DESIGN AND OFF-DESIGN THERMODYNAMIC MODEL OF A GAS TURBINE FOR PERFORMANCE PREDICTION

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Abstract. *There are some types of faults that do not leave "signatures" in the vibration spectrum of a gas turbine. These faults can only be detected by other analysis techniques. One of these techniques is the gas turbine performance analysis or gas path analysis which relates the efficiency, mass flow, temperature, pressure, fuel consumption and power to the gas turbine faults. In this paper the methodology used in the development of a thermodynamic model that simulates the design and off-design operation of a gas turbine with a free power turbine will be presented. The results obtained are used to predict the gas turbine performance in both design and off-design operation point, and also to simulate some types of faults.*

***Keywords**: Thermodynamic model, design and off-design performance, gas turbine, fault simulation.*

1. INTRODUCTION

The main goal of current maintenance techniques is to use all analysis tools that can help in fault detection and, if possible, at the initial stage of development. As some faults can present the same symptoms, the use of different monitoring techniques, for instance, vibration and performance analysis, will increase the probability of ascertain the right fault that is developing in the gas turbine component.

With the advent of artificial intelligence techniques, data fusion became possible, for example, between models based techniques (performance analysis) and data based techniques (vibration analysis) through artificial neural network, thus allowing powerful analysis tools to be developed in the field of maintenance techniques (Marinai et al. 2004). This justifies the necessity of developing a thermodynamic model for fault detection and isolation.

Before the development of any diagnostic tool, it is essential to model the thermodynamic behavior of the gas turbine, in order to predict the engine operation point. This helps to identify the gas turbine that operates under fault condition from those which is operating in off-design condition. That is the reason for the performance prediction to be so important in the fault analysis.

2. FAULT DETECTION VIA PERFORMANCE (GAS PATH) ANALYSIS

From the point of view of faults that affect the gas turbine performance, only vibration monitoring is not sufficient for its correct identification and isolation.

Next, some types of faults that imply in performance loss and, therefore, can be identified by performance analysis technique will be presented.

2.1. Fouling

The accumulation of deposits in the blade surfaces causes an increase in the surface roughness; changes in aerofoil shape and/or in inlet angles and the reduction of the aerofoil-throat aperture. It's one of the most common causes of performance reduction found by users of gas turbine and it can be responsible for more than 70% of the performance loss during operation, according to Ogaji et al. (2002).

2.2. Tip Clearance

It influences both efficiency and flow rate capacity: an increase of 0,8% in tip clearance, for an axial compressor leads to a reduction of 3% in flow rate and a reduction of 2% in efficiency (Ogaji et al., 2002). The efficiency is more sensitive to tip clearance than to fouling.

2.3. Erosion

Surfaces exposed to impacts of particles suffer erosion and became rougher. As a result, changes occur in the blade profile increasing the blade clearance and therefore leading to a decrease in the gas turbine performance. In compressor, blades erosion leads to pressure loss in the compressor outlet and mass flow (Ogaji et al., 2002).

Another erosion effect is making the rear stages compressor susceptible to surge. Erosion of the turbine nozzles or blades increase turbine's flow and reduce its efficiency, and hence the power output.

2.4. Corrosion

When components material losses are caused by chemical reactions with contaminants entering gas turbine together with air, fuel, water or steam, the process is called corrosion. It happens most in the hot part of the gas turbine with the presence of elements such as vanadium, leading to an increase of turbine's blade corrosion (Ogaji et al., 2002). The effect is the performance reduction.

3. DESIGN AND OFF-DESIGN THERMODYNAMIC MODEL

The simulation of gas turbine performance can be divided into two categories: design and off-design point analysis.

The first category consists in selecting the best thermodynamic cycle to achieve a given (determinate) performance target: shaft power delivered, fuel consumption, etc.

This analysis is carried out, in a single operating point called design point and it represents the typical operating condition. The design point analysis allows the thermodynamic cycle optimization and the preliminary project of gas turbine components.

To estimate the performance under several operating conditions, it is necessary to create a model that has the ability to describe the engine components behavior in conditions different from that of design point analysis. This is called the off-design analysis.

The off-design analysis involves the solution of a set of linear equations in the form of $F(X) = 0$. The most used method to solve this equation set is the multidimensional Newton-Raphson (NR) method. This is the method used for the off-design model analysis.

The gas turbine model was developed using the LabVIEW (Laboratory Virtual Instruments Engineering Workbench) program).

Figure 1 presents the schematic drawing of the gas turbine with free power turbine (PT) modeled. This model includes second-order effects but it doesn't take relative humidity into account, once the component maps used didn't include this information.

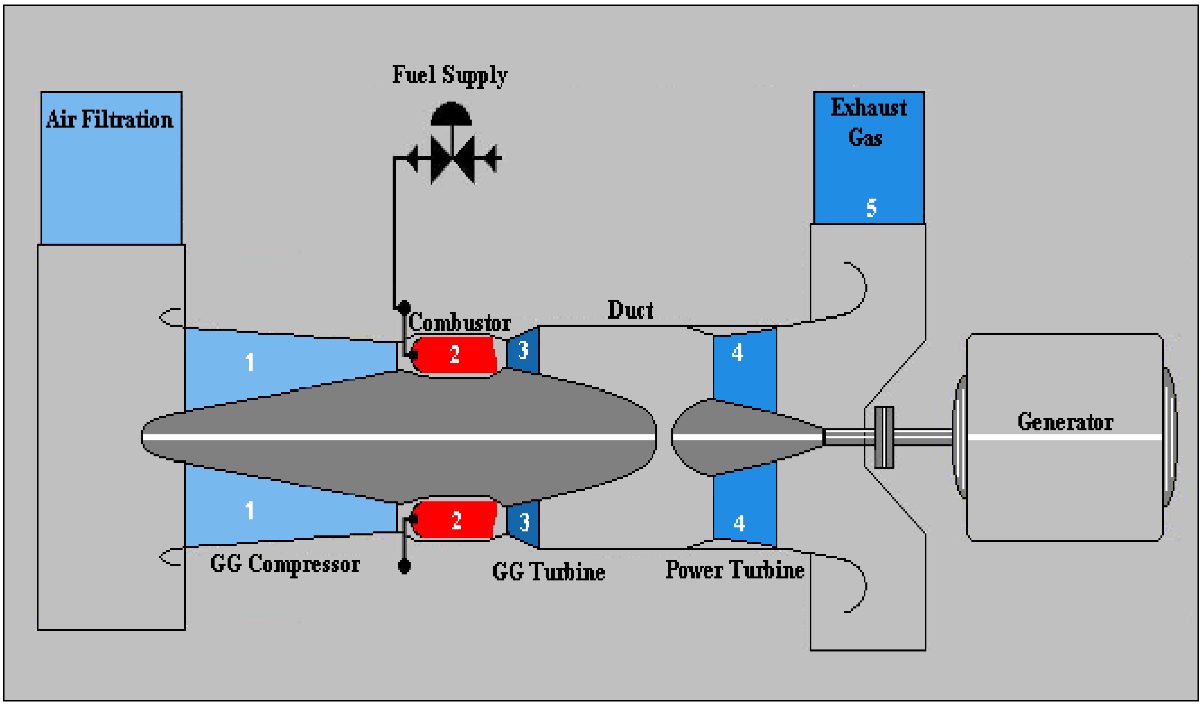

Figure 1. Gas Turbine with Power Turbine

3.1. Ambient Conditions

To calculate the gas turbine performance in an industrial plant, all data are given in a single point called ISO condition. These conditions are:

- ✓ Ambient pressure of 101,325 kPa, sea level;
- ✓ Ambient temperature of 15,8 °C;
- ✓ Relative humidity de 60%;
- ✓ Pressure losses in the inlet and outlet ducts were not considered.

3.2. Matematical Model For Admission and Discharge Ducts – Design and Off-Design Point Analysis

The inlet duct is considered as adiabatic, so the total enthalpy in the outlet (Ht_2) is equal to total enthalpy in the inlet (Ht_1),

$$Ht_1 = Ht_2 \tag{1}$$

Considering the air as an ideal gas, Eq.(1) leads to:

$$Tt_1 = Tt_2 \tag{2}$$

In design point analysis (DP), pressure loss, ΔPt, and corrected flow are known. Thus, the pressure in the duct exit can be calculated using the Eq. (3):

$$Pt_2 = Pt_1 - \Delta Pt\big)_{DP} \tag{3}$$

To calculate the pressure loss in the off-design point analysis, it is necessary to calculate the pressure loss factor, (PLF), during the design point analysis, using Eq. (4):

$$\frac{\Delta Pt}{Pt_1} = PLF \times \left(\frac{\dot{m}\sqrt{Tt_1}}{Pt_1} \right)^2 \tag{4}$$

Therefore, Eq. (5) gives the Pressure Loss Factor (PLF)

$$PLF\big)_{DP} = \Delta Pt\big)_{DP} \times \left(\frac{Pt_1}{m^2 \cdot Tt_1} \right)_{DP} \tag{5}$$

Where:

PLF [$kPa^2/(kg^2s^{-2}K)$] is the Pressure Loss Factor (PLF);
$\dot{m}$ [kg/s] is the mass flow through the duct;
Tt_1 [K]and Tt_2 [K] are the temperatures in duct inlet and outlet, respectively;
Pt_1 [kPa] and Pt_2 [kPa] are the duct inlet and outlet pressures, respectively;

For the Off-Design Point analysis (ODP) it is considered that the calculated PLF with Eq. (5) does not change, or that the pressure loss off-design point is given by Eq. (6):

$$\Delta Pt\big)_{ODP} = PLF\big)_{DP} \times \left(\frac{m^2 \cdot Tt_1}{Pt_1} \right)_{ODP} \tag{6}$$

And, thus, the pressure in duct outlet, in off-design point analysis, can be calculated using Eq. (6):

$$Pt_2 = Pt_1 - \Delta Pt\big)_{ODP} \tag{7}$$

3.3. Compressor Model

The complete model of the compressor depends on 3 stages:

1) Compressor modeling for Design Point Analysis (DPA);
2) Modification of the compressor maps (from Bringhenti, 1999) whose information (corrected mass flow, efficiency, rotation and compression ratio) will be used in compressor off-design point analysis;
3) Compressor modeling for Off Design Point Analysis (ODP);

3.3.1. Compressor Model for Design Point Analysis (DPA)

The compressor is considered as adiabatic. The compressor isentropic efficiency (η_C) is defined as:

$$\eta_C = \frac{H_{t2I} - H_{t1}}{H_{t2} - H_{t1}} \tag{8}$$

Where:
H_{t2I} is the stagnation enthalpy in the compressor exit, after an isentropic process;
H_{t1} is the stagnation enthalpy in compressor inlet;
H_{t2} is the stagnation enthalpy in compressor outlet;

The compression rate (rc) is defined as:

$$rc = \frac{P_{t2}}{P_{t1}} \tag{9}$$

Where:
P_{t2} is the stagnation pressure in compressor outlet;
P_{t1} is the stagnation pressure in compressor inlet;

The outlet total temperature, Tt2, is calculated using the polynomial that relates enthalpy with temperature. The total pressure in compressor outlet is:

$$P_{t2} = rc \cdot P_{t1} \tag{10}$$

The specific work consumed by compressor (W_C) is given by Eq. (11):

$$w_C = H_{t2} - H_{t1} \tag{11}$$

And the consumed power is given by Eq. (12):

$$\dot{W}_C = \dot{m} \cdot w_C \tag{12}$$

3.3.2. Compressor Model for Off-Design Point Analysis (ODPA)

When the compressor operates at the off-design point, its performance characteristics were not known. Therefore, a compressor map is used to interpolate the correspondent values of mass flow and efficiency for this operating condition.

The properties calculation in the compressor exit is similar to the design point model (DPA)

3.4. Combustion Chamber Model

The combustion chamber model also depends on 3 stages:

1) Combustion chamber modeling when the gas turbine operates in the design point (DPA);
2) Modification of the combustion chamber map (also obtained in Bringhenti (1999)) whose information (combustion chamber pressure, temperature difference in the inlet and exit of combustion chamber and efficiency) will be used in the off-design point analysis;
3) Off-design model of the combustion chamber using the performance maps (ODP);

3.4.1. Combustion Chamber Model For Design Point Analysis (DPA)

Combustion chamber is modeled as adiabatic. Applying the law of energy conservation, taking into account the fuel flow in combustion chamber, the combustion efficiency (η_{CC}) is given by Eq. (13)

$$\eta_{CC} = \frac{(\dot{m}_{ar} + \dot{m}_f)H_{t2} - \dot{m}_{air}H_{t1}}{\dot{m}_f \cdot LHV} \tag{13}$$

Where
H_{t2} is the enthalpy of stagnation in combustion chamber (CC) outlet;
H_{t1} is the enthalpy of stagnation in combustion chamber (CC) inlet;
$\dot{m}_f$ is the fuel flow entering CC;
$\dot{m}_{air}$ is the air flow that enters in CC;
LHV is the Low Heat Value of the fuel;

The fuel-air ratio (FAR) as a function of the enthalpy in CC outlet and the combustion efficiency is given by Eq. (14):

$$FAR = \frac{H_{t2} - H_{t1}}{\eta_{CC} \cdot LHV - H_{t2}} \tag{14}$$

Applying the law of mass conservation in CC, it leads to:

$$\dot{m}_2 = (1 + FAR) \cdot \dot{m}_1 \tag{16}$$

Where:
$\dot{m}_1$ is the air flow that enters in CC;
$\dot{m}_2$ is the combustion products flow in CC outlet ;

The CC control parameter can be the fuel flow or the outlet total temperature (T_{t2}).

3.4.2. Combustion Chamber Model For Off-Design Point Analysis (ODPA)

When the combustion chamber operates in an off-design point, its performance characteristics were not known. Therefore, it is necessary to interpolate the combustion efficiency from the component map. In order to do that, it is necessary to know the CC temperature in (ΔT) and outlet pressure (P_{t2}).

The properties calculation in the CC outlet is similar to the design point model (DPA)

3.5. Turbine Model

In this case, we have to model the compressor turbine that (CT) and the power turbine (PT).
The turbine modeling consist of:

1) Turbine model for design point analysis (DPA);
2) Modification of the turbine maps (also obtained in Bringhenti (1999)) whose information (corrected mass flow, efficiency, rotation and work parameter (ΔH/Tt1)) will be used in turbine model for off design point analysis;
3) Turbine model for Off Design Point Analysis (ODP);

3.5.1. Turbine Model for Design Point Analysis (DPA)

Turbines, as well as compressors, are considered as being adiabatic. The turbine isentropic efficiency (η_T) is defined as (Eq.(17)):

$$\eta_T = \frac{H_{t1} - H_{t2}}{H_{t1} - H_{t2I}} \tag{17}$$

Where:
H_{t2I} é is the stagnation enthalpy in the turbine outlet, after an isentropic process;
H_{t1} is the stagnation enthalpy in turbine inlet;
H_{t2} is the stagnation enthalpy in turbine outlet;

The power extracted from turbine can be used to move the compressor, as in the compressor turbine (CT):

$$\dot{W}_{CT} = \frac{\dot{W}_C}{\eta_{mec}} \tag{18}$$

Where:
$\dot{W}_C$ is the power consumed by compressor;
η_{mec} is the mechanical efficiency due to losses in the shaft transmission between compressor and turbine;

On the other hand, considering that:

$$\dot{W}_{CT} = \dot{m} \cdot \left(H_{t1} - H_{t2}\right) \tag{19}$$

Using Equations (18) and (19) it comes that (Eq.(20):

$$H_{t2} = H_{t1} - \frac{\dot{W}_C}{\dot{m} \cdot \eta_{mec}} \tag{20}$$

3.5.2. Turbine Model for Off-Design Point Analysis (ODPA)

When the turbine operates in an off-design point, its performance characteristics were not known. Therefore, it is necessary to interpolate the correspondent values of work parameters and efficiency from the component map.

The calculation procedure is similar to that of design point analysis (DPA).

4. MODEL VALIDATION / CONCLUSIONS

The model presented in this work was tested considering the outlet combustion chamber temperature (T_{t3}) as the control parameter. The results were compared with those obtained by the model developed by BRINGHENTI (1999). Figure (2) presents a comparison between the shaft power produced by the two models when the outlet combustion chamber temperature varies from 1100 to 1450 K.

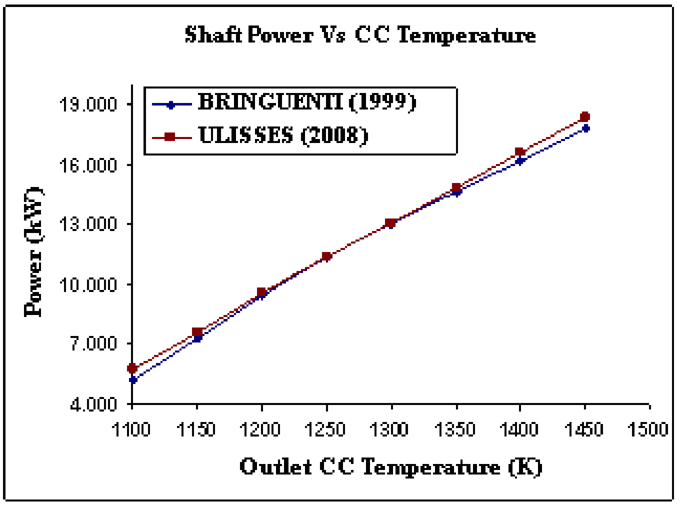

Figure 2. Shaft power obtained by Ulisses (2008) and Bringhenti (1999) models for different values of combustor chamber temperature

As can be observed in Fig. (2), in design point operation (when T = 1350 K) the results obtained for shaft power produced by the two models were very similar. However, when the gas turbine operates in off-design condition (when T ≠ 1350 K) the difference between the shaft power produced increases, reaching almost 5%. The same results are obtained if specific fuel consumption is used instead of shaft power. As a consequence of these results, small changes in component models are being done, especially in the turbine model, in order to reduce this difference.

Figures (3) and (4) show the main screen of the developed software for input data and results, respectively.

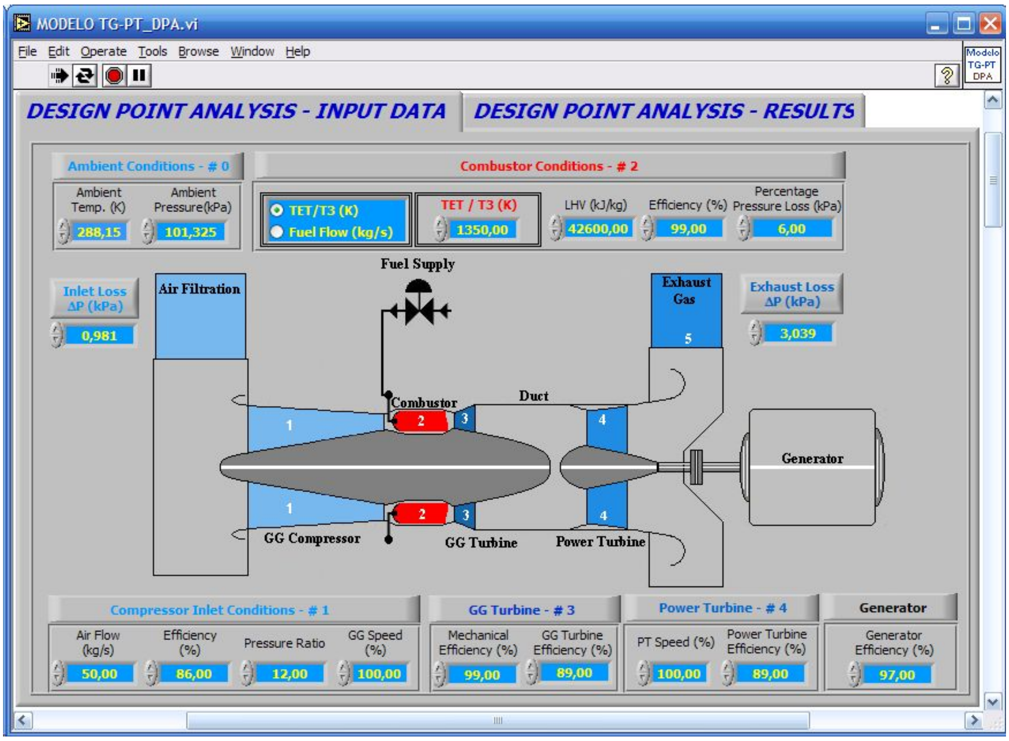

Figure 3. Main screen for the design point analysis

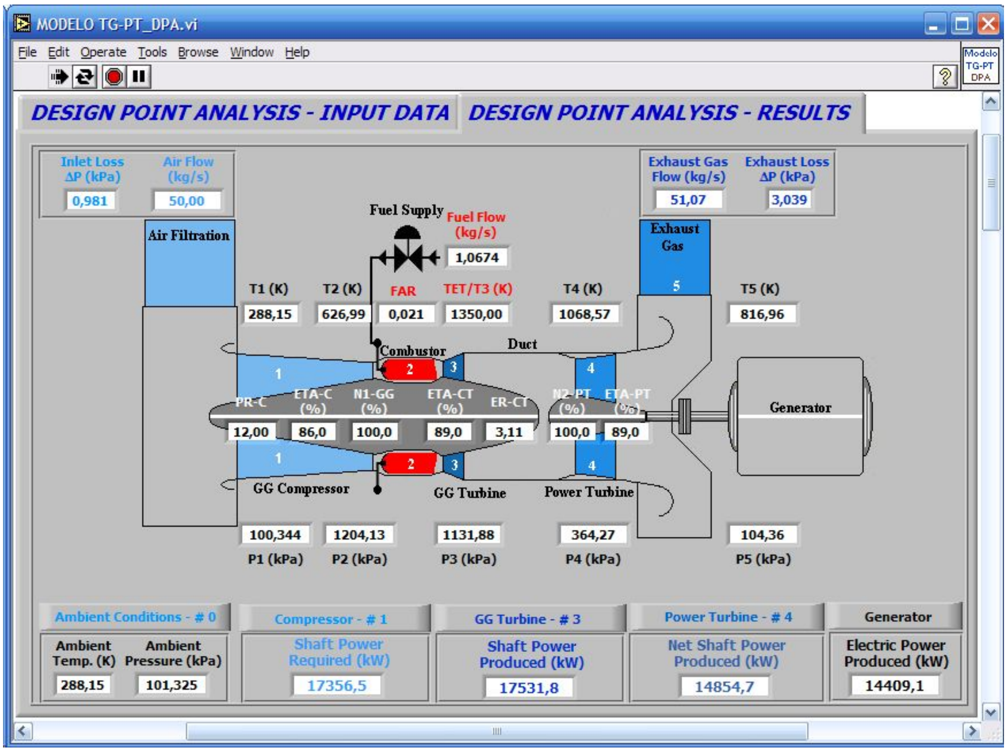

Figure 4. Screen with the design point analysis results

5. ACKNOWLEDGEMENTS

To ANP (Agência Nacional do Petróleo) for supporting this research.

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