Computer Program for Studying the Operation of Gas Turbine Plants

Pavel ATĂNĂSOAE, Gicu OICĂ
Combined Heat and Power Plant of Suceava
str.Energeticianului nr.1, RO-720166 Suceava
patanasoae@sesv.ro; bepa@sesv.ro

***Abstract* — The combined heat and power plants equipped with gas turbines under various alternatives give an attractive solution in ensuring the heat demand under the form of hot water for the residential areas of urban settlements. Recovering the heat contained in the burning gas for steam generation and hot water supply represents the reason for using these plants in cogeneration applications. Lately, it comes out that there are used more and more average and small power solutions placed at consumers which have high performances. The paper presents a computer program made in the Visual Basic language for studying the operation of gas turbine plants. With this program, people are able to perform the thermodynamic analysis of gas turbine plants both in nominal and not nominal working regimes. The paper also presents the results of using the program for a 24 MW gas turbine.**

***Index Terms* — thermal power generation, computer aided software engineering, power system modelling, turbines**

I. INTRODUCTION

The current trend manifested in the domain of electrical and thermal energy generation consists in achieving small sources placed at consumers that have high performances. The use of gas turbines agree with this trend because of the advantages that those plants present, which consist in their compactness, high efficiency, the power level and the possibility of using them in cogeneration conditions. Also, the gas turbine plants have offered the possibility of adequate use of the burning gas heat achieved from the fuel burning, through realising combined cycles of gas-steam power plants, sensitively increasing the conversion efficiency and outrunning the usual barrier of 0,42-0,43, which is specific to the classical thermal power plants with steam turbines. It is very important to know the mode in which the gas turbine plants behave in real atmospheric conditions, adequate to the thermal plant emplacement, and more importantly, during a year when these conditions can present large variations.

II. CALCULATION METHODOLOGY FOR THE NOT NOMINAL WORKING REGIMES OF THE GAS TURBINE PLANTS

The not nominal working regimes of the gas turbine plants are divided in two great categories: not nominal working regimes generated by the atmospheric parameters modification and not nominal working regimes resulted from the electrical load variation (partial load).

The mode in which the gas turbine plants respond from the viewpoint of the performances at the working conditions modification depend on the working characteristics of the main components (compressor, gas turbine) and on the type of adjustment performed on the whole plant.

The working characteristics of the compressor and the gas turbine respectively, are functions of the following form:

$$\varepsilon_k, \eta_k = f\left(D_a, \frac{\sqrt{T_0}}{p_0}, \frac{n_k}{\sqrt{T_0}}\right) \tag{1}$$

$$\varepsilon_{TG}, \eta_{TG} = f\left(D_{ga}, \frac{\sqrt{T_3}}{p_3}, \frac{n_{TG}}{\sqrt{T_3}}\right) \tag{2}$$

The upper expressions are relatively complex. Their shape differs from one type of gas turbine plant to another, being generally determined on experimental base, too.

As can be seen, the main parameters which describe the working of the two components of the gas turbine plant are:

- the air flow and the burning gases respectively: D_a, D_{ga};
- the thermodynamic parameters at the entrance in the compressor and in the turbine: p_0,T_0,p_3, T_3;
- the rotations: n_k, n_{TG}.

The working point of the gas turbine plant will result at the crossing of the two characteristics given on the relations (1) and (2). It is mentioned that the gas turbine is one that establishes in decisive mode the working point of the gas turbine plant. In this respect, the analysis of the not nominal working regimes start from the observation that the gas flow at the entrance in the turbine remains constant on a spacious loading scale whatever the rotation value is. Therefore the following relation is valid:

$$D_{ga}\frac{\sqrt{T_3}}{p_3} = ct \tag{3}$$

The load adjustment type of the gas turbine plant is very important. This will require the mode in which a series of parameters from the relations (1) and (2) will change once with the working regime (D_a, T_3, n_k, n_{TG}).

In figure 1 it is presented the calculation methodology for the not nominal working regimes of the gas turbine plants [1]. The input data will depend on the analysed not nominal working regime type: the atmospheric parameters and the long-expected electrical load respectively.

The gas turbine plant operating in a not nominal working regime has got as an effect a change of its performances. This thing is caused by the change of the working thermodynamic point (the ε_k – T_3 parameters couple doesn't correspond the nominal point anymore) and the change of the component elements' efficiency (η_k, η_{TG}).

The main cause of the performance changes on the gas turbine plant is of thermodynamic nature. Therefore, in the performed analysis the η_k and η_{TG} variations respectively will not be taken account. This estimation does not sensitively affect the resulted qualitative conclusions.

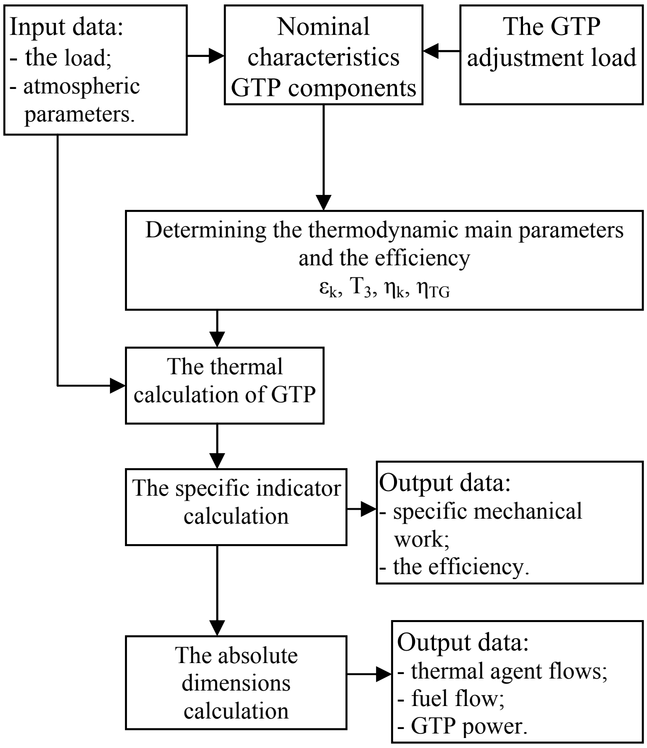

Figure 1. Calculation methodology for the not nominal regimes of the gas turbine plants.

Based on the presented methodology it was made a computer program in the Visual Basic language (figure 2). With this program, the thermodynamic analysis for the gas turbine plants can be performed both in nominal and not nominal working regimes generated by the change of atmospheric parameters or resulted from the power variation.

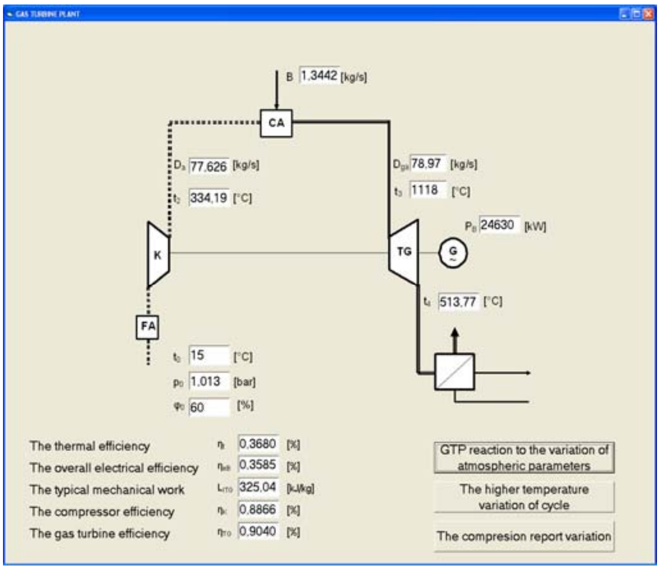

Figure 2. Computer program for studying the operation of the gas turbine plants.

III. THE EFFECT OF THE ATMOSPHERIC TEMPERATURE VARIATION

In figures 3, 4, 5, 6, 7 and 8 are presented the variations of the main working parameters from the gas turbine plant depending on the atmospheric temperature (in the conditions $p_0 = p_0^n$, $\varphi_0 = \varphi_0^n$). During the t_0 atmospheric temperature variation it is considered that the temperature at the entrance in the gas turbine remains constant and equal with the nominal value.

Assimilating the air with a perfect gas and starting from relation:

$$p \cdot V = R \cdot T \tag{4}$$

it results the flow of the mass air aspirated by the compressor for any working regime:

$$D_a = D_a^n \cdot \frac{T_0^n}{T_0} \tag{5}$$

Therefore, the growth of the atmospheric temperature leads at the mass flow diminution aspirated by the compressor when the volumetric flow is not changing (figure 3).

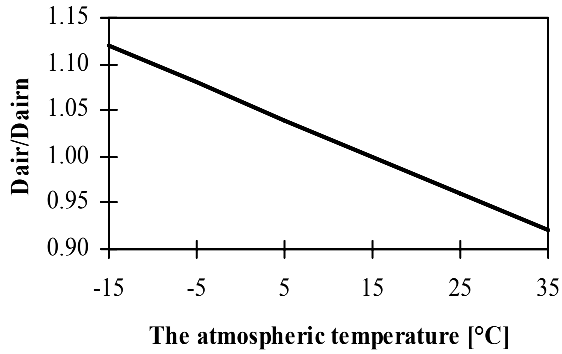

Figure 3. The relative variation of the air mass flow aspirated by the compressor depending on the atmospheric temperature.

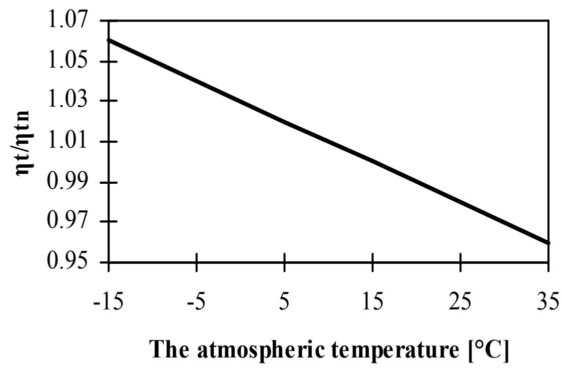

Figure 4. The relative variation of the thermal efficiency depending on the atmospheric temperature.

The diminution of the temperature at the entrance in the compressor has a favourable effect both on the thermal efficiency and on the specific mechanical work developed by the gas turbine plant (figure 4 and 5).

The power supplied by the gas turbine plant for one working regime is:

$$P_B = \eta_m \cdot \eta_G \cdot \eta_{tr} \cdot L_{GTP} \cdot D_a \tag{6}$$

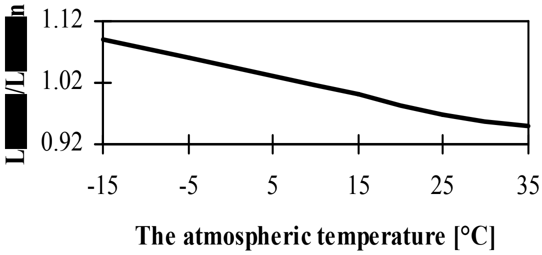

Figure 5. The relative variation of the specific mechanical work depending on the atmospheric temperature.

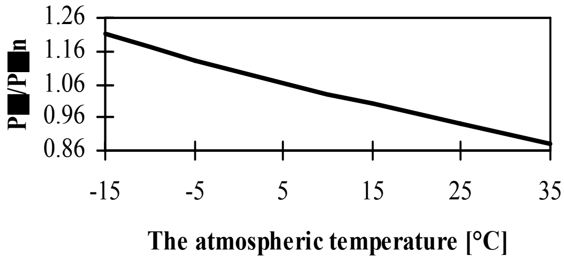

Figure 6. The relative variation of the power depending on the atmospheric temperature.

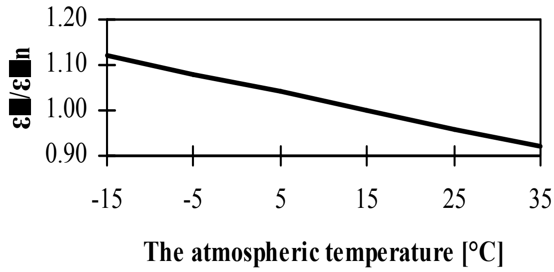

Figure 7. The relative variation depending on the atmospheric temperature.

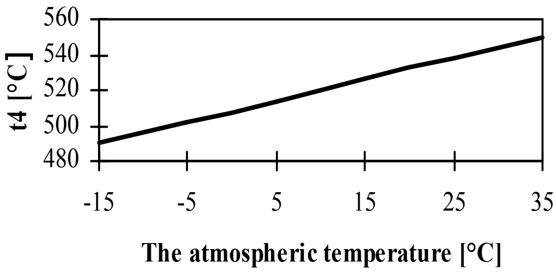

Figure 8. The variation of the exhaust temperature of the gas turbine plant depending on the atmospheric temperature.

At the diminution of the atmospheric temperature will result an increase of the power because of both increasing the specific mechanical work developed by the turbine L_{GTP} and the air flow D_a (figure 6).

When the temperature of the gas turbine remains constant at the entrance, the modification of the thermal agent flow leads at the variation of the compression report and the evacuation temperature of the gas turbine plant (figures 7 and 8). On low atmospheric temperatures, the thermal level at the gas turbine evacuation can cause problems when the gas turbine plant is stipulated with extern heat recovery (cogeneration). For removing this difficulty, on the compressor aspiration can be attached a device in which the aspirated air is heated, based on a share of the heat contained in the burning gases evacuated from the gas turbine plant.

IV. THE ATMOSPHERIC PRESSURE VARIATION EFFECT

Starting from the perfect gas equation (relation no. 4), it can be observed that when the atmospheric temperature is constant, the atmospheric pressure variation leads at the modification of the specific air volume. For a constant volumetric flow at the entrance of the compressor it will result a modification of the aspirated air mass flow and implicitly the generated power of the gas turbine plant. The relative variation of the gas turbine plant power depending on the atmospheric pressure is presented in figure 9.

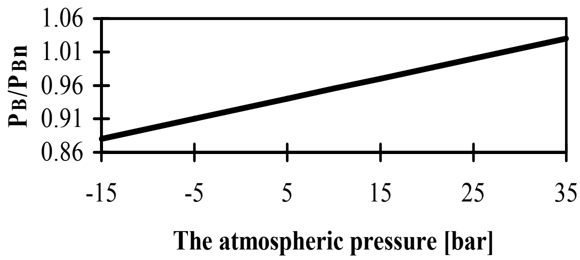

Figure 9. The relative variation of the gas turbine plant power depending on the atmospheric pressure.

V. THE EFFECT OF THE ATMOSPHERIC HUMIDITY VARIATION

The steam water density is smaller than dry air. Therefore, the increasing of the air humidity contributes at the increasing of the specific volume. Using the same reasoning used for analysing the other atmospheric parameters too, it results a diminution of the mass flow aspirated by the compressor. This function can be described by a linear relation:

$$D_a = D_a^n \cdot (1{,}003876 - 0{,}61116 \cdot x_0) \tag{7}$$

The dependence relation between the relative humidity and the absolute one respectively, is:

$$x_0 = 622 \cdot \frac{\varphi_0 \cdot p_s}{p_0 - \varphi_0 \cdot p_s} \tag{8}$$

where p_s represents the saturation pressure which is adequate to the dry thermometer temperature.

For the ISO atmospheric conditions, it results:

$$x_0^n = 0{,}006342 \text{ [kg } H_2O\text{/kg dry air]} \tag{9}$$

Figure 10 shows the relative variation of the gas turbine plant depending on the absolute humidity of the atmospheric air.

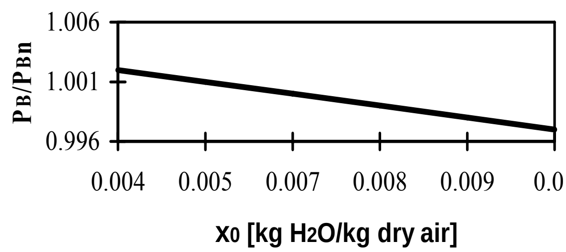

Figure 10. The relative variation of the gas turbine plant power depending on the absolute value of the atmospheric humidity.

As can be seen, the increasing of the atmospheric humidity leads to the diminution of the gas turbine plant performances.

VI. PERFORMANCE INDICATORS OF THE GAS TURBINE COGENERATION PLANTS

The burning gases evacuated from the gas turbine have a raised thermal potential, their temperature being usually between 400 and 600 °C. In these conditions it becomes interesting the heat recovery contained in these burning gases with a view to generating steam and hot water for supplying the industrial and urban consumers.

The thermal power recovered from the burning gas evacuated from turbine is:

$$Q_R = D_{ga} \cdot c_p \cdot (T_4 - T_5) \text{ [kW]} \tag{10}$$

For the temperature of the burning gases evacuated in the atmosphere it is considered the value t_5 = 125 °C, value required for avoiding the dew point.

The thermal power contained in the steam delivered to consumers is:

$$P_Q = Q_R \cdot \eta_{CR} \quad \text{[kW]} \tag{11}$$

where η_{CR} is the recovery boiler efficiency.

The recovery degree of the evacuated heat from the gas turbine plant is:

$$\beta = \frac{P_Q}{Q_4} \tag{12}$$

where Q_4 is the heat contained in the burning gases evacuated from the turbine:

$$Q_4 = B \cdot H_i \cdot \eta_{CA} \cdot (1 - \eta_t) \text{ [kW]} \tag{13}$$

The cogeneration index is defined as the report between the jack power of the generator and the thermal power generated for delivery:

$$y = \frac{P_B}{P_Q} \quad \left[\frac{kW}{kW}\right] \tag{14}$$

The gross overall efficiency of the primary energy use expression is the following:

$$\eta_{gl.brut} = \frac{P_B + P_Q}{B \cdot H_i} \tag{15}$$

The net overall efficiency of the cogeneration plant is:

$$\eta_{gl} = \frac{P_E + P_Q}{B \cdot H_i} = \eta_E + \eta_Q \tag{16}$$

where the P_E electrical power delivered is:

$$P_E = P_B \cdot \eta_{SI} \quad \text{[kW]} \tag{17}$$

The electrical energy generation efficiency of the cogeneration plant is:

$$\eta_E = \frac{P_E}{B \cdot H_i} \tag{18}$$

The heat generation efficiency of the cogeneration plant is:

$$\eta_Q = \frac{P_Q}{B \cdot H_i} \tag{19}$$

VII. CONCLUSION

The computer program allows the thermodynamic analysis of the gas turbine plants both in nominal and not nominal working regimes generated by the modification of the atmospheric parameters or resulted from the power variation. The computer program represents a useful tool in studying the working of the turbine plants and observing some important conclusions.

The power generated by the gas turbine cogeneration plant is not practically influenced by the heat recovery. The recovery boiler introduces on the burning gas circuit hydraulic resistances comparable with those introduced by a noise damper and therefore the electrical energy generation will not be affected by the heat delivery into cogeneration. The external heat recovery effect is an increasing of the primary energy overall efficiency. From a thermodynamic viewpoint this work is a result of the evacuated thermal power diminution at the cold source cycle.

The atmospheric temperature represents an element that has a great influence on the gas turbine plant performance. Thus it becomes very important to know the yearly variation of this parameter at concrete conditions of the gas turbine plant emplacement for estimating its working mode. Generally, companies that produce gas turbine plants offer correction curves depending on the atmospheric temperature for the main performance indicators (power, efficiency, turbine exhaust temperature etc.).

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