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ANALYSIS OF THE REAL BEHAVIOR AND OPTIMIZATION OF GAS TURBINE CYCLES

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Lucrarea prezintă o analiză comparativă, pe baze exergetice, a efectului utilizării modelului de gaz perfect şi real în studiul ciclurilor instalaţiilor de turbine cu gaze. Un interes deosebit este acordat studiului proceselor de ardere. O analiză parametrică scoate în evidenţă regimurile optime de funcţionare din punct de vedere economic şi al puterii produse. Analiza exergetică oferă strategia pentru realizarea modificărilor structurale în vederea creşterii eficienţei sistemului cu turbine cu gaze. Studiul prezintă variaţia distrugerilor şi pierderilor de exergie la modificarea parametrilor funcţionali.

The paper deals with a comparative analysis, based on the exergetic concept, of the effect of using ideal and real gas models in the study of gas turbine cycles. A special interest is focused on the combustion process. A parametric study reveals the optimum operating regimes from economic and power output point of view. The exergetic analysis gives the strategy for structural improvements of the gas turbine system. The variation of exergy destruction and losses at changes in the operating parameters is shown.

Keywords: gas turbine, energetic efficiency, exergetic efficiency, exergetic destruction, enthalpy of formation

1. Introduction

According to the Kyoto Protocol the gas emissions with green house effect have to be drastically reduced. The exergetic analysis and the use of efficient methods of investigation based on real properties of the thermal agent, represent the single strategy able to point out the useful energy destructions that take place inside the borders of the considered system. The decrease in exergy destructions and losses leads to the enhancement of the systems' efficiency.

The common way of analyzing the behavior of a gas turbine system is to consider the thermal agent as an ideal or perfect gas and to ignore the change in its

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composition across the cycle [1-5]. Even if the internal irreversibilities and the external losses are taken into account by internal and external efficiencies, the perfect or ideal gas model brings the results of the analysis away from reality.

To point out the major influence that the consideration of real gas properties and the combustion process play in the correct estimation of the system performance, a comparative analysis based on both ideal and real gas models will be done.

In most of the studies presented in the technical literature, the gas turbine systems are analyzed based on the first principle of thermodynamics only, or on the global exergetic efficiency [1-7]. Such techniques bring no much information about the causes of the internal inefficiencies and cannot give all the solutions for improving the design of the system.

In this paper, to give a better insight for improving the gas turbine system efficiency, exergetic analysis will be carried on pointing out the internal exergy destructions and making in this way the search for the optimum operating regimes and optimum system structures possible.

2. Comparative analysis of the gas turbine system performance based on the ideal and real gas model

To point out the influence of using an ideal or real gas model for the thermal agent, the comparative analysis has been carried out on a simple gas turbine system with and without regenerative recovery (figures 1, 2).

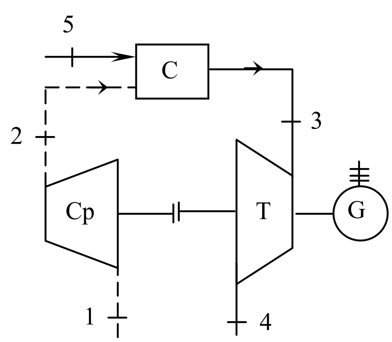

Fig.1 Simple gas turbine system

c-combustor;Cp-compressor
G-electrical generator; T-turbine

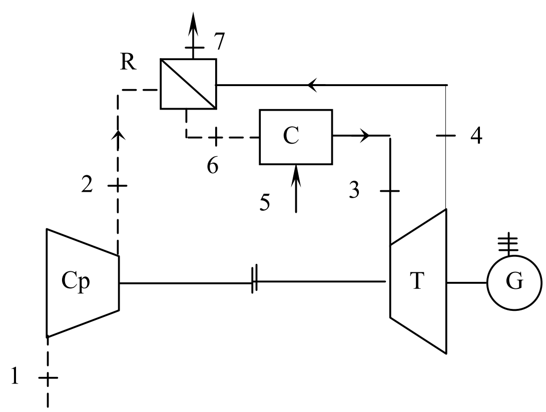

Fig. 2 Regenerative gas turbine system

C-combustor; Cp-compressor; G-electrical generator
R-recuperator; T-turbine

2.1. Ideal gas model

When using the ideal gas model for the simple gas turbine systems, the energetic efficiency and the specific mechanical work are given by the relationships (1) and (2) [1-5].

$$\eta_t = \frac{\eta_j \cdot \eta_{comb}(\theta \cdot \eta_{st} \cdot \eta_{sc} - \pi_c^{\frac{k-1}{k}})}{\eta_{sc}(\theta - 1) - (\pi_c^{\frac{k-1}{k}} - 1)} \tag{1}$$

$$w_{sp} = \frac{c_p \cdot T_1}{\eta_{sc}} \eta_j (\theta \cdot \eta_{st} \cdot \eta_{sc} - \pi_c^{\frac{k-1}{k}}) \tag{2}$$

In relationship (1), η_j is the Joule ideal cycle energetic efficiency:

$$\eta_j = 1 - \frac{1}{\pi_c^{\frac{k-1}{k}}} \tag{3}$$

In relations (1-3) the pressure drop is negligible.

It is easy to observe that, when the temperature at the turbine inlet increases (T_3, θ), the energetic efficiency calculated on the basis of the ideal gas model has no extreme and tends asymptotically to the limit:

$$\lim_{\theta \to \infty} = \eta_{comb} \cdot \eta_j \cdot \eta_{st} \tag{4}$$

The specific mechanical work of the system reaches the maximum value

$$w_{sp}^{opt} = \frac{c_p \cdot T_1}{\eta_{sc}} (\theta \cdot \eta_{sc} \cdot \eta_{st} - 1)^{\frac{1}{2}} \tag{5}$$

for the optimum value of the compression ratio

$$\pi_c^{opt} = (\theta \cdot \eta_{st} \cdot \eta_{sc})^{\frac{k}{2(k-1)}} \tag{6}$$

To reduce the energy loss through the flue gases, their heating capacity can be recovered with a regenerative air preheater (figure 2). The energetic efficiency of the system becomes for this case:

$$\eta_t = \frac{\eta_j \cdot \eta_{comb}(\theta \cdot \eta_{st} \cdot \eta_{sc} - \pi_c^{\frac{k-1}{k}})}{\eta_{sc}(\theta - 1) - (\pi_c^{\frac{k-1}{k}} - 1) - \mu[\eta_{sc} \cdot \theta(1 - \eta_{st} \cdot \eta_j) - \eta_{sc} - \eta_j \cdot \pi_c^{\frac{k-1}{k}}]} \tag{7}$$

The ideal gas model does not account for the real behavior and change in composition and mass flow rate of the thermal agent along the cycle [6].

2.2. Real gas model and combustion process

For a real model the fuel and the air composition have to be specified [7].

Let us consider that the fuel is methane (CH_4) and the molar composition of the standard air is $x^0_{N_2} = 0{,}7748$; $x^0_{O_2} = 0{,}2059$; $x^0_{CO_2} = 0{,}0003$; $x^0_{H_2O} = 0{,}019$. The delivered power of the system is Pe = 65 MW.

In the combustion process the reactants enter the combustion chamber and the different products leave the chamber.

To be able to write a coherent energy balance equation for the combustor, a unique reference must be taken for all substances participating in the reaction.

According to this aim, enthalpies and Gibbs functions of formation at the standard reference state $T_{ref} = 298{,}15$ K and p_{ref} = 1 bar, and absolute entropies are used.

Based on the common reference state the enthalpies and entropies are calculated as:

$$\bar{h}(T) = \bar{h}^0(T) \tag{8}$$

where $\bar{h}^0(T)$ is the molar enthalpy of formation, and

$$\bar{s}(T,p) = \bar{s}^0(T) - \bar{R} \cdot \ln \frac{p}{p_{ref}} \tag{9}$$

with $\bar{s}^0(T)$ the absolute molar entropy.

For the combustor the energetic balance equation, for 1 kmol of fuel, becomes [5]:

$$(\eta_{comb} - 1) \cdot LHV = \hat{n}_g \cdot \bar{h}_g(T_3) - \bar{h}_f(T_5) - \hat{n}_a(T_2) \tag{10}$$

where

$$\hat{n}_g \cdot \bar{h}_g(T_3) = \hat{n}_{O_2} \cdot \bar{h}_{O_2}(T_3) + \hat{n}_{CO_2} \cdot \bar{h}_{CO_2}(T_3) + \hat{n}_{N_2} \cdot \bar{h}_{N_2}(T_3) + $$

$$+ \hat{n}_{H_2O} \cdot \bar{h}_{H_2O}(T_3) \tag{11}$$

For e fixed inlet temperature T_3, equation (10) gives the composition of the combustion gases leaving the combustor.

2.3. Comparative performance analysis based on the ideal and real thermal agent composition models

The comparative behavior of the energetic efficiency and the calculated specific mechanical work based on the ideal and real thermal agent model, at the variation of the compression ratio π_c and the inlet temperature T_3 in the turbine, are presented in figures 3-8.

The design and operating characteristics of the equipment components are: $\eta_{sc} = 0{,}86$; $\eta_{st} = 0{,}87$; $\eta_{comb} = 0{,}98$.

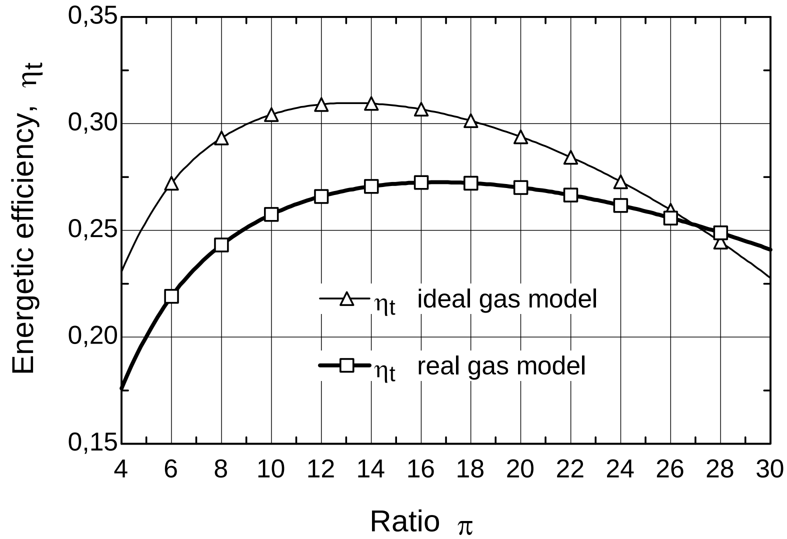

Fig. 3: Energetic efficiency against the compression ratio $\pi = p_2/p_1$ for a simple gas turbine system ($\theta = T_3/T_1 = 4$)

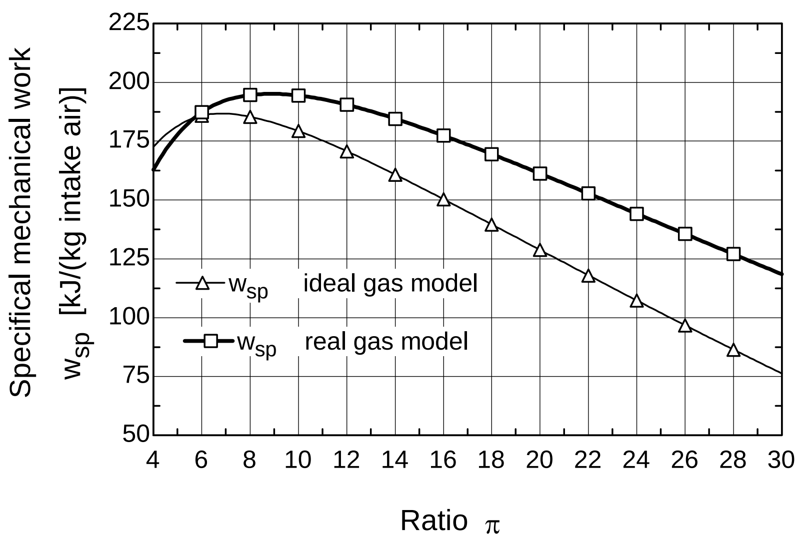

Fig.4: Specific mechanical work against the compression ratio $\pi = p_2/p_1$ for a simple gas turbine system ($\theta = T_3/T_1 = 4$)

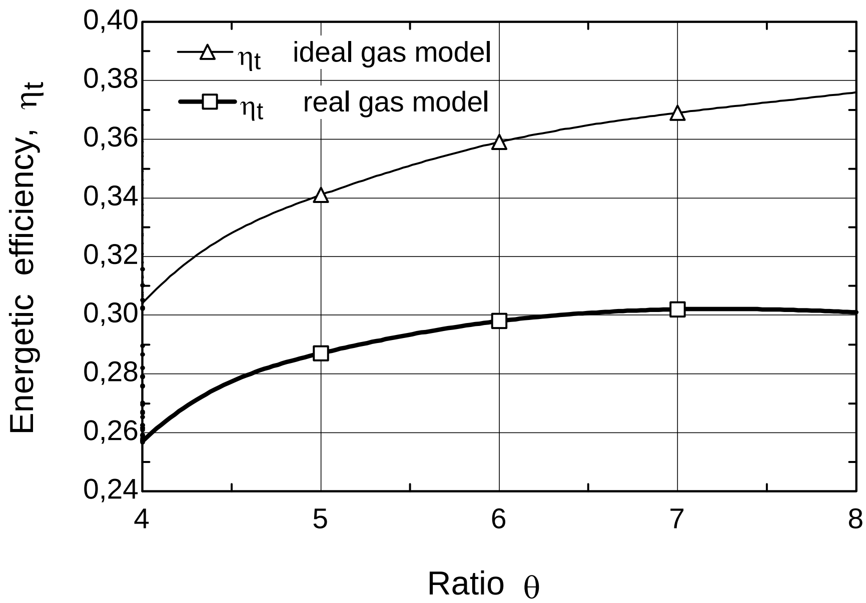

Fig. 5: Energetic efficiency against the temperature ratio $\theta = T_3/T_1$ for a simple gas turbine system ($\pi = p_2/p_1 = 10$)

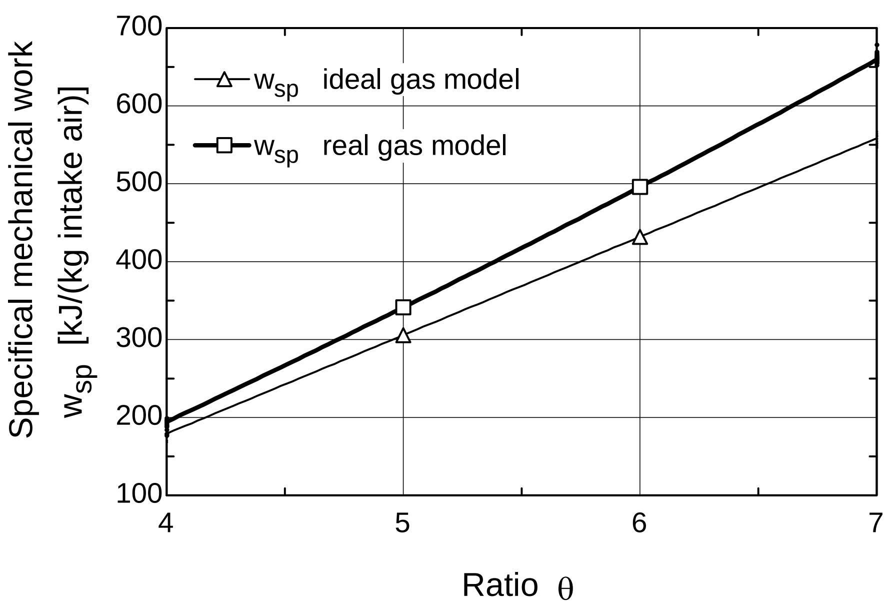

Fig.6: Specific mechanical work against the temperature ratio $\theta = T_3/T_1$ for a simple gas turbine system ($\pi = p_2/p_1 = 10$)

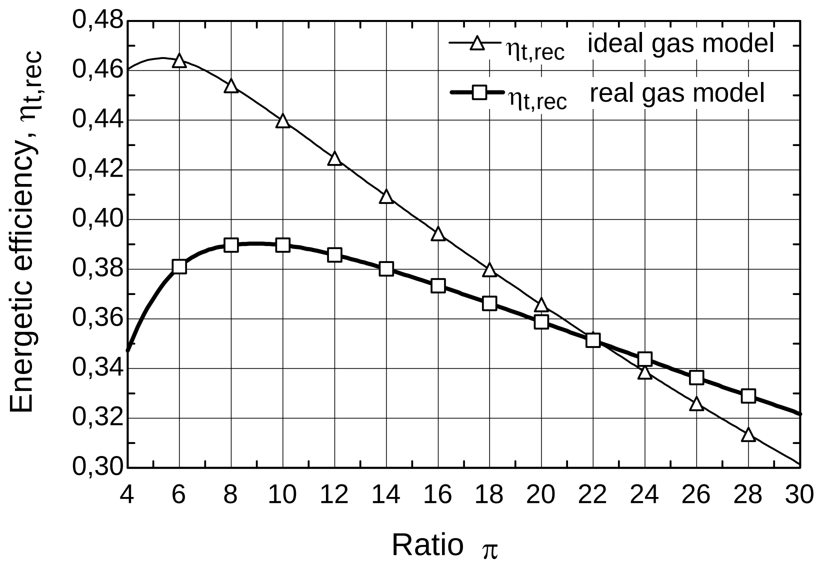

Fig. 7: Energetic efficiency against the compression ratio $\pi = p_2/p_1$ for a regenerative gas turbine system ($\theta = T_3/T_1 = 5$)

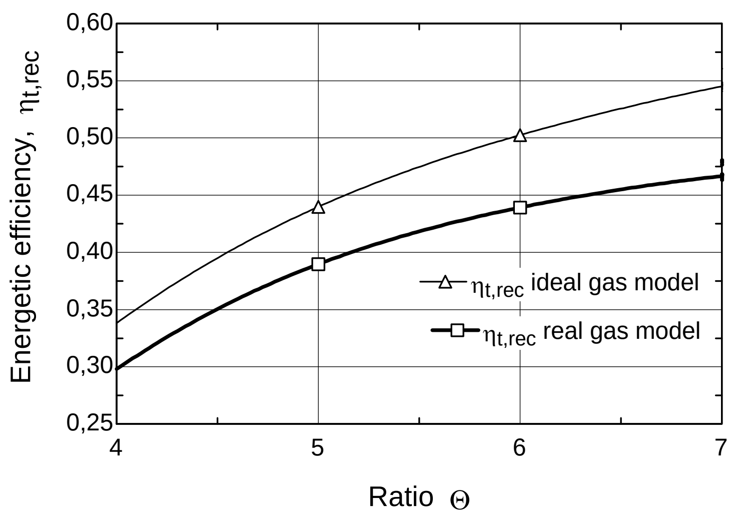

Fig.8: Energetic efficiency against the temperature ratio $\theta = T_3/T_1$ for a regenerative gas turbine system ($\pi = p_2/p_1 = 10$)

As mentioned before for the simple gas turbine system, while for the ideal gas model η_t increases continuously with θ (Eq. 4), when accounting for the real model η_t reaches a maximum (fig. 5).

Figures 3-8 show important differences between the values calculated based on the simplified model and the real case. The maximum energetic efficiency and maximum specific work move to larger compression ratios in the real case, as compared with the ideal gas model study (figures 3, 4, 7, 8).

3. Exergetic analysis

As expected, figures 3 and 7 reveal an important increase in the energetic efficiency, due to recovery of the potential heating loss, carried out by the exhaust gases otherwise.

Besides losses, the system is characterized by important internal destructions too.

The magnitude and location of these destructions can only be revealed by exergetic analysis [8-11].

The total exergy for one mol of substance, in a specified case, is:

$$\bar{e}x^{TOT} = \bar{e}x^{TM} + \bar{e}x^{CH} \tag{12}$$

where the thermomecanical exergy becomes:

$$\bar{e}x^{TM}(P,T) = \bar{h}(T) - \bar{h}_0(T_0) - T_0\left[\bar{s}(P,T) - \bar{s}_0(P_0,T_0)\right] \tag{13}$$

For the chemical part of the exergy, the combustion gases example can be considered:

$$\overline{ex}^{CH}_{gas} = \bar{R}\cdot T_0\cdot\left(x'^{g}_{N_2}\cdot ln\left(\frac{x'^{g}_{N_2}}{x^{0}_{N_2}}\right) + x'^{g}_{O_2}\cdot ln\left(\frac{x'^{g}_{O_2}}{x^{0}_{O_2}}\right) + x'^{g}_{CO_2}\cdot ln\left(\frac{x'^{g}_{CO_2}}{x^{0}_{CO_2}}\right)\right) \tag{14}$$

At the thermomechanical equilibrium with the environment the combustion gases are represented by a gaseous mixture and condensed water; x'^{g}_{i} (Eq. 14) represents the molar fraction of a component *i* in the gaseous mixture.

4. Parametric optimization of the gas turbine system

The minimum of the sum between the exergy losses and destructions leads to a maximum efficiency.

Not only the variation of the parameters of the same flow chart leads to an operating and constructive optimum, as a structural change of scheme does as well.

The exergetic analysis of a simple gas turbine system (fig.1), presented in figures 9 and 10, points out a large exergy loss with the flue gases at the outlet of the turbine. A recovery of a part of the flue gas heating potential is recommended.

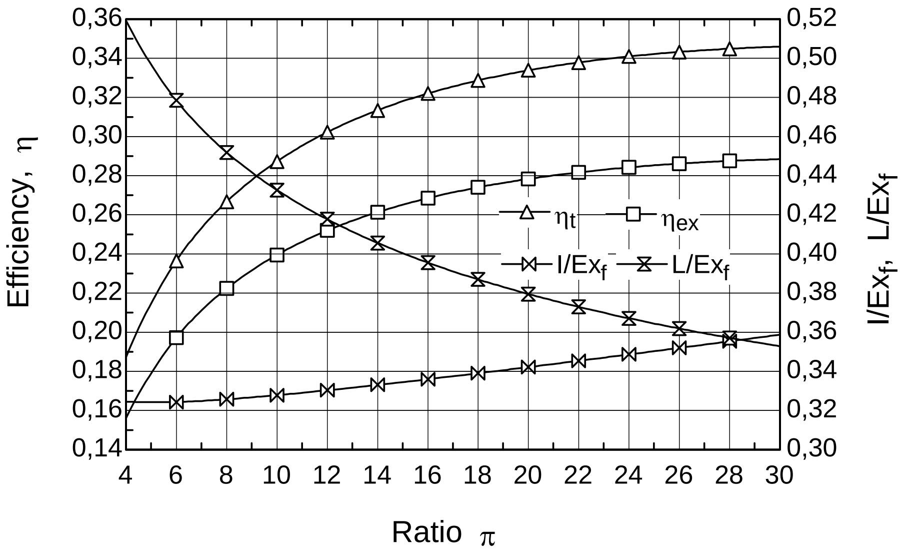

Fig. 9: Variation of energetic and exergetic efficiencies, relative losses and exergy destructions, against the compression ratio $\pi = p_2 / p_1$, for the simple gas turbine system (θ =5)

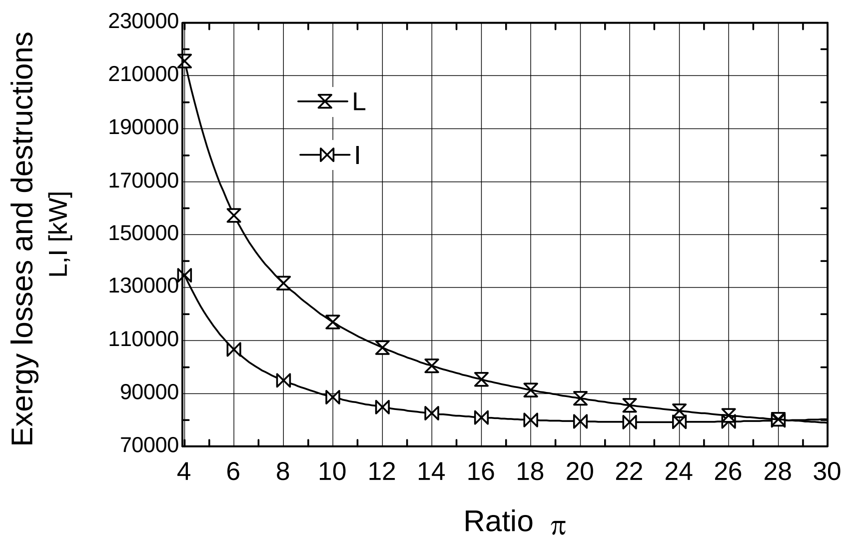

Fig. 10: Variation of total losses and exergy destructions, against the compression ratio $\pi = p_2 / p_1$, for the simple gas turbine system (θ =5)

With the aim of accomplishing this goal, figure 2 shows a structural change of the schematic of the gas turbine cycle. A regenerative heat exchanger to preheat the compressed air before entering the combustor is added to the simple gas turbine system.

The beneficial effect of this structural change is observed by comparative exergetic analyses between the behavior of the simple gas turbine cycle (figures 9,10) and the regenerative one (figures 11,12).

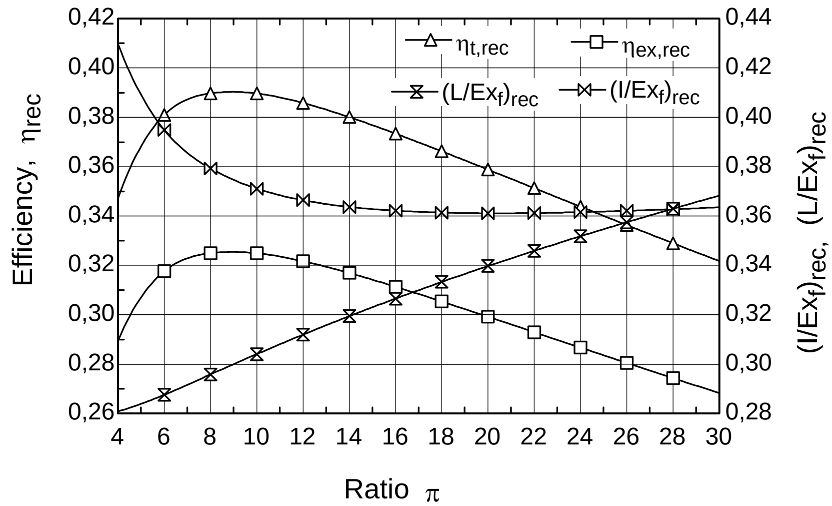

Fig. 11: Variation of energetic and exergetic efficiencies, relative losses and exergy destructions, versus the compression ratio $\pi = p_2 / p_1$, for the regenerative gas turbine system (θ =5)

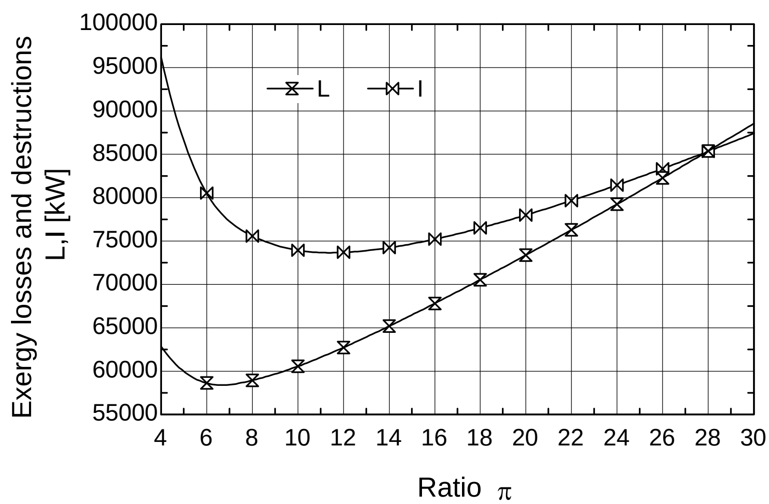

Fig. 12: Variation of total losses and exergy destructions, against the compression ratio $\pi = p_2 / p_1$, for the regenerative gas turbine system (θ =5)

If for a regenerative gas turbine system (fig.2) the exergy loss has greatly decreased, still it remains room for improvement by decreasing the internal exergy destruction (figure 12 compared to figure 10).

Observing that the maximum efficiency (and minimum exergy destruction) is reached for an isothermal compression and expansion, a two stages compression and expansion system is shown in figure 13.

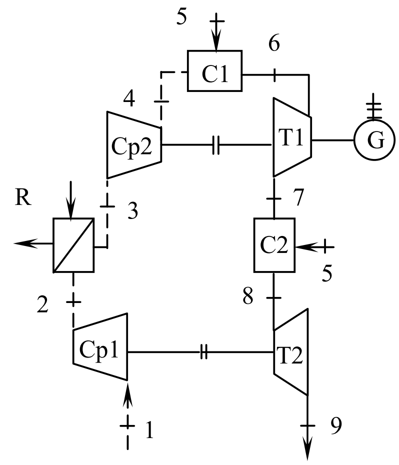

Fig.13: Two stages compression and expansion gas turbine system
C-combustor; Cp-compressor; R-recuperator; T-turbine

The exergetic performance of the gas turbine system with reheat and intercooling (fig.13) is shown in figures 14 and 15.

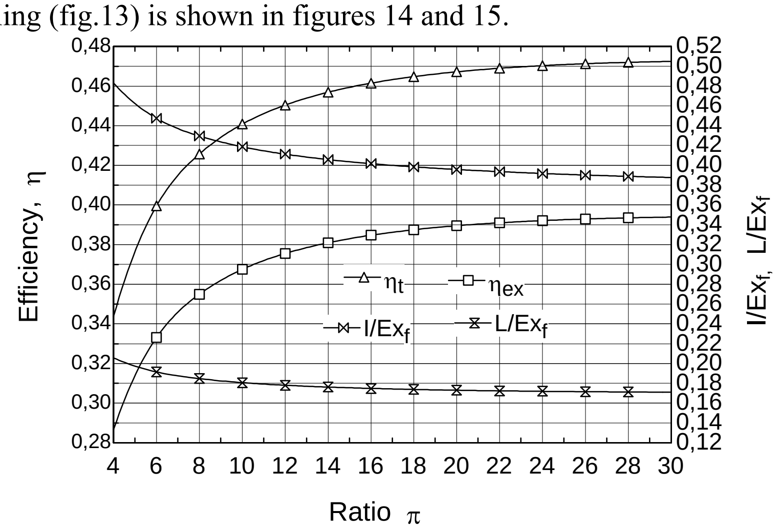

Fig. 14: Variation of energetic and exergetic efficiencies, relative losses and exergy destructions, against the compression ratio π, for the two stages compression and expansion gas turbine system ($\theta = 5$)

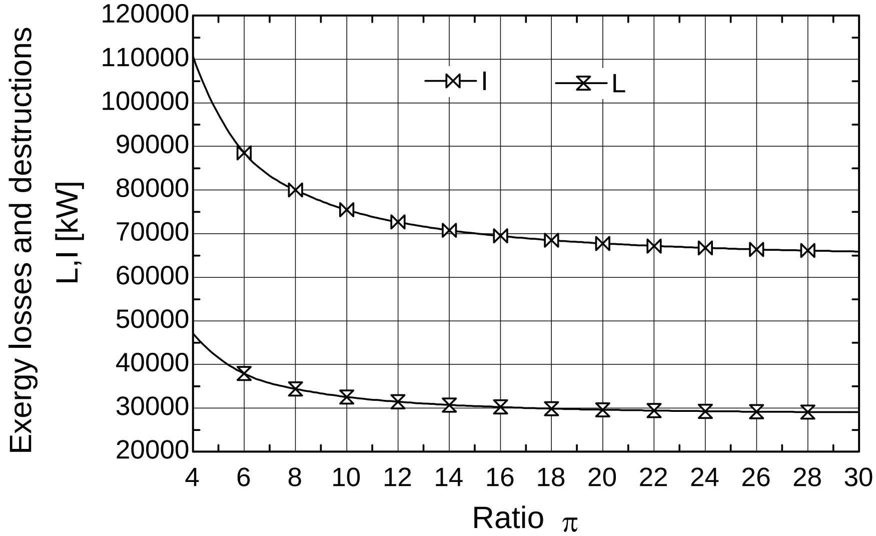

Fig. 15: Variation of the total exergetic losses and destructions, against the compression ratio $\pi = p_2 / p_1$, for the two stages compression and expansion gas turbine system ($\theta = 5$)

In comparison with figures 11 and 12, figures 14 and 15 show the decrease in the exergy loss and destruction brought by the fractionated compression and expansion.

Figures 14 and 15 show that while with this last scheme the exergy loss, mainly due to the heat potential carried on by the flue gases, reaches its minimum, there are still opportunities for improvement by reduction in exergy destruction.

5. Conclusion

The use of the energetic analysis points out only the losses of the system at the interaction with the environment. The analysis based on the first principle is not able to account for the internal destructions which are connected intrinsically to each thermodynamic process.

The energetic performances of different systems cannot be compared because they do not account for the intensive parameters of the energy carriers in connection with the intensive parameters of the environment.

The global exergetic efficiency gives the true image of the behavior of the energetic system but it is not able to give any information for improvement.

The exergetic analysis makes evident the internal exergy destructions leading to the operating and design optimization of the gas turbine systems.

The exergetic analysis is the only method able to give a strategy for structural improvement and optimization of complex systems.

The exergetic analysis of the gas turbine system has revealed the source and the behavior of the exergy destructions and losses at the variation of the decisional parameters pointing out the strategy for changes in the system structure. Based on the information given by the exergetic analysis a regenerative air preheater and a two stages compression and expansion have been introduced. The effect of these design structural changes was revealed by exergetic analysis.

The exergetic analysis leads to the operating and design optimization of gas turbine systems.

To obtain insights close to reality a real model for the thermal agent and the combustion process has to be considered.

Nomenclature

C = combustor
Cp = compressor
$\dot{E}x$ = exergy, kW
$\bar{e}x$ = molar exergy, kJ/kmol
G = electrical generator
$\bar{g}^0$ = molar Gibbs function of formation, kJ/kmol
$\bar{h}$ = molar enthalpy, kJ/kmol
$\bar{h}^0$ = molar enthalpy of formation, kJ/kmol
$\dot{I}$ = exergy destruction due to internal irreversibility, kW
k = adiabatic coefficient
LHV = Low heating value, kJ/kmol
$\dot{m}$ = mass flow rate, kg/s
n = number of kilomols
$\hat{n}$ =number of kilomols per kilomol of fuel
R = recuperator
$\bar{s}$ = molar entropy, kJ/(kmol K)
$\bar{s}^0$ = absolute molar entropy, kJ/(kmol K)
T = turbine
x = molar fraction
$\dot{W}$ = mechanical power, kW
w_{sp} = specific mechanical work, kJ/kg

Greek letters

η_{comb} = combustor efficiency
η_{ex} = exergetic efficiency
η_j =Joule ideal cycle energetic efficiency
η_{sc} = compressor isentropic efficiency
η_{st} = turbine isentropic efficiency
η_t = energetical efficiency (first law efficiency)
π_c = compression ratio

θ = temperature ratio

Indices

a = air
g = gas
f = fuel
rec = recovery
0 = equilibrium with the environment

Superscripts

CH = chemical
TM = thermomechanical
TOT = total
0 = reference state, refers to a substance from the environment

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