Optimizing the efficiency of an externally fired gas turbine

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Abstract:

In this work we study the optimal parameter configuration (from a thermodynamic point of view, without economical considerations) for different operating and design variables of a thermodynamic model for a realistic Brayton cycle with an externally fired gas turbine, with the aim of maximizing the efficiency. As the analytical form of the optimization function is unknown, the functional values are calculated by running a numerical model varying the parameter values to explore the search space. Specifically, we use a thermodynamic numerical model previously developed by our group that includes an arbitrary number of turbines and compressors.

In the empirical evaluation we consider five parameters as inputs of the system: the fuel-air equivalence ratio, the compression and expansion pressure ratios, and the number of compression and expansion stages. In addition, we also study which are the parameters that have a greater incidence in the outputs of the system. More in detail, the experimental results show strong improvements in the efficiency of up to 52% for different fuels when compared to reference values.

Keywords:

Externally fired gas turbines, multi-stage turbine, optimization, simulated annealing.

1. Introduction

One of the highlights at international level related to power generation issues is the dependence on fossil fuels. Around the world efforts are being made to change the energy matrix and to turn towards renewable energy. Unlike solar and wind energy, a renewable energy source that stands out for its versatility is biomass [***1***]. The use of biomass as an energy source for electricity production is not new, but for universal use it is necessary to expand its range of use and improve energy conversion systems.

There are many systems that can use biomass as a fuel to generate electrical energy, most are based on the Rankine cycle that uses fuel to generate superheated steam which is expanded in a steam turbine [***2***], the Stirling engine uses an external heat as energy input that can be from any source including biomass combustion. On the other hand, a well know technology of gas turbines commonly uses clean fuels that could be adapted to use dirty fuel by incorporating an externally fired combustion chamber and a hot temperature heat exchanger [***3,4***]. The performance of this kind of systems do not vary the net power but the efficiency decreases due to the loses that are generated in the combustion chamber and the heat exchanger.

In the present work, a thermodynamic model is used for the considered cycle. The model starts from the principles of Classical Reversible Thermodynamics and then several losses sources are considered in order to reproduce the behavior of real systems. In this way a computer program is developed, capable of representing the behavior of the system performance and, then it is performed

an optimization analysis of the configuration parameters. This approach is known as black-box optimization since the optimization function is unknown and it only can be viewed in terms of its inputs and outputs. Usually, one or two parameters are considered for optimization once the reminders are considered as fixed.

A review of the literature shows papers that present results of optimization of multistage gas turbines [***5***] and works that directly present the optimization of externally fired gas turbines [***6***]. The present work attempt to merge both, optimize a multi stage externally fired gas turbine (EFGT).

The rest of the paper is structured as follows. In Section 2 we summarize the numerical model to simulate the behavior of an EFGT. The parameter configuration as an optimization problem is described in Section 3. Then, Section 4 describes the optimization algorithm used in this work and followed by the experimental evaluation in Section 5. Finally, a few concluding remarks and future work are offered in Section 6.

2. Theoretical Model

The model studied consists of an arbitrary number of N_t turbines and N_c compressors, with the corresponding $(N_c - 1)$ intercoolers and $(N_t - 1)$ intermediate burners (see Fig. 1). The ideal cycle (see Fig. 2) can be summarized as follows: the air, as working fluid, at pressure P_1, and temperature T_1, is compressed by N_c adiabatic compressors, to pressure P_2, and temperature T_2; then increases its temperature up to T_3 in an isobaric ceramic heat exchanger (HTHE). This air is expanded, by N_t adiabatic turbines, to a pressure and temperature P_4 and T_4, respectively. Next, the air is used to supply the combustion chamber to burn a solid fuel. The exhaust gases are used as the hot fluid in the HTHE at temperature T_{ad}, then are released to the ambient at T_{ch}. Between each compression process, heat is extracted by the intercooler in order to decrease the temperature, at each compressor inlet, to T_1. In the same manner, between each expansion process, heat is supplied by the intermediate burners in order to increase the temperature, at each turbine inlet, to T_3. The burned gases are released to the ambient at temperature T_{e2}.

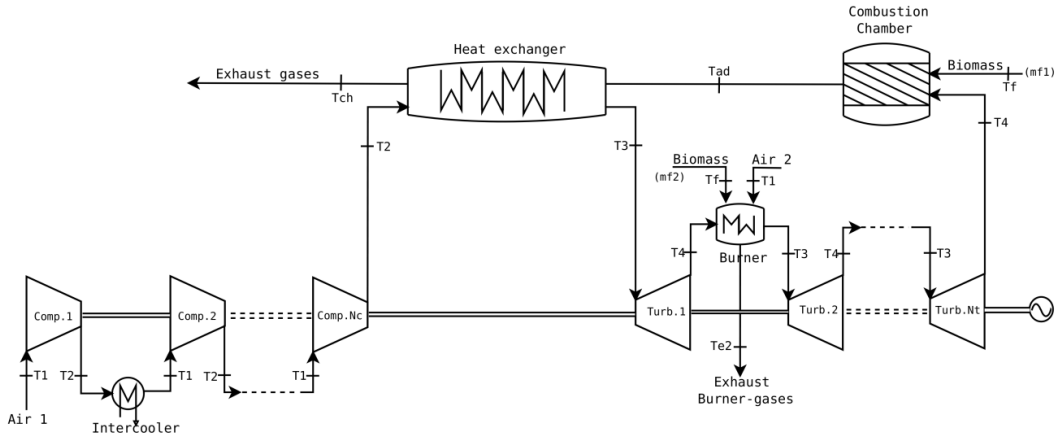

Fig. 1. Scheme of a multi-stage externally fired gas turbine, with burners between expansion processes and intercoolers between compression processes.

All compressors are considered to have the same pressure ratio r_c as well as the turbines, with a pressure ratio r_t. The turbines inlet temperature, T_3, is fixed according to a constructive and metallurgical limit. In the main combustion chamber the biomass is burned with clean air out coming from the last turbine at temperature T_4. The equivalence ratio of this combustion, ϕ_1, is calculated so that the adiabatic flame temperature T_{ad} obtained allows to reach T_3 at the exit of the ceramic high temperature heat exchanger (HTHE) [***7***,***8***]. The combustion in the burners, which allows to heat the clean air from T_3 to T_4 in a separate circuit, uses air from the ambient at T_1. The equivalence ratio of this combustion, ϕ_2, is calculated so that the adiabatic flame temperature

obtained matches with the one reached in the main combustion chamber. This procedure ensures that the heat exchange is physically possible and allows setting a pinch point of 100K, on the exhaust gases temperature T_{e2} as a construction criteria.
The present work attempts to reproduce realistic performance records for an EFGT starting from the ideal model and introducing the main irreversibilities existing in real plants as pressure drops, non-ideal heat exchangers, and isentropic efficiencies in compressors and turbines.

2.1. Thermodynamic model

The thermodynamic model of a multi-step Brayton cycle is based on those presented by Sánchez-Orgaz et al. [***9***,***10***] and Roco et al. [***11***]. The present work adapts the numerical model of [***12***] for the use of an external biomass-fuelled combustion chamber, taking into account the chemical reactions in the combustion of solid fuels. Fig. 2 represents the thermodynamic cycle, considering pressure drops and irreversibilities in compression and expansion processes, that are considered as non-adiabatic. Next we detail the main assumptions and definitions for each stage.

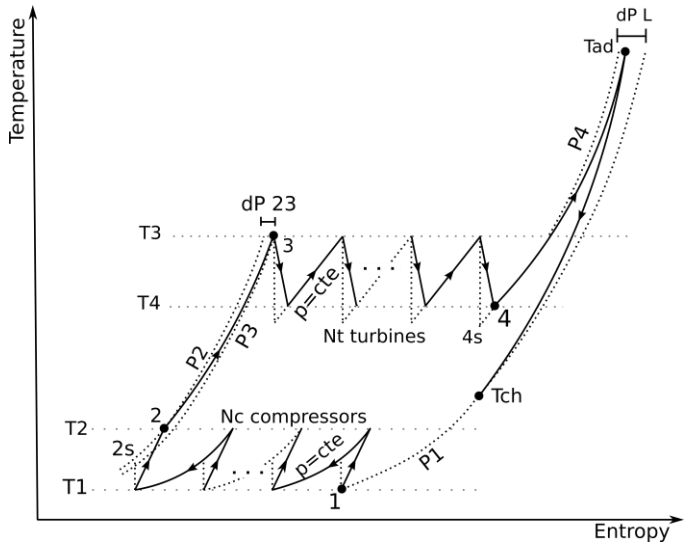

Fig. 2. Temperature-entropy diagram for a multi-stage externally fired Brayton cycle.

2.1.1. Compression and expansion

Defining the pressure ratio of a compressor as, $r_c = ({p_2}^i/{p_1}^i)$, where ${p_1}^i$ and ${p_2}^i$ are the inlet and outlet pressures of the *i*th compressor respectively, considering N_c compressors with the same value of r_c, P_1 as the inlet pressure of the first compressor (assumed in this work as atmospheric pressure) and P_2 the outlet pressure of the last compressor (see Fig. 2), it is easily proven that

$$P_2 = P_1 r_c^{Nc} \qquad (1)$$

Since the ideal compression process is isentropic, and considering $\bar{\gamma}_1$ as the mean isentropic coefficient between points 1 and 2s (see Fig. 2), we define the isentropic compressor pressure ratio,

$$a_c = \frac{T_{2s}}{T_1} = r_c^{\frac{\bar{\gamma}_1 - 1}{\bar{\gamma}_1}} \qquad (2)$$

In order to take into account the compression irreversibilities, the isentropic efficiency is defined as

$$\varepsilon_c = \frac{T_{2s} - T_1}{T_2 - T_1} \qquad (3)$$

With equations (2) and (3) it is possible to obtain [***9***];

$$\frac{T_2}{T_1} = Z_c = 1 + \frac{a_c - 1}{\varepsilon_c} \qquad (4)$$

where Z_c is defined for convenience. The pressure ratio of one turbine, r_t, is calculated using N_t, P_2, and all the pressure losses in the circuit.The pressure drop in the cold side of the heat exchanger is denoted as dP_{23}, and the pressure drop in the hot side of the heat exchanger is dP_L. Considering the intercoolers and re-heaters (intermediate burners) as isobaric, the global pressure ratio for the N_t turbines is calculated as

$$r_{t_G} = \frac{P_3}{P_4} = \frac{P_2 - dP_{23}}{P_1 + dP_L} \quad (5)$$

where P_3 is the pressure at the inlet of the first turbine and P_4 is the pressure at the outlet of the last one. Proceeding in the same way as for (1), the pressure ratio of one turbine is,

$$r_t = \left(r_{t_G}\right)^{1/N_t}. \quad (6)$$

In the same way as for compression, the isentropic turbine pressure ratio is:

$$a_t = \frac{T_3}{T_{4s}} = r_t^{\frac{\bar{\gamma}_2 - 1}{\bar{\gamma}_2}}, \quad (7)$$

The following parameters and equations allow to relate temperatures with the turbine pressure ratio:

$$\varepsilon_t = \frac{T_3 - T_4}{T_3 - T_{4s}}, \quad (8)$$

$$\frac{T_4}{T_3} = Z_t = 1 - \varepsilon_t(1 - a_t^{-1}). \quad (9)$$

where $\bar{\gamma}_2$ is the mean isentropic coefficient of air among the state 3 and 4 (see Fig. 2), and ε_t is the isentropic efficiency of the turbines considered identical.

2.1.2. Heat input

Since the system is externally fired, the process of heat addition to the working fluid is performed by a ceramic heat exchanger between burned gases, coming from a combustion chamber, and the working fluid. To solve all temperatures of heat exchanger we use the NTU method and energy balance on the heat exchanger,

$$\dot{m}_{a,1}[h_a(T_3) - h_a(T_2)] = \dot{m}_{g,1}\left[h_{g,1}(T_{ad}) - h_{g,1}(T_{ch})\right], \quad (10)$$

where, $\dot{m}_{a,1}$ is the air mass flow in the compression process, h_a and $h_{g,1}$, the enthalpy of air and combustion gases, from combustion chamber, respectively, at the temperatures mentioned (see Fig. 2), T_{ad}, the adiabatic flame temperature, T_{ch}, the exhaust temperature, and $\dot{m}_{g,1}$, the mass flow rate of burned gases. The value of $\dot{m}_{g,1}$ is obtained by a mass balance for the combustion chamber:

$$\dot{m}_{g,1} = \dot{m}_{f,1} + \dot{m}_{a,1} \quad (11)$$

where $\dot{m}_{f,1}$is the fuel mass flow. After obtaining T_{ad}, it is easy to determine the equivalence ratio to reach this temperature, solving the combustion process and burned gases composition (see Secs. 2.3. and 2.4.).

Other heat inputs come from the intermediate burners. Again, using the NTU method and energy balance and the hypothesis of pinch point (explained at the beginning of Section 2), it is possibly to determine the exhaust temperature, the secondary flow mass, $\dot{m}_{f,2}$, and equivalence ratio of the burners, ϕ_2.

$$\dot{m}_{a,1}[h_a(T_3) - h_a(T_4)] = \dot{m}_{f,2}.h_f\left(T_f\right) + \dot{m}_{a,2}h_a(T_1) - \dot{m}_{g,2}h_{g,2}(T_{e2}) \quad (12)$$

where the subscript *2* denotes properties at the burners hot side. It is important to notice that the composition of the exhaust burners gases is different from the one calculated in the main combustion chamber.

2.1.3. Power and efficiency

With all temperatures of the cycle and main mass flow air, it is possible to calculate its net power by,

$$P = N_t \dot{m}_{a,1}[\, h_a(T_3) - h_a(T_4)] - N_c \dot{m}_{a,1}[\, h_a(T_2) - h_a(T_1)] \qquad (13)$$

In order to take into account the quality of conversion energy in the system we consider the fuel conversion efficiency, that relates the heating value and usable energy of the thermal engine. It expression for time rates is:

$$\eta = \frac{P}{\dot{m}_f \, LHV} \qquad (14)$$

Where LHV, is the lower heating value of the biomass and $\dot{m}_f$ is the total fuel rate that enters to the system. The mass, $\dot{m}_f$, is simply obtained by adding $\dot{m}_{f,1}$ and $\dot{m}_{f,2}$

$$\dot{m}_f = \dot{m}_{f,1} + \dot{m}_{f,2} \qquad (15)$$

where $\dot{m}_{f,1}$ is obtained with the knowledge of the value of the equivalence ratio ϕ_1 and $\dot{m}_{f,2}$, directly by solving chemical reactions at the intermediate burners.

2.2. Heat Exchangers

To evaluate the performance of the heat exchanger it is necessary to introduce its effectiveness, ε, which varies according to the working conditions, such as mass flow rates, fluids properties and temperatures, using the NTU method [***13***]. The effectiveness of the heat exchanger is defined as (see Fig. 2 for the temperatures),

$$\varepsilon = \frac{T_3 - T_2}{T_{ad} - T_2}, \qquad (16)$$

For any heat exchanger it is shown [***13***] that $\varepsilon = f(NTU, C_r)$, considering the *number of transfer units*, *NTU*, as

$$NTU = \frac{UA}{C_{min}} \qquad (17)$$

where UA is the global exchange coefficient of the heat exchanger, C_{min} the minimum heat capacity rate and $C_r = \frac{C_{min}}{C_{max}}$ heat capacity ratio .

We make use of a counter-flow scheme to model the heat exchanger [***8***,***13***]

$$\varepsilon = \frac{1 - e^{-NTU\,(1 - C_r)}}{1 - C_r e^{-NTU\,(1 - C_r)}} \qquad (18)$$

From the correlations of the *Nusselt* number for internal flow [***13***] and considering the hypothesis that the thermodynamic properties of the working fluids do not change considerably within the range of temperatures taken into account, nor due to its composition, the convection coefficient for air and exhaust gases, are only affected for variations in the mass flow rate of any of the fluids. Therefore, NTU could be expressed by,

$$NTU = \frac{K}{C_{min}\left[(\dot{m}_a)^{-0.8} + (\dot{m}_g)^{-0.6}\right]} \qquad (21)$$

where the coefficient *K* depends on the geometry of the heat exchanger and thermodynamic properties of the working fluids. From the experimental work of Batista de Mello, et al. [***8***], the effectiveness of a ceramic high temperature heat exchanger can reach up to values of 0.84. In the present work the value of *K* is adjusted to reach this efficiency at same conditions.

2.3. Combustion Model

In order to solve the chemistry and the energetic of combustion we assume a certain wet fuel with a determined chemical composition and humidity. The considered chemical reaction is [***14***],

$$C_{\frac{a}{12}}H_{\frac{b}{1}}O_{\frac{c}{16}}N_{\frac{d}{14}} + fH_2O + \frac{\alpha_q}{\phi}(O_2 + 3.76N_2) \longrightarrow \beta CO_2 + \gamma H_2O + \epsilon CO + \theta H_2 + \nu O_2 + \mu N_2 \qquad (R1)$$

where a mole of dry fuel is represented by $C_{\frac{a}{12}}H_{\frac{b}{1}}O_{\frac{c}{16}}N_{\frac{d}{14}}$, with a, b, c, d the amount of each element in mass percentage. The coefficient f represents the moles of water per mole of dry fuel, ϕ the fuel-air equivalence ratio and α_q the stoichiometric amount of O_2 within the air, per mole of dry fuel. It is important to notice that this model does not consider the possible presence of sulfur in the fuel composition. The reason of this is that in spite that most of biomass fuels do have a percentage of sulfur in its compositions, in general it is not considerable. Particularly in this work the biomass fuels considered have less than 0.1% of sulfur. The combustion is solved following the procedure described by Medina et al. [***15***].

3. Parameter configuration as an optimization problem

In this section we formally introduce the optimization problem associated to the multistep externally fired gas turbine.

An optimization problem $\mathcal{P}$ can be defined by a search space $\mathcal{S}$ and an objective function f to be maximized, where $f: \mathcal{S} \to \mathbb{R}$. Each element of $\mathcal{S}$ is a candidate solution of $\mathcal{P}$ and it can be composed by several variables (which are known as decision variables). Solving the optimization problem consists in finding a solution s^* such that $s^* = \arg\max_{s \in \mathcal{S}} f(s)$ and it corresponds to make an optimal decision given a set of constraints. s^* is called a globally optimal of $\mathcal{P}$ and it is not need to be unique in $\mathcal{S}$.

The optimization problems in which $\mathcal{S} \subseteq \mathbb{R}^n$ are known as continuous optimization problems. Continuous optimization problems can be usually solved fast using the Steepest descent or Gradient descent method. However, if the analytical form of the objective function is not known a priori, it is impossible to apply such methods. This family of problems is usually known as black-box optimization problems, since the objective function can be viewed as a black-box, whose internals are unknown and only can be viewed in terms of its inputs and outputs.

Optimization algorithms for black-box optimization problems can only obtain feedback from the search space through the evaluation of candidate solutions. This usually involves running a numerical model or simulation. As a consequence, such algorithms have to be able to choose which candidate solution of the search space needs to be evaluated only based on the information gathered from the solutions that the algorithm has already evaluated.

Table 1. Parameters involved in the optimization problem

Description	*Range*	*Type*
Air flow rate, kg/s	$0.5 \leq \dot{m}_{a,1} \leq 5$	Continuous
Pressure ratio of compression	$2 \leq r_c \leq 50$	Continuous
Numbers of compressors	$1 \leq N_c \leq 5$	Discrete
Numbers of turbines	$1 \leq N_t \leq 5$	Discrete
Inlet turbine temperature, K	$1000 \leq T_3 \leq 1500$	Continuous

In this work, we consider the optimization problem of maximizing the efficiency (η) of an externally fired gas turbine. Thus, the objective function is the efficiency and the value of the objective function corresponding to a candidate solution is obtained by running the thermodynamic model described in Section 2. Table 1 presents the decision variables (i.e. the parameters that can be adjusted) of our optimization problem, including a description of each parameter, the range of possible values and whether the variable is continuous or discrete. The ranges of possible values of each parameter are realistic and they can be found in [***3***,***4***,***5***]. The limitation in the number of

compressors and turbines responds to represent realistic and viable configurations, which results from a combination of technical, operational system and economic aspects. All the parameters are taken as independent, so the only constraints in the problem formulation are the ranges considered for each of them. Fig. 3 shows the interaction between the optimization algorithm and the thermodynamic model.

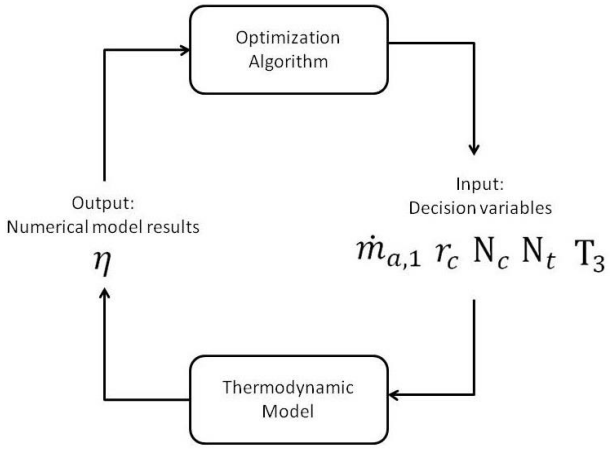

Fig. 3 Scheme interaction between the optimization algorithm and the thermodynamic model.

4. A Simulated Annealing for maximizing the efficiency

In this section, we introduce the optimization algorithm used in this work for optimizing the efficiency of the gas turbine and how it is instantiated for tackling the problem at stake.

4.1. Simulated Annealing

Simulated Annealing (SA) is a global optimization method proposed by Kirkpatrick et al. [***16***] and Cerny [***17***] that is inspired in the process of annealing in metallurgy. The annealing process involves heating and then slowly cooling a material in a controlled manner to obtain a strong crystalline structure. The heat causes an increase in the energy of the atoms of the material and can thus move freely from their initial positions. The slow cooling allows to recrystallize the material in new configurations with lower energy than the original one.

In the SA algorithm, the energy of the system is represented by the cost of the current solution of the optimization problem that is being solved. The goal of the SA algorithm is to find the lowest energy of system, i.e., find the global minimum cost solution of the optimization problem. The algorithm starts from an initial solution that is modified iteratively in order to find a better solution. When the temperature of the system is high (i.e. during the heating stage), the algorithm accepts the degradation of the solution with the goal of escaping from local optima. As the system is cooled, the probability of accepting moves that degrade the solution decreases.

The pseudo-code of SA [***18***,***19***] is presented in Fig. 4. The SA algorithm generates an initial solution and initializes the temperature parameter T. In each iteration, the SA generates a random neighborhood solution (s'). If the solution is better than the current solution, the movement to the new solution is accepted. Otherwise, the new solution is accepted with a probability that depends on the cost of the current solution, the cost of the new solution and the current temperature. This probability follows the Boltzmann distribution $e^{-\frac{\mathrm{f}(s\prime)-\mathrm{f}(\mathrm{s})}{\mathrm{T}}}$ that assures that the probability of accepting this kind of movements decreases as the temperature decreases. Several tries of the generation of the neighborhood solution are made at each level of temperature. The temperature T is gradually decreased according to a cooling schedule. In particular, we adopted the geometric cooling schedule that is one of the most popular cooling shedules. In the geometric schedule, T is updated with αT, where $\alpha \in (0,1)$.

```
Output: Best solution found
s = generateInitialSolution();
T = TMAX;
while not stopping condition do
     repeat
          s' = generate a random neighborhood solution;
          if f(s') < f(s) then
               s = s';
          else accept s' with probability P(T,f(s),f(s'));
          endif
     until stopping condition
     T = Update(T);
endwhile
```

Fig. 4 Pseudocode of the Simulated Annealing Algorithm

4.2. Instantiation of SA to our optimization problem

Since SA is designed as a minimization algorithm and we are dealing with a maximization problem, we have considered f as $-\eta$. The initial solution is generated with the following values: $\dot{m}_{a,1} = 2.75$, $r_c = 26.0$, $N_c = 3$, $N_t = 3$ and $T_3 = 1250.0$, which corresponds to the values of the half of the ranges for each decision variable, in order to not skew the search direction.

The random neighborhood solutions are generated using the following procedure. To generate a new solution, the current solution is randomly modified. First, a value between 1 and 5 it is randomly drawn to decide which decision variable is modified. If the change has to be performed in $\dot{m}_{a,1}$, r_c or T_3, a number is drawn from the standard normal distribution and it is added to the current value of the incumbent decision variable. On the other hand, if the change has to be performed in one of the discrete decision variables (N_c or N_t), a number drawn from the standard normal distribution is used to determine in which direction the current value is changed, and then one is subtracted or added to the current value of the incumbent decision variable. In all cases, if the new value of the modified variable is out of its range, the value is corrected.

The SA algorithm uses the geometric cooling schedule with $\alpha = 0.8$ with an initial temperature of 1 and a stop temperature of $1x10^{-8}$. The maximum number of tries within the same temperature is set in 300.

5. Experimental Results

This section describes the results obtained by the application of the optimization algorithm to the EFGT model.

5.1. Experimental Settings

The execution platform is a PC with a Quad Core Intel i7 2600 processor at 3.40 GHz with 16 GB RAM using Linux O.S. The EFGT simulation was implemented in Octave 4.0.0. All the executions were run as single-threaded applications. Taking into account that Simulated Annealing is a stochastic technique, thirty independent runs of SA have been performed for each different fuel.

5.2. Experimental Analysis

In order to compare our experimental results with a realistic scenario, we have determined a reference configuration based on the data sheet of AE-T100E Micro Turbine Externally Fired fueled with Methane, Biomass Eucalyptus and Iso-octane. In the reference configuration the values of the decision variables are: T_3=1123.15, $\dot{m}_{a,1} = 0.8$, r_c= 4.5, N_c=1, and N_t=1.

Table 2 presents the efficiency result of the EFGT model obtained with SA and the comparison with the reference configuration. In particular, the table shows the mean (and standard deviation) and the best efficiency over 30 independent executions for the three different fuels considered (Methane, Biomass Eucalyptus and Iso-octane). Additionally, the table also presents the improvement of the best efficiency obtained with SA over the efficiency of the reference configuration as a percentage.

Table 2. Experimental results obtained with Simulated Annealing

Fuel	*Mean efficiency (± std. dev)*	*Maximum efficiency*	*Reference efficiency*	*Improvement (%)*
Methane	0.3375 (± 0.0229)	0.3501	0.2303	52.02
Biomass Eucalyptus	0.3393 (± 0.0027)	0.3457	0.2279	51.69
Iso-octane	0.3484 (± 0.0154)	0.3551	0.2329	52.47

The results of the previous table allows us to highlight that the use of an optimization algorithm produces a strong improvement in the efficiency of the EFGT. It should be noted that this improvement is independent of the fuel considered, since the percentage of improvement is similar (above a 50%) for all the evaluated fuels.

The best configuration, i.e. the configuration with the maximum efficiency, for Methane is $\dot{m}_{a,1} = 0.5$, N_c=5, N_t=1, T_3=1432.08 and r_c=8.08, for Biomass Eucalyptus is $\dot{m}_{a,1} = 0.5$, N_c=5, N_t=1, T_3=1431.28 and r_c=8.54, and for Iso-octane is $\dot{m}_{a,1} = 0.5$, N_c=5, N_t=1, T_3=1425.70 and r_c=8.00.

The experimental evaluation also shows that the best efficiency result is obtained in most of the executions (86 of 90, i.e. more than 95%) using the same values for the decision variables $\dot{m}_{a,1}$, N_c and N_t. In particular, in 27 out of 30 executions for Methane, 30 out of 30 executions for Biomass Eucalyptus and 29 out of 30 executions for Iso-octane the best efficiency is obtained with 0.5, 5 and 1 for the decision variables $\dot{m}_{a,1}$, N_c and N_t, respectively.

The difference in the values of the two decision variables (T_3 and r_c) of the best configuration that do not match for all the evaluated fuels is marginal. In the case of T_3, the best configuration is 1432.08 for Methane, 1431.28 for Biomass Eucalyptus and 1425.70 for Iso-octane. As a consequence, the difference is less than 7 K, which implies less than 1.5% of the search space for this decision variable (1000 <= T3 = 1500). The r_c decision variable shows a similar behavior since the best configuration values are 8.08, 8.54 and 8.00 for the same fuels, which stands for a 1.1% of the search space for this decision variable.

These results can be physically understand as follows:

- Related to the air mass flow rate, the less $\dot{m}_{a,1}$ implies less burned gases, and both improve heat exchanger efficiency. Therefore, it is expected that the higher values of turbine efficiency are obtained with lower values of air mass flow rate.
- Increasing the numbers of compressors, during compression stage, makes the overall process tend to an isothermal process which consumes less energy than isentropic process. Therefore, more compressors imply a better efficiency, since it increases the specific net power shaft.
- Due to system configuration, for each two turbines a reheater is needed, which means an intermediate burner, in order to increase the cold air temperature exit the turbine, to hottest

air temperature enters to the next one. These process burns more dirty fuel and releases exhaust gases at elevated temperature to the ambient. Considering fuel conversion efficiency, its value tends to decay when more fuel is burned and more and more the temperature of releases exhaust gases.

6. Summary and Conclusions

In this work, we have studied the configuration parameters of a Brayton cycle with an externally fired gas turbine as an optimization problem, considering design variables (pressure ratio, numbers of turbines and number of compressors) and operating parameters (air flow rate and inlet turbine temperature). Specifically, we designed a Simulated Annealing algorithm for maximizing the efficiency of a Brayton cycle with an externally fired gas turbine.

As main results we stress the following points:

- The experiment shows that the best result of efficiency are reached for lower values of air flow rate and numbers of turbines. On the contrary, for the number of compressors, the best result of efficiency is reached for higher values. As we stated previously, these are expected results due to physical considerations.
- It is important to highlight the applicability of the method, since it is able to find configurations that improve the efficiency around 50% over a reference configuration. Additionally, the efficiency improvements are almost independent of fuels studied in this work.
- The higher values of improvement in efficiency are expected since the best configuration implies an increase in the numbers of compressors. In future work, we plan to address a multi-objective approach including other optimization aspects as the economical cost of the system, since it is an aspect that limits the quantity of turbines and compressors of the system.

Acknowledgments

A. Durante, G. Pena and P.L Curto-Risso acknowledge support from ANII under grant FSE-1-2014-1-102079. P. L. Curto-Risso, M. Pedemonte and P. Ezzatti acknowledge support from Universidad de la República, Agencia Nacional de Investigación e Innovación and Sistema Nacional de Investigadores, Uruguay. M. Pedemonte and P. Ezzatti also thank support from Programa de Desarrollo de las Ciencias Básicas, Uruguay.

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