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A new approach to optimization of cogeneration systems using genetic algorithm

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ABSTRACT

Application of Cogeneration systems based gas turbine for heat and power production is increasing. Because of finite natural energy resources and increasing energy demand the cost effective design of energy systems is essential. CGAM problem as a cogeneration system is considered here for analyzing. Two new approaches are considered, first in thermodynamic model of gas turbine and cogeneration system considering blade cooling of gas turbine and second using genetic algorithm for optimization. The problem has been optimized from thermodynamic and thermoeconomic view point. Results show that Turbine Inlet Temperature (TIT) in thermodynamic optimum condition is higher than thermoeconomic one, while blade cooling technology must be better for optimum thermodynamic condition. Heat recovery of recuperator is lower in thermoeconomic case; also, stack temperature is higher relative to thermodynamic case. The sensitivity of the optimal solution to the decision variables is studied. It has been shown that while for both thermodynamic and thermoeconomic optimum condition, pressure ratio, blade cooling technology factor and pinch-point temperature difference (only for thermoeconomic case) has the lowest effect, turbomachinery efficiencies (epically compressor polytropic efficiency) have the major effect on performance of cycle. Finally; a new product known as Mercury 50 gas turbine is studied for a cogeneration system and it has been optimized thermoeconomicly. Results show good agreement with manufacturer data.

KEYWORDS

cogeneration system, CGAM, genetic algorithm, optimization

1. INTRODUCTION

In optimization of complex energy systems, the thermodynamic optimization aims to minimize the thermodynamic inefficiencies: exergy destruction and exergy losses via fuel mass flow minimization. This criterion leads to impractical solutions such as null pinch-point and infinite heat exchanger surface. The objective of thermoeconomic optimization, however, is to minimizing costs, including costs of thermodynamic inefficiencies and system capital cost.

In recent years, several efforts have been made to optimize the CGAM problem from thermoeconomic point of view.The CGAM problem refers to a cogeneration plant with 30MW power capacity and 14 kg/sec of saturated steam at 20 bars. The structure of the plant is shown in Fig (1). The plant consist of a gas turbine adopted by a recuperator that uses part of the thermal energy of exhaust gases and a heat recovery steam generator for producing steam. It is assumed that gas turbine works in design condition and environmental is in ISO conditions. The fuel is natural gas with a lower heating value (LHV) equal to 50000 kJ/kg.

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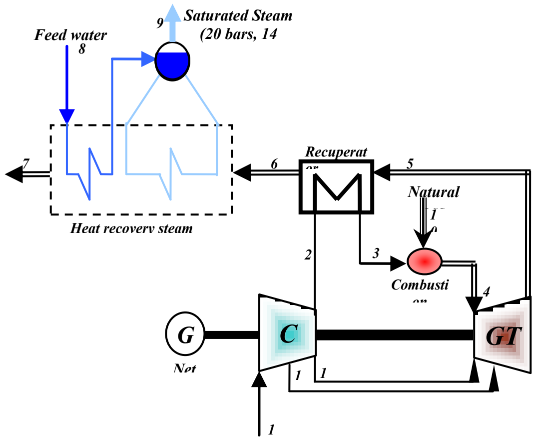

Fig.1: Flow diagram of the CGAM problem

Tsatsaronis and Pisa [1] proposed iterative exergoeconomic optimization procedure based on exergoeconomic variables. Exergy costing is applied for thermoeconomic analysis of the cycle resulting in cost evaluation of all streams. It seems that iterative exergoeconomic optimization procedure, as described in [1], [2] is based on linguistic descriptions using expert opinion. According to this characteristic of iterative exergoeconomic optimization, Tsatsaronis applied fuzzy inference system to employ his method [3]. By means of fuzzy logic, the knowledge of expert is translated in quantified mapping. This quantified knowledge base cause to move toward optimum condition, while initial condition of system does not affect the result. Frangopoulos [4] applied the thermoeconomic functional approach to CGAM problem optimization, and obtained the same results as Tsatsaronis [1]. The relevant sensitivity study showed that turbine isentropic efficiency has the highest effect on the performance in thermoeconomic-optimized condition.

In this paper, a new approach to the modeling and optimization of CGAM cycle is presented. The cycle has been modeled considering blade cooling and we used GA for optimization, which more suitable for such a problem. While it is too difficult to derive explicit functions describing the fuel mass flow and the constraints versus decision variables and thus conventional optimization techniques are not suitable. GA enables us to optimize the structure is simulated without independent of direct functions meaning simultaneous simulation and optimization.

2. THERMODYNAMIC MODEL

A standard gas turbine cycle is considered for the present analysis. The cycle consists of a compressor, a combustion chamber and a cooled turbine.

2.1. Compressor

Using first law of thermodynamic and knowing exit temperature of compressor and the location of extraction for blade cooling, we can determine consumed work and isentropic efficiency. Here polytropic efficiency has been used to calculate exit condition. For one kilogram of inlet air to compressor, entropy change can be written as [5]:

$$ds = \bar{c}_{PM}\frac{dT}{T} - R\frac{dP}{P} \tag{1}$$

Using the concept of polytropic efficiency, final exit temperature can be determined as:

$$\bar{c}_{PM}\frac{dT}{T} = \frac{R}{\eta_{\infty c}}\frac{dP}{P} \tag{2}$$

By integrating this equation, exit condition can be known, but compressor work consumption cannot be determined until blade cooling air and its location is determined.

2.2. Combustion Chamber

Inlet fuel (natural gas) is composed of CH4, C2H6, C3H8 and C4H10. Exit temperature of combustion chamber is an input to the model and then fuel consumption is calculated. To consider heat flux between combustion chamber and the environment and incomplete combustion, combustion chamber efficiency is introduced as the ratio of theoretical fuel consumption (complete and adiabatic combustion) to actual fuel consumption. Using thermodynamic laws, combustion equation can be written as:

$$\begin{aligned}
&\bar{\lambda}(aC_{x1}H_{y1} + bC_{x2}H_{y2} + cC_{x3}H_{y3} + dC_{x4}H_{y4}) + \\
&(y_{O_2}.O_2 + y_{N_2}.N_2 + y_{Ar}.Ar + y_{CO_2}.CO_2 + y_{H_2O}.H_2O) \\
&\rightarrow (y_{CO_2} + \bar{\lambda}.n_{\alpha t}).CO_2 + (y_{H_2O} + .5n_{\beta t}.\bar{\lambda}).H_2O \\
&+ y_{N_2}.N_2 + y_{Ar}.Ar + (y_{O_2} - B.\bar{\lambda}).O_2
\end{aligned} \tag{3}$$

$$\bar{\lambda} = \frac{n_{fuel}}{n_{air}} \tag{4}$$

Where $\bar{\lambda}$ is:
a, b, c and d are molar ratios of inlet fuel to combustion chamber, y_i is the molar ratio of inlet air, $n_{\alpha t}$ and $n_{\beta t}$ are sum of the carbon and hydrogen moles in the fuel. B is $n_{\beta}(n_{\alpha t}+0.25)$, and finally, $\bar{\lambda}$ can be written as:

$$\begin{aligned}\bar{\lambda} = &\left[\begin{array}{l} y_{N_2}.\Delta\bar{h}_{N_2} + y_{O_2}.\Delta\bar{h}_{O_2} + y_{Ar}.\Delta\bar{h}_{Ar} \\ + y_{CO_2}.\Delta\bar{h}_{CO_2} + y_{H_2O}.\Delta\bar{h}_{H_2O} \end{array}\right]_{T_{exit\_con}}^{Tflame} \\ &/[LHV + \sum_{i=1}^{4} a_i.\Delta\bar{h}_{C_{xi}H_{yi}} \uparrow_{298.15}^{T_{fuel\_in}} - n_{\alpha t}.\Delta h_{CO_2} \uparrow_{298.}^{T_{fla}} \\ &- .5n_{\beta t}.\Delta\bar{h}_{H_2O} \uparrow_{298.15}^{T_{flame}} + B.\Delta\bar{h}_{O_2} \uparrow_{298.15}^{T_{flame}}]\end{aligned} \tag{5}$$

Then fuel air ratio based on mass flow can be written as:

$$\begin{aligned} f_{theoretical} &= \frac{m_{fuel}}{m_{air}} = \frac{n_{fuel}}{n_{air}}.\frac{MW_{fuel}}{MW_{air}} \\ &= \bar{\lambda}.\frac{MW_{fuel}}{MW_{air}} \end{aligned} \tag{6}$$

Actual fuel air ratio will be determined with combustion chamber efficiency as:

$$\left[\eta_{C.C.} = \frac{f_{theoretical}}{f_{actual}}\right] \tag{7}$$

A correct value for state-of-art of gas turbines is between 0.99 and 0.999. Molar ratio of combustion products and mass flow entering turbine are determined at the exit of combustion chamber.

2.3. Turbine Blade Cooling

Cooled turbine model is based on El-Masri's work [6, 7], which has been modified and re-used by Bolland [8]. In this model, blade temperature is an input (usually 1123K) and expansion path is considered to be continuous, instead of actual expansion (stage-by-stage expansion) [9]. Expansion path is divided into large number of sub-stages with low-pressure ratio. This model has been applied where parametric analysis of gas turbine is our goal and the knowledge of expansion path is not important. However, such a model cannot deliver information about expansion path. For every sub-stage, mass of coolant can be determined as:

$$\frac{m_c}{m_g} = \frac{C_{P.g}}{C_{P.c}}.\sigma.\frac{dT_g}{T_{amb.}}.\frac{(T_g - T_b)}{(T_b - T_c)} \tag{8}$$

Where σ is defined as:

$$\sigma = \frac{St.\dfrac{A_{w,stage}}{A_g}}{C.(k-1).M_a^{\,2}.\varepsilon} \tag{9}$$

In the above equation, mc is the mass flow of coolant for sub-stage; mg is the mass flow of gas entering sub-stage; C_{pg} and C_{pc} are the specific heat of gas and coolant respectively, dT_g is the difference between inlet and outlet temperature; T_b is the temperature of blade; T_c is the temperature of coolant entering blade. Parameter St is the Stanton number and is equal to 0.005 with good accuracy [10, 11]. Ratio of ($A_{w,stage}/A_g$) for every stage is 10 and C ranges between 0.3 and 0.5. The constant C depends on stage geometry and velocity triangles. Nondimensional parameter σ is a critical parameter in gas turbine design with ε having large influence on σ., its value changes depending on cooling technology. For convective cooling, its value is equal to 0.3 and for film cooling, equal to 0.5. For high-technology gas turbine, σ is equal to 0.1-0.15, while for others it is 0.4-0.45 [12]. Other turbine models such as stage by stage models have also been introduced to our model, but such a detailed model is not suitable for this case [12]. Validation of whole gas turbine model is explained in [13].

2.4. Modeling of Heat Recovery Steam Generator

In this study, a single pressure HRSG has been modeled. ***IAPWS*** (International Association for Properties of Water and Steam) standard for properties of water and steam has been used [14]. The model inputs are: the pressure and the mass flow of steam, approach temperature and pressure drop in different heat exchangers. Validation of this model is described in [13].

3. ECONOMIC MODEL

It is necessary for thermoeconomic optimization to state the cost of components as a function of the decision variables. We use updated cost functions for the components of the gas turbine simple cycle. The advantages of these functions compared to the previous work [1], [2], [3] are consideration of blade cooling and the use of polytropic efficiencies instead of isentropic efficiencies in cost functions evaluation. The purchased costs of the gas turbine components are calculated as follows, while the costs of HRSG and recuperator are given in [1], [16] respectively.

$$C_{compreaaor}=c1\left[\left(\frac{m_{air}\sqrt{R_1T_1}}{P_1.m_{cr}}\right)\Big/\left(\frac{m_{air}\sqrt{R_1T_1}}{P_1.m_{cr}}\right)_{ref}\right]^{c3}.\frac{cpr^{c4}\ln(cpr)}{\left(1-\eta_{\infty,c}\right)^{c2}} \quad (10)$$

$$C_{combustor}=cc1\left[\frac{\left(m_4.V_4\right)}{\left(m_4.V_4\right)_{ref}}\right]^{cc5} \quad (11)$$
$$\times\frac{1}{dP_p{}^{cc2}}\times\left[1+\exp\left(cc3\left(\frac{T_4}{T_4ref}\right)-cc4\right)\right]$$

$$C_{turbine}=t1\left[\frac{\left(m_5.V_5\right)}{\left(m_5.V_5\right)_{ref}}\right]^{t5}\times\frac{\ln(cpr)}{\left(1-\eta_{\infty,t}\right)^{t2}} \quad (12)$$
$$\times\left[1+\exp\left(t3\left(T_4/T_5ref\right)-t_4\right)\right]$$

The coefficients used in the cost functions are reported in table (1). Furthermore, the cost rate associated with fuel is obtained from $C_f=c_f m_f LHV$ Where the fuel cost per energy unit (on an LHV basis) is c_f=0.004$/MJ. So, the objective functions for thermodynamic and Thermoeconomic optimizations are m_f and $c_f m_f LHV+\sum Z_i$ respectively, where Z_i is the purchased cost of the equipment.

4. OPTIMIZATION PROCEDURE

In design and optimization of thermal systems, it is convenient to identify two types of independent variables, Decision variables and parameters. Parameters are independent variables whose values are specified. They are kept fixed in optimization process. Here the following parameters are defined:

- System Products

The net power generated by the system is 30 MW. Saturated steam is supplied by the system at P_9=20 bars and m_9=14 kg/sec.

- Compressor

Inlet air condition is ISO condition and air molar analysis is 0.7748 N2, 0.2059 O2, 0.0003 CO2, and 0.019 H2O (g).

- Recuperator

Pressure drops 3% on the gas side and 5% on the airside.

- Heat Recovery Steam Generator

T8=298.15 K, P8=20 bars, P7=1.013 bars. Pressure drop: 3% on the gas side and 3% on the economizer and Approach temperature is 15 K.

- Combustion Chamber

T10=298.15 K. pressure drop 4% and combustion chamber efficiency is 99.5%.

- Gas Turbine

Blades maximum temperature assumed 1123 K.

Table (1) - Coefficients of the gas turbine cost Function

c1[$]	6420.8
c2	0.15
c3	0.85
c4	0.3
cc1[$]	2340
cc2	0.995
cc3	5.479
cc4	34.36
cc5	0.6
P_{ref}	1.01325
T_{ref} [K]	288.15
M_{ref} [kg/s]	1
t1[$]	7533.7
t2	0.29
t3	4.185
t4	23.6
t5	0.75
R_{ref}[J/kg.K]	289.2
mcr_{ref}	0.9586

While parameters remain fixed, decision variables varied in optimization process. In this model the compressor pressure ration CPR,

compressor polytropic efficiency $\eta_{\infty,c}$, compressor inlet mass flow m_{air}, turbine inlet temperature T_4, turbine polytropic efficiency $\eta_{\infty,t}$, the turbine blade cooling parameter σ, pinch-point temperature in HRSG ΔT_{PP} and effectiveness coefficient of recuperator ε are considered as decision variables. It is important to note that fuel mass flow and system products including net power and saturated vapor cannot be state explicitly as a function of decision variables. Maximum and minimum values for decision variables have been shown in table 2.
Minimum and maximum values for decision variables has been selected from manufactures data for gas turbines in the net power range of 30 MW [17], [18], [12].

The optimization problems of energy systems are usually nonlinear. There are some techniques like gradient methods, which need explicit objective functions, or complex search methods, which need a clearly defined searching space and require much calculation time. Because of the characteristics of this problem, it was decided to use a genetic algorithm optimization method. Genetic algorithm (GA) is a suitable tool for this problem optimization.

Table (2) - Decision variables changes range

Variable	*Minimum value*	*Maximum value*
CPR	7	16
m_{air}	90	140
$\eta_{\infty,c}$	0.85	0.92
$\eta_{\infty,t}$	0.85	0.91
T_4	1233	1533
σ	0.2	0.5
ε	0.6	0.9
ΔT_{PP}	0	30

Genetic algorithms are a stochastic search method, which motivated by the hypothesized natural process of evolution in biological populations, where genetic information store in chromosomal string evolve over generations to adapt favorably to a static or changing environment. The algorithm is based on elitist reproduction strategy, where members of population, which are deemed most fit, are selected for reproduction, and are given the opportunity to strengthen the chromosomal makeup of progeny generation. This approach is facilitated by defining a fitness function or a measure indicating the goodness of a member of the population in the given generation during the evaluation process.

To represent designing a chromosome-like strings, the design variables are converted to their binary equivalent and thereby mapped into a fixed length string of 0 s and 1 s. A number of such strings constitute a population of designs, with each design having corresponding fitness value. The fitness function includes the objective function and a number of penalization functions, which depend on the constraints of the physical model.

The starting population is selected randomly in the domain lying between the minimum and maximum values of X and then the following genetic operators apply to improve results.

- *Reproduction.* Individuals are selected and the probability of selection is based on their fitness value. The new population pool has higher average fitness value than the previous pool.
- *Crossover.* In the two-point crossover approach, two matting parents are selected at random; the random number generator is invoked to identify the sites on the strings, and the stings of 0 s and 1 s enclosed between the chosen sites are swapped between the mating strings.
- *Mutation*. A few members from the population pool are taken according to probability of mutation P_m, and a 0 to 1 or vice versa are switched at randomly selected mutation site on the chosen string.

The process of reproduction, crossover and mutation constitute one generation of the GA. After several generations, the GA is stopped and the best point among the values taken as the optimal point. Being a probabilistic search method, GA's are very good at finding global optima. Furthermore, GA's need only function values and not gradient information, which makes them easy for real systems where accurate gradient information is difficult to obtain, and local optima may occur. Especially for this problem (CGAM problem), which explicit formulation for cost function and constraints is impossible. In order to define suitable penalty functions for this problem the following considerations has been done:

- For avoiding the acid droplet, the exhaust gas temperature of the HRSG T_{exh} should not be below 120°C.
- Net electric power $\dot{W}_{net}$ generated is 30MW.
- 14 kg/sec saturated steam $\dot{m}_{steam}$, as a utility, at 20 bars should be produced.

With these conditions, the penalty functions are defined as:

$$\mathrm{Pen}(\dot{W}_{net}) = \begin{cases} \mathrm{Abs}(\dot{W}_{net} - 30000) & \text{if } \mathrm{Abs}(\dot{W}_{net} - 30000) \geq 0.3 \\ 0 & \text{other wise} \end{cases} \quad (13)$$

$$\mathrm{Pen}(T_{exh}) = \begin{cases} \mathrm{Abs}(T_{exh} - 120) & \text{if } T_{exh} \leq 120 \\ 0 & \text{if } T_{exh} > 120 \end{cases} \quad (14)$$

$$\mathrm{Pen}(\dot{M}_{steam}) = \begin{cases} \mathrm{Abs}(\dot{M}_{steam} - 14) & \text{if } \mathrm{Abs}(\dot{M}_{steam} - 14) \geq 0.00014 \\ 0 & \text{other wise} \end{cases} \quad (15)$$

Then the fitness functions for Thermodynamic and Thermoeconomic optimizations are defined as:

$$F_{fitness}(X) = \dot{m}_f + P_1.\mathrm{Pen}(\dot{W}_{net}) + P_2.\mathrm{Pen}(\dot{M}_{stem}) + P_3.\mathrm{Pen}(T_{exh}) \quad (16)$$

$$F_{fitness}(X) = c_f \dot{m}_f LHV + \sum_{i=1}^{5} \dot{Z}_i + P_1.Pen(\dot{W}_{net}) + P_2.Pen(\dot{M}_{steam}) + P_3.Pen(T_{exh}) \quad (17)$$

Where X is the decision variables vector which are encoded to binary equivalent quantities in given ranges and P_i is coefficient to adjust the weight of each penalty/

5. RESULTS AND DISCUSSIONS

Based on methodology described in the later sections, the CGAM problem was solved to minimize mass flow of fuel for thermodynamic optimization and total cost for thermoeconomic optimization. Therefore the results will be discussed in two separate parts for thermodynamic and thermoeconomic optimization. Then sensitivity analysis of optimum performance with 10% change in decision variables is done to determine the effective variables. Then the thermoeconomic optimization is used to determine the best design condition for a new product of solar turbine (Mercury 50), as a new cogeneration cycle with a gas turbine that has been equipped with recuperator.

5.1 Thermodynamic Optimization

Table (3) shows the decision variables values in optimum condition. In thermodynamic optimization, the object is to minimize fuel mass flow rate, so inlet air mass flow to compressor must decrease to minimize f actual. In HRSG section, null pinch-point temperature difference is obtained to minimize exergy destruction. In addition, the stack temperature must reach its minimum value (table (4)) to minimize exergy loss so it needs the inlet temperature of gas entering HRSG increase. In the T-Q diagram shown in fig (2), it has been shown that with low mass flow and high inlet temperature of gas it is possible to reach minimum stack temperature. To increase inlet gas temperature to HRSG, turbine inlet temperature or CPR must increase, and to minimize fuel consumption, recuperator effectiveness must increase. Increase in CPR lowers heat recovery of recuperator from gas turbine exhaust and increases fuel consumption of gas turbine. Increase in TIT also increase fuel mass flow, but due to effect of increase in coolant mass flow with increasing TIT (which increase inlet air mass flow) increase in CPR is a better option here. Also increasing TIT has more effect on fuel mass flow growth relative to increase in CPR.

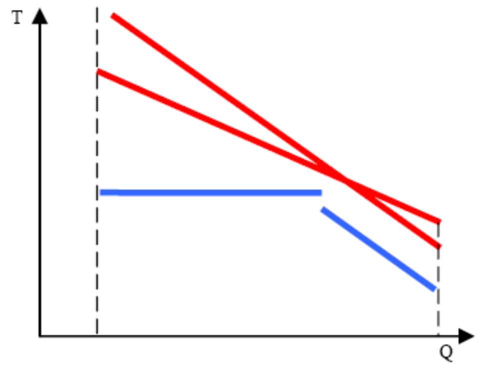

Fig .2: T-Q diagram for HRSG

In high TIT, if we use high values for σ (low blade cooling technology), mass flow of coolant will increase and for a specific inlet air mass flow, it decreases power production. As mentioned above, in thermodynamic optimization, keeping inlet air mass flow as low as possible cause to less fuel consumption, therefore low values of σ (high blade cooling technology) must be considered to reach the best thermodynamic conditions (low fuel flow rate).

Table (3) - Results for thermodynamic optimization

Variable	*Value*	*Variable*	*Value*
CPR	14.45	ε	0.6967
m_{air}	106.69	T_4	1372.9
$\eta_{\infty,c}$	0.871333	σ	0.2
$\eta_{\infty,t}$	0.91	ΔT_{PP}	0

The results of gas turbine cycle optimization have similarity to real cycle conditions except for blade cooling parameter. As described in [12] this value of σ is suitable for large heavy-duty industrial gas turbines where turbine inlet temperature reaches 1600 K and higher.

Polytropic efficiency of compressor and turbine has large influence on gas turbine cycle efficiency and power. A detail analysis of their influence will be done after thermoeconomic results discussion.

Table (4) shows thermodynamic conditions in various points of CGAM cycle that has been calculated for best thermodynamic design performance viewpoint.

Table (4) - Values of temperatures for the streams and coolant mass ratio in the thermodynamic optimal condition

T_2	T_3	T_5	T_6	M_C / M_{air}
683.49	749.07	773.33	717.17	0.04756

In addition, the results for power production, steam and fuel mass and stack temperature are shown in table (5).

Table (5) - Values of dependent thermodynamic variables for thermodynamic optimum design

m_{fuel}	W_{net}	m_{steam}	T_{exh}
1.59822	29999.7478	13.99996197	393.4

5.2. Thermoeconomic Optimization

As mentioned before, thermodynamic optimization does not show real solution for system and we must consider thermoeconomic approach to minimize total cost of the plant.

Table (6) shows the decision variables values in optimum condition for thermoeconomic optimization case. Most of the results have great difference with thermodynamic optimum condition, especially m_{air}, T_4, σ and ΔT_{PP}. Here pinch point temperature difference is greater than zero to decrease area and cost of HRSG. Inlet air mass flow increases relative to thermodynamic case because of no requirement to minimize fuel flow. Also from economic viewpoint, it is better to increase power with increasing mass flow rather than increasing turbine inlet temperature to high values (Because of exponential relation of TIT and cost of turbine and CC), (See sensitivity analysis part). These two effects cause stack temperature to be greater than its lower bound. Decreasing slope of gas side cooling line in T-Q diagram, inlet air temperature entering HRSG will decrease (while decreasing inlet temperature, mass flow will increase and total amount of energy needs for steam production remains constant). These lead to decrease turbine inlet temperature and increase in CPR relative to the thermodynamic case, therefore effect of recuperator on gas turbine cycle performance and cost will be lowered. Actually, recuperator effectiveness reaches its minimum bound to minimize cost, while fuel consumption increases. Another interesting feature is the upper bound value for cooling parameter σ. Lower turbine inlet temperature, less need for blade cooling and less influence of cycle performance with variations of σ. In addition, inlet air mass flow is high and a high decreasing in value (low blade cooling technology) of σ cause to small power reduction and small decrease in turbine exhaust temperature. Furthermore, this high mass flow is vital for HRSG section.

Table (6) - Results for thermoeconomic optimization

Variable	*Value*	*Variable*	*Value*
CPR	12.5635	ε	0.60967
m_{air}	120.31	*T4*	1365.538
$\eta_{\infty,c}$	0.8987	σ	0.5
$\eta_{\infty,t}$	0.8653	ΔT_{pp}	21.1

Table (7) shows thermodynamic conditions in various points of CGAM cycle that has been calculated for best thermoeconomic design performance viewpoint. In addition, the results for power production, steam and fuel mass and stack temperature are shown in table (8).

Table (7) - Values of temperatures for the streams and coolant mass ratio in the thermoeconomic optimal condition

T_2	T_3	T_5	T_6	M_C / M_{air}
639.02	743.05	808.43	711.79	0.08424

Table (8) - Values of dependent thermodynamic variables for thermoeconomic optimum design

m_{fuel}	W_{net}	m_{steam}	T_{exh}
1.7259668	30000.26	13.999964	425.049

Fig - (3) Shows distribution of components and fuel costs for thermoeconomic-optimized condition. As shown, fuel cost in one year is more than cost of total cost of components.

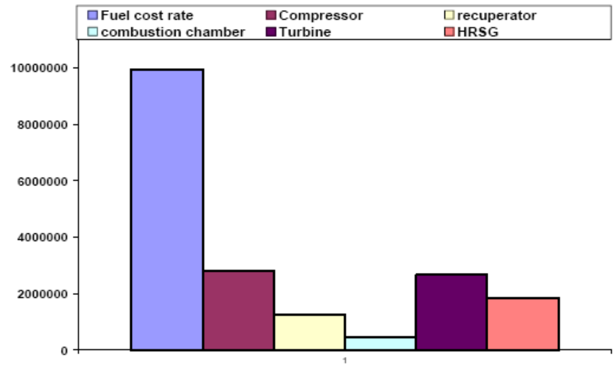

Fig .3: Fuel cost and cost of CGAM problem components

5.3. Sensitivity Analysis

In order to investigate the effective variables that have large influence on optimum performance, 10 % change around the optimum point values in decision variables is done. For thermodynamic optimization case, effect of variation in fuel mass flow, power production, steam mass flow and fitness function with variation of decision variables is shown respectively. Fig (4) shows that inlet air mass flow, turbine inlet temperature and recuperator effectiveness has the highest effect on fuel flow consumption, while turbomachinery efficiency, CPR and blade cooling technology have lower influence on fuel mass flow. Increase in m_{air} and TIT increase m_{fuel} (maximum 10%) and increase in ε reduce that (maximum 4%). It must be emphasized that increase in turbo machinery efficiency lowers heat recovery in recuperator.

Fig (5) shows that compressor and turbine polytropic efficiency changes power production significantly (maximum 24%). Increases in efficiencies reduce compressor work and increase turbine work, so the net power will increase. m_{air} and TIT have less influence and blade cooling technology causes a little changes power. Increase in turbine inlet temperature changes net power less than increase in m_{air}, while its decrease changes it more than m_{air}. It is due to the fact that with increasing TIT, more air for blade cooling will be extracted and less air will enter combustion chamber, the result is decrease in power production. This feature will be inversed for lowering TIT. Other variables do not affect net power significantly. These trends continue in fig (6) while most variables affect steam mass flow production except blade cooling technology less and more. Again, compressor polytropic efficiency has the highest effect (because of change in heat recovery of recuperator, maximum 17%) and σ and CPR have the lowest effect respectively.

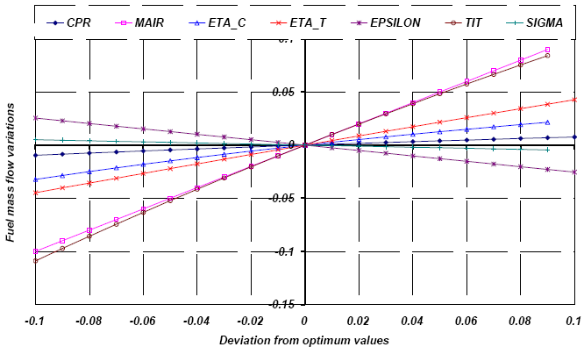

Fig .4: Effect of change in decision variables on fuel mass flow around the thermodynamic optimum condition

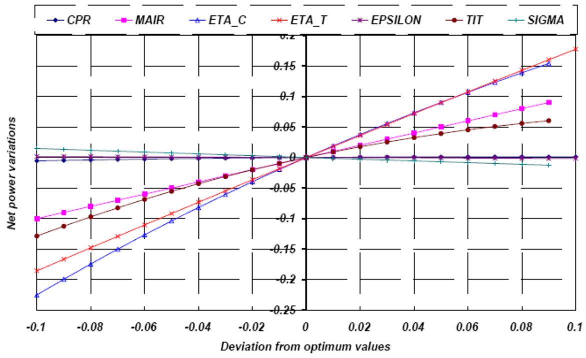

Fig .5: Effect of change in decision variables on net power around the thermodynamic optimum condition

Before we introduce results for fitness function, we can conclude that some variables as compressor polytropic efficiency always has large influence on cycle parameters and outputs, while σ and CPR changes them little. It seems that this trend will continue for fitness function.

Fitness function consists of both inlets and outlets of the plant, so its variation according to decision variables helps to understand general behavior of the plant. Fig (7) approves our prediction and as can be seen, the most important decision variable is compressor polytropic efficiency. As described by Horlock [19], "polytropic efficiency exerts a major influence on the optimum operating point of cooled gas turbines: for moderate turbomachinery efficiency the search for enhanced outlet temperature is known to be logical, but for high turbomachinery efficiency this is not necessary so". As shown, reducing polytropic efficiency changes optimum condition more than increase its value. According to this act, monitoring the performance of the compressor and application of fault diagnostic methods to predict occurred fault is very important.

Turbine polytropic efficiency is the second effective variable on fitness function. Because of blade cooling mass flow influence, TIT and m_{air} cause to similar trend and quantities when TIT increases, while for decrease in TIT, its influence is higher than m_{air} and its quantities will be close to the polytropic efficiency of gas turbine.

Other variables like ε, σ and CPR have similar effect and its reasons have been described in later paragraphs.

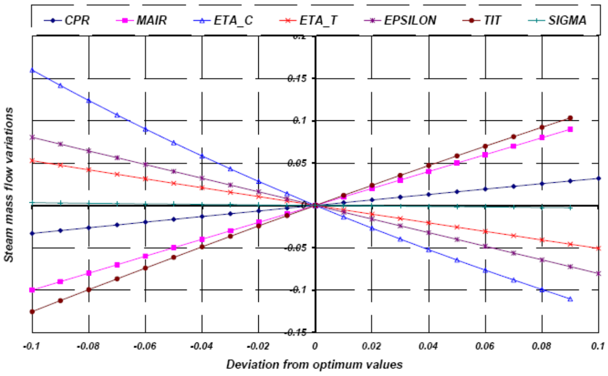

Fig .6: Effect of change in decision variables on steam mass flow around the thermodynamic optimum condition

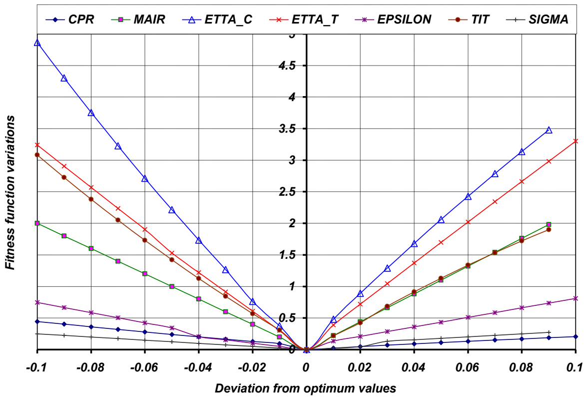

Fig.7: Effect of change in decision variables on fitness function around the thermodynamic optimum condition

Sensitivity analysis around thermoeconomic optimum condition for fuel cost rate, power and steam mass flow yields similar trends, therefore their graphs will not show here. Actually, it can be concluded that change in these set of decision variables yields to similar variations for foregoing depended variables (fuel and steam mass flow and power production).

Only thermoeconomic fitness function will be described here. Again, here, turbomachinaries efficiencies have highest influence, but TIT has less influence relative to m_{air} when its value increases. It is due to this fact that for optimum thermoeconomic condition, TIT has lower value relative to optimized thermodynamic case, so increase in TIT, will cause to less coolant mass flow increment, less power reduction and a little fuel flow reduction. Also influence of ε is lower due to its lower value and higher cost.

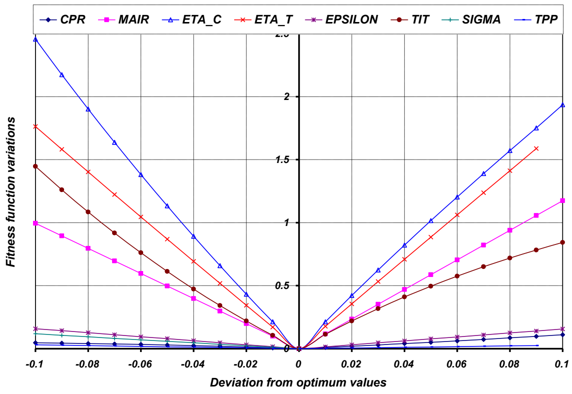

Fig .8: Effect of change in decision variables on fitness function around the thermoeconomic optimum condition

5.4. New Product (Mercury 50)

At the end of this section, the researchers analyze a new product of solar turbine company, introduced as MERCURY 50. This new gas turbine equipped with a recuperator is compatible for CHP applications. Thermodynamic conditions and amount of productions for the cycle according to manufactures data is shown in table (9) [20].

Table (9)-characteristics of mercury50 cycle

W_{net}	m_{steam}	T_{exh}	CPR	m_{air}	P_{STEAM}	ΔT_{PP}
4600	1.75	408	9.9	17.9	9.6	17

Thermoeconomic optimization is done for this case. Results for cycle conditions and dependent decision variables values are shown in table (10) and cost of the plant components are shown in table (11).

Results show that there is a little difference in decision variables between manufactures data and thermoeconomic optimum case. This may be due to the following reasons:

- Estimations in cost of the components (15%)
- Estimations in thermodynamic models especially in blade cooling models
- Pollutant emissions control and cycle modifications relative for the case are not considered.

However, generally the results are quite close to the manufacture data. Thermodynamic conditions of the plant relative to the thermoeconomic-optimized case are shown in table (12)

Table (10) - Values of dependent thermodynamic variables for thermoeconomic optimum design

Variable	*value*
m_{fuel}	0.23789
W_{net}	4599.8
m_{steam}	1.751
T_{exh}	421
T_2	583.451
T_3	765.79
T_5	832.834
T_6	661.528
M_C / M_{air}	0.0530117

Table (11) – Cost values of mercury 50 components in thermoeconomic optimized condition

Compressor	463483.4($)
Recuperator	532751.93($)
Combustion chamber	170088.344($)
Turbine	615690.562($)
HRSG	281386.795($)

Table (12) - Results for thermoeconomic optimization of mercury50

Variable	*Value*	*Variable*	*Value*
CPR	9.7143	ε	0.7258
m_{air}	17.276	$T4$	1344.546
$\eta_{\infty,c}$	0.9153	σ	0.3742
$\eta_{\infty,t}$	0.898	ΔT_{pp}	15.37

6. CONCLUSIONS

In this paper, a new approach for thermodynamic and thermoeconomic optimization of the CGAM problem is presented. Modifications in both of the thermodynamic model of the cycle and method of optimization are done. Results show that:

- Minimizing fuel mass flow, which minimizes exergy losses and destructions, is target of thermodynamic optimization. Therefore, CPR has high value; TIT, blade cooling technology and inlet air mass flow have low value. ΔT_{PP} is zero and ε has a moderate value.
- Thermoeconomic optimization increase inlet air mass flow, lower TIT, σ reach maximum value and ε reach minimum value.
- For thermodynamic optimum condition, blade cooling technology reaches its minimum value, while in thermoeconomic case; σ has the highest possible value.
- Turbomachinery efficiency (especially compressor polytropic efficiency) is the most important variable that affects design condition of the whole plant. While this trend is not quite similar for fuel and steam mass flow and power production, fitness function that describes behavior of the whole plant, is very sensitive to these efficiencies. TIT and inlet air mass flow have lower influence, while TIT's influence decreases with increasing its value.
- Thermoeconomic optimum sensitivity analysis yields similar results, but influence

of TIT will again lower because of reduction of its value. In addition, recuperator affects fitness function less than thermodynamic case due to decrease in its value and high cost of that component.

- Application of this procedure for the new product, MERCURY 50, yields reasonable results. Therefore, the results can be acceptable in practical cases.

7. NOMENCLATURE

A, Ag, Aw	Area, gas path, wall path (m^2)
f	Fuel air ratio (mass basis)
ISO	ISO condition (15°C, 1.01325 bar, 60% relative humidity)
T	Temperature (K)
V	Specific volume (m^3/kg)
c_f	Fuel cost per energy unit ($/Mj)
c_P	Specific heat at constant pressure (kj/kg.K)
DP	Pressure Drop (bar)
h	Specific Enthalpy (kj/kg)
m	Mass flow rate (kg/s)
MW	Molecular weight (kg/kmol)
M_a	Mach number
P	Pressure (bar)
Q	Heat (kj)
R	Gas constant (kj/kmol.K)
s	Specific Entropy (kj/kg.K)
St	Stanton Number
CPR	Pressure ratio
ΔT	Temperature Difference (K)

Abbreviations

GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
ECO	Economizer
GA	Genetic Algorithm
LHV	Lower Heating Value (kJ/kg)
CGAM	**C**. Frangopoulos, **G**. Tsatsaronis, **A**. Valero, **M**.Spakovsky

Greek symbols

η_∞	Polytropic Efficiency (c: compressor, t: turbine)
η_{is}	Isentropic Efficiency
η_{cc}	Combustion Chamber efficiency
$\bar{\lambda}$	Fuel to air mole fraction
ε	Heat exchanger effectiveness
σ	Non dimensional parameter (blade cooling)

Subscripts

Exh	Exhaust
G	Gas
PP	Pinch Point
M	Mixture
Ref	Reference
A	Air
AP	Approach Point
Act	Actual
B	Blade
C	Coolant
C.C.	Combustion Chambers
C.P.	Combustion Products

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