TRANSIENT PERFORMANCE MODELLING OF A HEAVY DUTY GAS TURBINE USED FOR POWER GENERATION

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Abstract. *The Correct prediction of gas turbine transient behavior provides important information about its dymanic performance. The data produced by such a calculation is of fundamental importance for understanding and avoiding undesirable phenomena, such as compressor surge, rotating stall, and unsuccessful starting. Moreover, it is possible to gather information to feed the control system design. The aim of this paper is to present a model which has been developed to predict the transient performance of heavy duty gas turbine for power generation. The gas turbine components are modeled as blocks, which are connected to each other for energy and mass transfer. The components of the model are based on the application of the priciples of conservation os mass, energy and momentum, and the system of equation is solved by a Newton-Raphson numerical method. A set of plena is introduced in the model to account for the volumes of the main components. Firstly, the numerical model is calibrated using OEM data at design point ISO conditions. Simulation tests of the transient model have been carried out for an industrial, 83 MW, single-shaft heavy duty gas turbine. Finally, time history of the main variables that describe the gas turbine transient behavior are presented and the results are discussed.*

Keywords*: Gas turbine, Transient simulation, Off-design simulation*

1. INTRODUCTION

Gas turbine engines are a vital part of today's industry, providing both mechanical power and heat for several industry sectors, from transportation to cogeneration systems. The growing need for reliable electricity has encouraged the design of stationary gas turbine which operates on multiple fuels such as diesel, natural gas and low calorific fuels. To better monitor and control these engines, a complete analysis for prediction of transient operation is required with accompanying mathematical description, since gas turbines undergo transient operation due to startups, changing loads and sudden shutdowns.

During transient operation of heavy duty gas turbine engines, the control system must keep the limits of certain parameters, such as turbine inlet temperature (TIT) and the rotational speed (N) within their design range. In addition, the time required for the control system to react should be as short as possible to guarantee a safe and reliable operation. The turbine inlet temperature, which is a very important parameter in the performance of a gas turbine, is limited by the turbine blades material mechanical resistance. Furthermore, the rotational speed should remain constant due to the electric grid connection, which cannot withstand fluctuations of the plant frequency. An accurate prediction of the transient behavior of the gas turbine engine is essential for stable operation, fault diagnosis and also to improve the control system. Basically, the transient performance analysis deals with the operating regime where engine parameters are changing with time.

Analytical and experimental investigations of gas turbine transient behavior began around early 1950s. Several models and methods for predicting the transient behavior of gas turbine have been proposed and applied in analyzing the system dynamics. In previous work, non-linear aero-thermal models were developed using mathematical models which are composed of a set of differential equations plus a set of non-linear equations (Kim et al., 2001 and Camporeale et al., 2006).

The purpose of this paper is to present a model which has been developed to predict the transient performance of a heavy duty gas turbine for power generation. The mathematical model developed in FORTRAN accounts for the principle of conservation of mass, energy and momentum as well as the rotor inertia. The transient behavior of a typical industrial 83 MW single-shaft gas turbine equipped with compressor variables guide vanes (VIGVs), which has the design point characteristic given in Tab 1, is carried out. The model presented here is capable of performing design point, off-design and transient operation studies of any type of engine. The program has proven to be reliable, accurate, and extremely versatile. Without making any change to the program itself, a user can simulate an engine configuration merely by changing the used data sheet.

Table1. Design point parameter of a typical industrial gas turbine engine @ ISO conditions (15º C, 1atm and 60% RH).

Parameter	Value	Unit
Power Output	83.00	MW
Heat Rate	11,150	kJ/kWh
Exhaust Flow Rate	298.3	kg/s
Exhaust Temperature	537.2	ºC
Shaft Speed	3,600	rpm

2. PERFORMANCE OF GAS TURBINE

The off-design performance of gas turbine engine components is represented by the usual maps of characteristics. These can be obtained from the testing of the components or the theoretical analysis. The steady-state operating point of an engine must satisfy mass flow continuity and power balance between each compressor and turbine on the same shaft. The calculation of steady-state operation point, which satisfies the matching conditions described above, requires solving non-linear equations, which are solved by the Newton-Raphson numerical iteration method.

Engine performance while operating transiently differs from steady-state operation. The steady-state operation point determination, which is well known, depends on satisfying the essential conditions of compatibility of mass flow, work and rotational speed between the various components (Cohen et al., 1996 and Walsh and Fletcher, 1998). However, during transient operation the compatibility conditions must be modified.

The simulation of transient behavior of gas turbine needs to account for the inertia of the rotor, the gas dynamics of the lumped volume of each component and also the heat transfer between the metal parts and the fluid, which lead to changes on the dimension of the engine components. For a thorough analysis of dynamic characteristics of gas turbine, unsteady three-dimensional calculation is required. It is, however, inefficient to apply the unsteady tree-dimensional simulation, for it requires a great computational effort. It is known that the unsteady one-dimensional simulation gives sufficiently accurate results for integral engine proposes (Kim et al., 2001).

Herein, gas dynamics is modeled by one-dimensional conservation of mass, momentum and energy, applied to the different volumes identified in the engine. The different simulation models are identified by the simplifications applied to these equations. Some authors use the mass conservation equation only. Others consider the conservation of mass and energy, neglecting the momentum equation and assuming constant pressure within the volume (Alves, 2003).

The one-dimensional compressible flow and conservation of mass, energy and momentum equations, as well as the dynamic of the shaft speed, are described below.

2.1 Mass Conservation Law

The difference between the inlet mass flow and outlet mass flow of the lumped volume associated to each component leads to a mass storage which results in a derivative of the pressure with respect to time:

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}\,. \tag{1}$$

Rewriting Eq. (1) using the equation of state for ideal gas, the derivative of pressure can be obtained:

$$\frac{dP}{dt} = RT\frac{\Delta\dot{m}}{V} + \frac{P}{T}\frac{dT}{dt}\,, \tag{2}$$

where $\dot{m}_{in}$ is the inlet mass flow of control volume, $\dot{m}_{out}$ is the outlet flow of the control volume, V is the volume associated with components, R is the universal gas constant, P is the pressure and T is the temperature.

2.2 Conservation of Energy

The difference between the inlet energy and outlet energy of the volume associated with each component will generate energy storage and the rate of change from the mass storage and the rate of change of specific internal energy,

$$\frac{dU}{dt} = \Delta(\dot{m}H) + \dot{Q}\,, \tag{3}$$

and

$$\Delta(\dot{m}H) = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out}\,, \tag{4}$$

where U is the specific internal energy, h_{in} is the enthalpy at the control volume inlet, h_{out} is the enthalpy at the control volume outlet and $\dot{Q}$ is the heat flux.

2.3 Conservation of Linear Momentum

From Newton's Second Law of Motion it is possible to calculate the mass storage inside the volume:

$$\frac{dm}{dt} = \frac{A \cdot \Delta(kP)}{L} - \frac{R_x}{L}, \tag{5}$$

and

$$R_x = \tau_w \pi D L, \tag{6}$$

$$k = \frac{1+\gamma M^2}{\left(1+\frac{\gamma-1}{2}M^2\right)^{\gamma/\gamma-1}}, \tag{7}$$

where A, L and D are the area, the length and the diameter of the volume, P is the pressure, τ_w is the friction, M is the Mach number and γ is the rate of specific heats at constant pressure and constant volume, respectively.

2.4 Rotating Shaft

The difference between the power delivered by the turbine and the power absorbed by the compressor and electric generator imposes a change of rotational speed within a certain period, i.e., acceleration and deceleration. If the engine is operating at equilibrium and is suddenly subject to an increase in fuel flow, the turbine temperature will increase much more rapidly than any increase in rotor speed because of the inertia of the rotor system. For a heavy duty gas turbine, the effect is an excursion along the constant speed line on the compressor characteristic towards surge. A sufficiently large step change will cause the compressor to surge.

The angular acceleration or deceleration, ω, of the shaft joining compressor, turbine, and applied load is given by the angular momentum equilibrium, depending on the inertia which includes the effects of the shaft and of the other connected devices (Camporeale et al., 2006),

$$\frac{d\omega}{dt} = \frac{1}{I\omega}\left(P_t - P_c - P_e\right), \tag{8}$$

where P_t is the power produced by the turbine, P_c is the power absorbed by the compressor, P_e is the power absorbed by electric generator and I is the momentum of inertia of the shaft and the connected equipments. There are several methods of numerical integration; Euler's explicit method is used in this paper. Explicit methods may lead to unstable behavior of the solution as they involve the prediction of the state variable at the next time step based on conditions of the current time step, however a judicious choice of time step can minimize such problems.

3. CONTROL SYSTEM

A typical gas turbine consists of three complex equipments in series: the compressor, the combustion chamber and the expander. The stable operation of the whole gas turbine depends on the stability of these three equipments. Generally, the range of operation is relatively small due to the instability of the compressor, limits of temperature in the turbine and flame stability within the combustion chamber. In compressors, the margin of instability, known as surge line, prevents the operation on pressure values above a certain threshold, which is a function of engine speed and mass flow. If the machine approaches this limit during transient operation, the control system of the gas turbine must act on a blow-off valve, or change the angle of the compressor inlet guide vanes (VIGV - Variable Inlet Guide Vane) to avoid instability. Thus, the main variables of control that can be manipulated in an industrial gas turbine are: (i) the fuel flow, (ii) the VIGV angle, and (iii) the blow-off flow.

Some turbine models, that also allow the control of variable vanes, known as nozzle guide vanes (NGV) located at the turbine inlet, just downstream to the combustion chamber. However, this operation system is more complex than the VIGVs due to the high temperatures involved, and is seldom used.

During transient operation the rotational speed of the gas turbine for power generation must remain constant. Two strategies to control the power generated by the gas turbine are possible. The first is the control of the gas turbine varying only the fuel flow. This strategy is simple and does not take into account the variation of air mass flow through the compressor by VIGVs. It is used when the gas turbine operates in "open loop", i.e., without the addition of a heat recovery boiler at its exhaust. In this situation, the control system compares the actual value of the rotational speed with a set point value. The difference between these two signals is an error signal, which is used to modulate the fuel flow valve, according Fig. 1.

The other strategy is to control the air flow into the compressor using the inlet guide vanes (VIGVs). The purpose of this strategy is to improve the performance of the Heat Recovery Steam Generator (HRSG), and consequently increase the efficiency of a combined cycle by maintaining high turbine exhaust temperature. As the air flow is reduced, closing the VIGVs, fuel flow control must also actuate in order to avoid large increases in the turbine inlet temperature (TIT) and also maintain a constant rotational speed. One methodology used to model a variable geometry gas turbine, for which geometric data is not available, involves the use of correction factors to modify the performance maps that represent the characteristics of the main components of the engine (Celis et al., 2008). These factors were used in the proposed model to simulate the actual compressor behavior during the power plant operation.

Due to high values of the turbine inlet temperature, which are impractical to measure directly, typically the manufacturers use, as a parameter to determinate its value, the turbine exhaust temperature and the compressor discharge pressure. However, in this study, the turbine inlet temperature, which is calculated, is used directly to control the VIGVs angle. The turbine inlet temperature is calculated and compared with a reference temperature, and an error is estimated to modulate the VIGVs angle, according Fig. 1.

The control system must ensure that the critical operating limits of rotational speed and turbine inlet temperature are never exceeded, and that compressor surge is avoided (Cohen et al., 1996). Various kinds of control logic have been proposed, such as PID control, state space control, etc. In this work, a conventional PID control is used to control the fuel flow and de VIGVs, as shown in Fig. 1 (Rowen, 1992, Hanent and Khan, 1993 and Yee et al., 2007). The control gains were tuned based in operation data engine. The inputs of the computational model are the ambient temperature and pressure and the power generated during a period of time. As shown in Fig. 1, following the VIGVs controller there is a limiter which represents the upper and lower limits of the VIGVs.

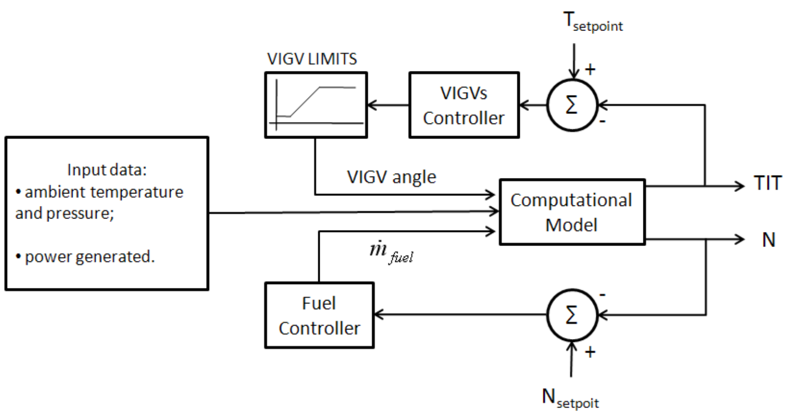

Figure 1. Block diagram of the implemented control system.

4. RESULTS AND DISCUSSION

Before evaluating the transient performance of a gas turbine it is necessary to simulate the performance of the engine on steady-state operation. To validate the proposed model, an off-design simulation of a heavy-duty gas turbine was carried out and the results are compared to operational data. The predicted results are shown in Fig 2. The results show good agreement with engine manufacturer data. Figure 2 also shows the usual procedure that is employed when the gas turbine operates at part load. At base load the VIGVs is totally open, indicating a high air mass flow into the compressor to a given ambient temperature. This condition leads to the maximum power that the gas turbine can deliver. If the power demand is reduced, this new condition will be achieved closing the VIGVs, thus reducing air mass flow into the compressor. As the gas turbine engine is operating in combined cycle mode, it is usual to maintain a constant firing temperature (TIT), what explains the exhaust temperature is slightly increased. This procedure is maintained until the limit of the VIGVs angle is reached. From this point onwards, additional power reduction is provided only by decreasing the fuel flow (Kim and Hwang, 2004).

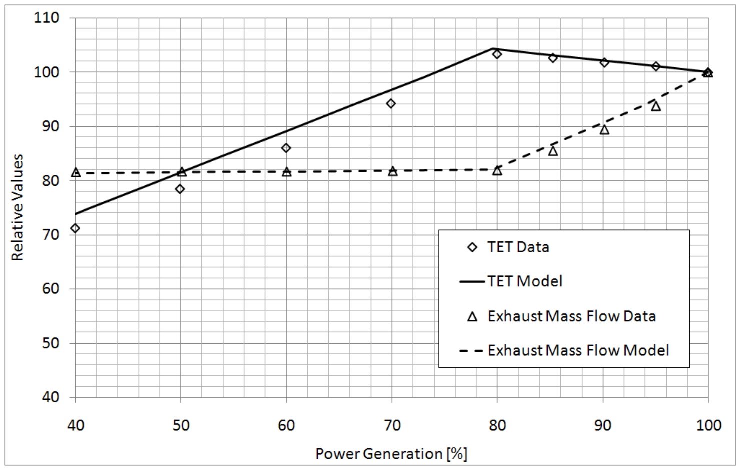

Figure 2. Off-design simulation compared with engine manufacturer data (ISO conditions).

The validation of the transient performance was done from operational field data, which is sampled every second during 52 minutes. This data is of outstanding nature, since during this period of time there are large (50%) variations in power generation, according Fig 3. The input data for the computational model is: (i) the ambient temperature and pressure and (ii) the power generated by the gas turbine.

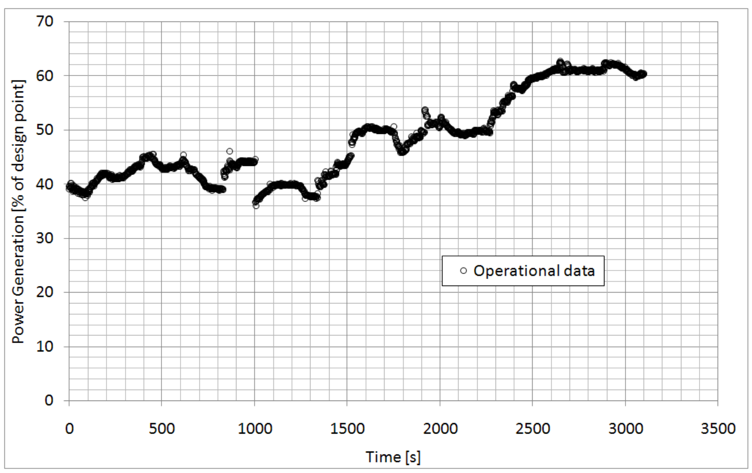

Figure 3. Power generated during transient operation.

Figure 4 shows a comparison between the modeled VIGVs angle and the operational results. Despite the discrepancies observed, the trends of simulated data are in agreement with the measured data. During 2,300 s, the VIGVs angle is totally closed. From this moment onwards, the control system modulates the VIGVs angle to maintain as high as possible turbine exhaust temperature, which leads to an increase on the efficiency of the combined cycle.

It can be observed that the model results exhibit small oscillations during the opening and closing of VIGVs angle, whereas the operating data is characterized by stepwise increments. This is explained as follows: the control system implemented in the computational program has a constant set point of temperature, thus every time that the turbine inlet temperature changes the control system closes or opens the VIGVs angle to maintain its constant. On the other hand, it could be speculated that the real control system has a range of operation values where the modulation of VIGVs angle does not change until a certain limit of temperature is reached.

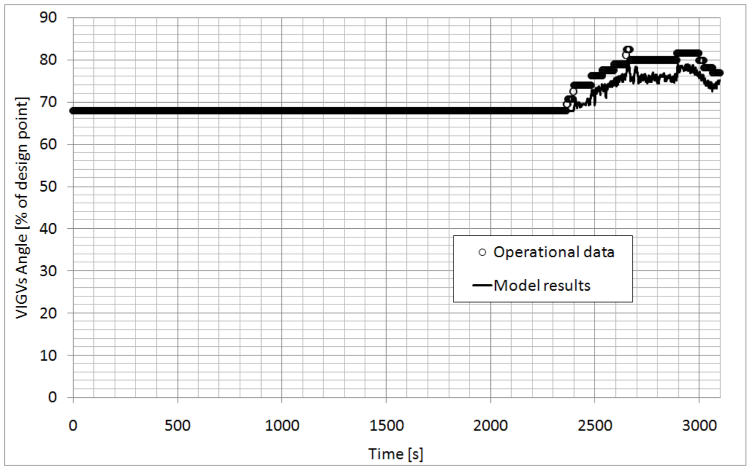

Figure 4. Comparison of compressor inlet guide vanes (VIGVs) transient simulated and engine data.

Figure 5 shows the comparison between the modeled turbine exhaust temperature and the measured engine power. It is observed that the trend of the computational model results is in agreement with the operational data. It is important to note that the average error for simulated data is less than 4%.

During the first 2,300 s the turbine exhaust temperature (TET) changes in accordance with variation of power generation. During this period the VIGVs are totally closed, thus all changes in power generation required by the electric system is done by controlling the fuel flow, which affects directly the value of TET. From 2,300 s until the end of transient period, the TET is maintained at the vicinity of its design value by modulating the VIGVs and also the fuel flow.

Note, also, that the modeled values are higher than the operational engine data ones. This is due to the fact that the turbine inlet temperature is not specified by the manufacturer, its value was estimated, thus the discrepancy observed. Another source of error are the compressor map, which was not taken from OEM's data, and the unknown measurement uncertainty associated to the instrumentation of the power plant.

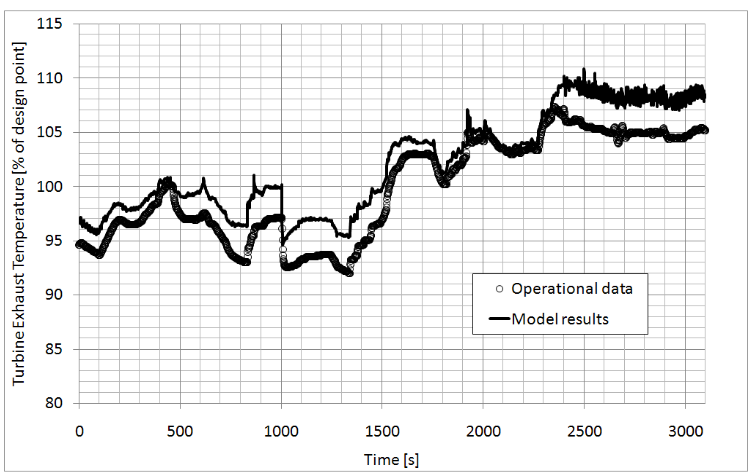

Figure 5. Comparison of turbine exhaust temperature transient simulated and engine data.

Figure 6 shows the comparison between the modeled fuel flow and the corresponding measurements at the power plant. One should observe that the model results are in accordance with the trend of operational data, presenting small deviation, less than 5%, at the beginning and at end of the transient regime. This discrepancy may be explained by the uncertainty that exists in the value of turbine inlet temperature assumed in the computational model, which is possibly different from the actual design value of the equipment. Generally, this information is not provided by manufacturers. Another source of discrepancy, although with a discrete impact, is the lower calorific heat value used in the computational model, which is constant, and the actual value used in the engine.

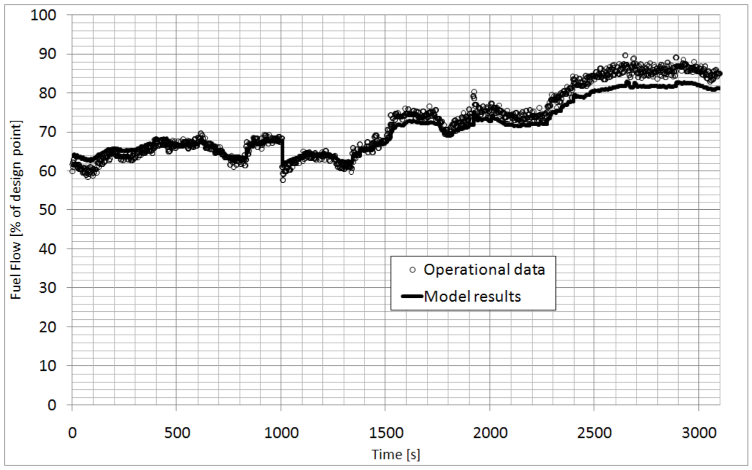

Figure 6. Comparison of fuel flow transient simulated and engine data.

Figure 7 shows a comparison between the calculated compressor discharge pressure and the operational data. Although the discrepancies are larger than those exhibited by other results, more than 10%, there is an agreement with the overall trend of the measured data. This discrepancy may be explained by the fact that the operation of a gas turbine is heavily dependent on the particular characteristics, or maps, of the compressor and expander. These maps are measured during the design of the equipment and are not provided by manufactures. For this simulation, a generic map is used, hence the discrepancy observed.

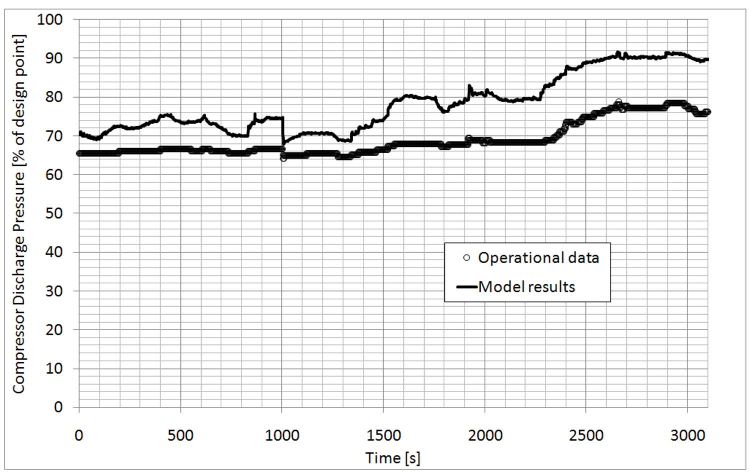

Figure 7. Comparison of compressor discharge pressure transient simulated and engine data.

Finally, Figure 8 shows the comparison between the calculated rotational speed and the operational data. Again, it is observed that the trend of the results of the computational program is in agreement with measured values. Note that the average error for simulated data is less than 2%.

It is also observed that instantaneous changes that occur during sudden load changes are simulated by the model. During the first part of the transient operation, the computational results have the higher discrepancies. The rotor inertia, which is a very important parameter in the transient modeling of the rotational shaft, is not available, thus its value was assumed.

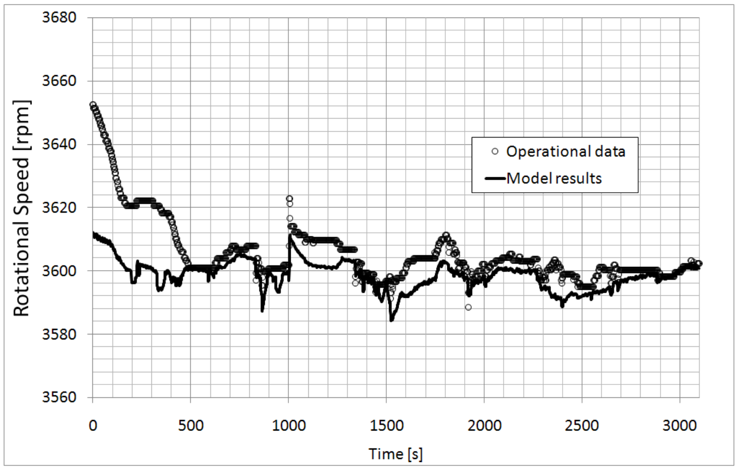

Figure 8 - Comparison of rotational speed transient simulated and engine data.

5. CONCLUSIONS

In this work a mathematical representation of a transient behavior of a heavy-duty gas turbine with control system was presented. The model developed in FORTRAN is capable to simulate the behavior of compressor variable inlet guides vanes (VIGVs) with accurate precision. The use of correction factors to modify the performance maps that represent the characteristics of the main components of the engine was done for this purpose.

It was also implemented the control system of fuel flow and VIGVs. Comparison between a real engine demonstrated that the model developed is able to simulate the trend of a transient operation of a heavy-duty gas turbine. The average error for almost simulated data is less than 4%. The discrepancies observed on compressor discharge pressure are larger than those exhibited by other results, more than 10%. This discrepancy may be explained by the fact that the operation of a gas turbine is heavily dependent on the particular characteristics, or maps, of the compressor and expander, which are not provided by manufactures.

The model developed consists of a project which aims to develop a computational program capable to simulate the performance of a combined cycle power plant. It can be also used in an on-line monitoring and fault diagnosis system of gas turbines. Therefore, the model developed is a piece of powerful tool to assist and improve the operation and maintenance of power plants.

Despite the assumptions made in the simulation, the results are deemed good enough to demonstrate the robustness of the computational code and the new features developed to simulate VIGV and fuel control systems. Differences between actual figures and those simulated are very encouraging.

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