculating the purchase costs (*Z*) for the system components [21]

System component	Exergy destruction
Air compressor	$Z_{\text{AC}} = \left(\frac{C_{11}\dot{m}_{\text{a}}}{C_{12} - \eta_{\text{AC}}}\right)\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right)$
Combustion chamber	$Z_{\text{CC}} = \left(\frac{C_{21}\dot{m}_{\text{a}}}{C_{22} - \frac{P_4}{P_3}}\right)[1 + \text{e}^{(C_{23}T_4 - C_{24})}]$
Gas turbine	$Z_{\text{GT}} = \left(\frac{C_{31}\dot{m}_{\text{g}}}{C_{32} - \eta_{\text{GT}}}\right)\ln\left(\frac{P_4}{P_5}\right)[1 + \text{e}^{(C_{33}T_4 - C_{34})}]$
Recuperator	$Z_{\text{Rec}} = \left(\frac{\dot{m}_g C_{\text{P,g}}(T_5 - T_6)}{U\left[\frac{(T_6 - T_2) - (T_5 - T_3)}{\log(T_6 - T_2)/(T_5 - T_3)}\right]}\right)^{0.6}$

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 study is to take the environmental impacts as producing the CO and NO_x into account. As it is discussed by Gulder, the adiabatic flame temperature in the primary zone of the combustion chamber is derived as follow [25]:

$$T_{pz} = A\sigma^{\alpha} \exp[\beta(\sigma+\lambda)^2]\,\pi^x\theta^y\psi^z \tag{21}$$

where π is a dimensionless pressure P/P_{ref} (p being the combustion pressure P_3, and $P_{ref} = 101.3$ kPa); θ – the dimensionless temperature T/T_{ref} (T being the inlet temperature T_3, and $T_{ref} = 300$ K); ψ is the H/C atomic ratio ($\psi = 4$, the fuel being pure methane); $\sigma = \Phi$ for being the fuel to air equivalence ratio (Φ is assumed constant); 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), and

$$x = a_1 + b_1 s + c_1 s^2 \tag{22}$$

$$y = a_2 + b_2 s + c_2 s^2 \tag{23}$$

$$z = a_3 + b_3 s + c_3 s^2 \tag{24}$$

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:

$$\dot{m}_{NO_x} = \frac{0.15\cdot 10^{16}\,\tau^{0.5}\exp\left(\dfrac{71100}{T_{pz}}\right)}{P_3^{0.05}\left(\dfrac{\Delta P_3}{P_3}\right)^{0.5}} \tag{25}$$

$$\dot{m}_{CO} = \frac{0.179\cdot 10^{9}\exp\left(\dfrac{7800}{T_{pz}}\right)}{P_3^{2}\tau\left(\dfrac{\Delta P_3}{P_3}\right)^{0.5}} \tag{26}$$

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.

Optimization procedure

Objective function

The objective function in this study 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 (NO_x and CO). Therefore, the objective function represents total cost rate of the plant in terms of dollar per unit of time.

$$\text{Objective function} = \dot{C}_{tot} + \dot{C}_{env} = \dot{C}_f + \sum_k \dot{Z}_k + \dot{C}_{env} \tag{27}$$

where
$$\dot{C}_{\text{env}} = \dot{m}_{\text{CO}} C_{\text{CO}} + \dot{m}_{\text{NO}_x} C_{\text{NO}_x} \tag{28}$$

The last part of the objective function expresses the environmental impact as the total pollution damage in \$/s due to CO and NO_x emission by multiplying their respective flow rates by their corresponding unit damage cost ($C_{\text{CO}} = 0.02086$ \$/kg$_{\text{CO}}$ and $C_{\text{NO}_x} = 6.853$ \$/kg$_{\text{NO}x}$) [27].

In this study, 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: the compressor pressure ratio (r_c), the compressor isentropic efficiency (η_{AC}), gas turbine isentropic efficiency (η_{GT}), the combustion chamber inlet temperature (T_3), and the turbine inlet temperature (T_4). 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 and listed in tab. 3 [23, 28].

Table 3. The list of constraints

Decision variable	Optimum design values using GA
$T_4 < 1600$	Material limitation
$r_c < 6$	Commercial availability
$\eta_{\text{AC}} < 0.9$	Commercial availability
$\eta_{\text{GT}} < 0.93$	Commercial availability

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

$$T_3 > T_2, \quad T_6 > T_2, \quad T_5 > T_3, \quad T_4 > T_3 \tag{29}$$

Genetic algorithm

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

GA is an optimization technique based on natural genetics. GA were developed by Holland [29] in an attempt to simulate growth and decay of living organisms in a natural environment. Even though originally designed as simulators, GA proved to be a robust optimization technique. The term robust denotes the ability of the GA for finding the global optimum, or a near-optimal point, for any optimization problem. The basic idea behind GA 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 GA attractive as an optimizers. Nevertheless, the major advantages of the GA are as follows:
– constraints of any type can be easily implemented, and
– GA usually finds 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.

Results and discussion

Verification of optimization method

To the best knowledge of the author, the optimization of a 100 kW microturbine cycle has been not investigated before. Therefore, in order to ensure the validity of thermodynamic and economic modeling, as well as the optimization procedure (*i. e.*, GA), first a CHP unit with the same characteristics of classic well-known CGAM problem [21] is modeled and optimized by genetic algorithm method. The installation consisted of a gas turbine followed by an air preheater that used part of the thermal energy of the gases leaving the turbine, and a heat recovery steam generator in which the required steam was produced (fig. 2).

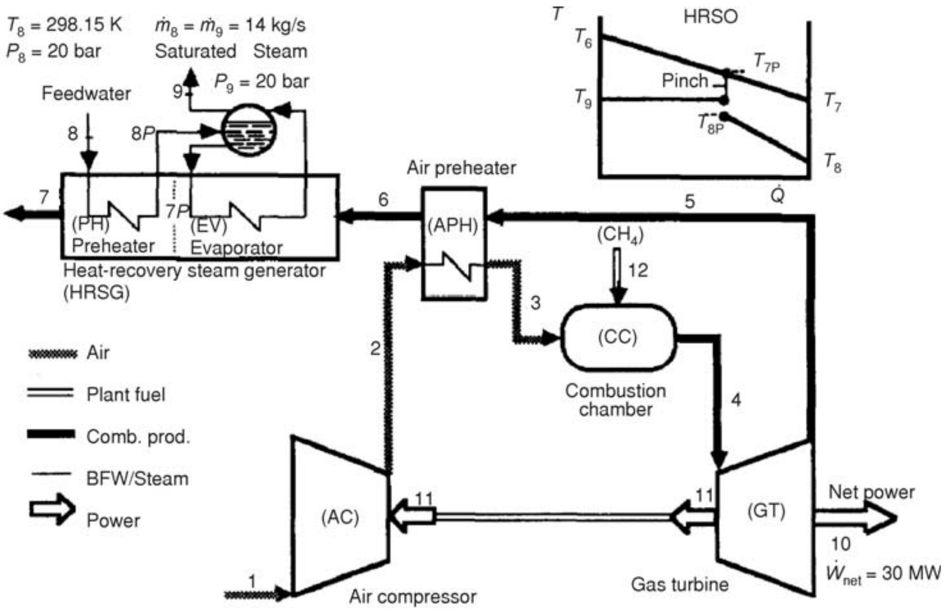

Figure 2. Flow diagram of the CGAM problem [21]

The following are the additional equations for HRSG part:

$$T_{8P} = T_9 - \Delta T_A;\ \ \Delta T_A = 15\ \mathrm{K} \tag{30}$$

$$T_7 = T_6 - \frac{\dot{m}_s (h_9 - h_8)}{\dot{m}_g C_{P,g}} \tag{31}$$

$$\dot{m}_g C_{P,g} (T_6 - T_{7P}) = \dot{m}_{st} (h_9 - h_{8P}) \tag{32}$$

where $\dot{m}_{st}$ = 14 kg/s is the saturated steam mass flow rate. Moreover, $h_9 - h_{8P}$ = 1956 kJ/kg and $h_9 - h_8$ = 2690 kJ/kg.

It is worth mentioning that the combustion equation in CGAM problem was assumed complete, shown as:

$$\begin{gathered}\lambda CH_4 + (x_{O_2} O_2 + x_{N_2} N_2 + x_{H_2O} H_2O + x_{CO_2} CO_2) \rightarrow (\lambda + x_{CO_2}) CO_2 + \\ + x_{N_2} N_2 + (x_{O_2} - 2\lambda) O_2 + (2\lambda + x_{H_2O}) H_2O\end{gathered} \tag{33}$$

In addition to the equations expressed, due to HRSG section, the following equations should be added to tab. 2 [21]:

$$Z_{\mathrm{HRSG}} = C_{51}\left[\left(\frac{\dot{m}_s(h_{8P}-h_9)}{\dfrac{(T_7-T_8)-(T_{7P}-T_{8P})}{\log\dfrac{T_7-T_8}{T_{7P}-T_{8P}}}}\right)^{0.8}+\left(\frac{\dot{m}_s(h_9-h_{8P})}{\dfrac{(T_{7P}-T_9)-(T_6-T_9)}{\log\dfrac{T_{7P}-T_9}{T_6-T_9}}}\right)^{0.8}\right]+C_{25}\dot{m}_\mathrm{s}+C_{53}\dot{m}_g^{1.2} \tag{34}$$

where C_{51} = 3650 \$/kWK, C_{52} = 11820 \$/kgs, and C_{53} = 658 \$/(kg/s)$^{1.2}$.

After applying these extra equations to our original model, the optimization code is performed and the results are compared with CGAM problem for verification purposes.

Table 4. The comparison of our simulation and optimization numerical output for CGAM problem with results reported in literature [21, 27]

Decision variable	Design values reported by [21]	Design values reported by [27]	Optimum design values using GA	Relative error	
				With [21]	With [27]
r_{AC}	8.5234	8.59859	7.553	0.1139	0.1216
η_{AC}	0.8468	0.84675	0.867	0.0239	0.0239
T_3 (K)	914.28	912.281	984.423	0.0767	0.0791
T_4 (K)	1492.63	1491.440	1475.84	0.0113	0.0105
η_{GT}	0.8786	0.8790	0.89	0.013	0.0125
Objective function in \$/s	0.3692	0.36204	0.3325	0.0994	0.0816

As shown in tab. 4 the results of the present model are in a good agreement in comparison with the classic work [21, 27] which ensures the correctness of the simulation code as well as GA code.

It should be noted that this difference between optimized values is just due to the optimization procedure. As GA is based on a random search, this difference is reasonable. Moreover, by applying genetic algorithm, the objective function decreases by a factor of 0.9 which is noticeable in total cost of the plant. Therefore, this verifies the validity of obtained global optimum as well as our simulation code.

Table 5. Optimization values of the MGT plant

Decision variable	Optimum design values using GA
r_{AC}	4.42
η_{AC}	0.821
T_3 [K]	906.32
T_4 [K]	1254.65
η_{GT}	0.854
Objective function [\$ per hour]	6.2818

Optimization results section

For the required power output of the MGT plant (100 kW), the obtained numerical values of the optimum design parameters are reported in tab. 5. Furthermore, the corresponding numerical values of the dependent variables are listed in tabs. 6 and 7.

Exergy results section

Exergy-based performance analysis for the MGT has been carried out in this study. It is known that the exergy is an excellent tool to analyze the cause of performance deterioration in MGT components, by investigating the behaviors of the exergy-related parameters, such as the 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.

The exergy destruction is summarized in tab. 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.

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 the 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 3 shows the exergetic efficiency values of each components of the system. The

Table 6. Values of the temperature and pressure for the stream in the optimal design of the plant

Flow	Temperature [K]	Pressure [bar]
1	298.15	1.013
2	488.8446	4.4775
3	906.32	4.2536
4	1254.65	4.0409
5	958.9140	1.0993
6	604.4348	1.0663

Table 7. Numerical values of the dependent variables in the optimal design

Decision variable	Optimum design values using GA
$\dot{m}_a$ [kgs^{-1}]	0.6377
$\dot{m}_f$ [kgs^{-1}]	0.0068
$\dot{W}_{AC}$ [kW]	122.0877
$\dot{W}_{GT}$ [kW]	222.9862

Table 8. Exergy data for the MGT plant of fig. 1

State	Exergy rates [kW]		
	$\dot{E}x_{ph}$	$\dot{E}x_{ch}$	$\dot{E}x_{total}$
1	0	0	0
2	129.9148	0	129.9148
3	297.1919	0	297.1919
4	524.3508	3.301	527.6518
5	260.4202	3.301	263.7212
6	77.1485	3.301	80.4495

Table 9. Exergy destruction data for the plant

Component	Exergy destruction	
	Rate [kW]	Percentage
Air compressor	18.357	9.77
Combustion chamber	137.8179	73.37
Gas turbine	15.9946	8.52
Recuperator	15.6668	8.34
Overall plant	187.8363	100

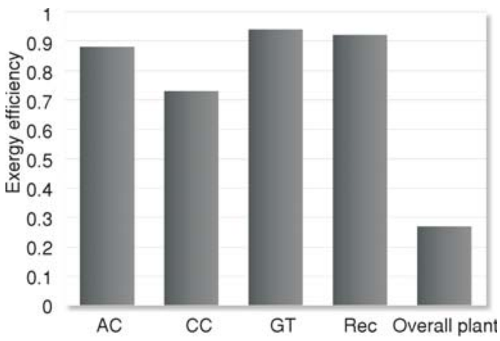

Figure 3. Exergy efficiency of each component of MGT

exergetic efficiency shows the percentage of the fuel exergy provided to a system that is found in the product exergy.

Conclusions

In the present research, the energy and exergy analysis and thermo-economic-environmental optimization of a typical 100 kW 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 – cost flow rate, [$\$s^{-1}$]
C_p – specific heat at constant pressure, [$kJkg^{-1}K^{-1}$]
c_f – cost of fuel per energy unit, [$\$MJ^{-1}$]
$\dot{E}x$ – exergy flow rate, [kW]
$\dot{E}x_D$ – exergy destruction rate, [kW]
h – specific enthalpy, [$kJkg^{-1}$]
M – molecular weights, [$kgkmol^{-1}$]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
ΔP – pressure drop
R – gas constant, [$kJkg^{-1}K^{-1}$]
r_{AC} – air compressor pressure ratio
S – entropy, [$kJkg^{-1}K^{-1}$]
$\dot{W}_{net}$ – net power output, [kW]
x – molar fraction of the reactants
y – molar fraction of the combustion products
Z – capital cost of a component, [\$]

Greek symbols

λ – ratio of fuel to air on molar basis, [–]
τ – residence time in the combustion zone, [s]

Subscripts

AC – air compressor
a – air
CC – combustion chamber
g – combustion gases
f – fuel
GT – gas turbine
Rec – recuperator
st – steam

Acronyms

CHP – combined heat and power
CRF – capital recovery factor, [–]
HRSG – heat recovery steam generator
LHV – lower heating value [$kJkg^{-1}$]

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