COMPARISON OF INDUSTRIAL GAS TURBINE TRANSIENT RESPONSES PERFORMED BY DIFFERENT DYNAMIC MODELS

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ABSTRACT

In this paper, the transient responses of different dynamic models of a generator drive single shaft gas turbine working in parallel with electrical mains, have been analyzed.

The time variations of the main thermodynamic and performance parameters are obtained by varying the fuel valve position, for an assigned IGV angle, and by varying the IGV angle for an assigned fuel valve position.

The influence of the dynamic phenomena due to the working gas mass contained in the gas turbine component volumes, of the metal mass thermal inertia and of the heat transfers, are taken separately into account. The dynamic balance of rotating masses is not instead considered, since the single shaft gas turbine works in parallel with electrical mains.

The developed analysis has shown the negligible influence of the phenomena due to the working gas mass contained in the gas turbine component volumes and a more significative effect of the metal mass inertia and of the heat transfers on the transient response. In particular, this influence is significative only for gas turbine and compressor outlet temperatures.

The influence on the gas turbine transient behaviour of the control system that uses the gas turbine outlet temperature in feedback has been also evaluated.

NOMENCLATURE

C  torque
c  specific heat
c_p  specific heat at constant pressure
F  friction force
g  gravity acceleration
I_g  moment of inertia of rotating masses connected to the gas turbine shaft reduced to the shaft speed
K  heat transfer coefficient
l  linear coordinate
M  mass flow rate
Ms  shell mass
N  rotational speed
P  power
p  pressure
Q  thermal power
T  stagnation temperature
t  time
v  velocity
z  altitude
β  pressure ratio
ρ  density

Subscripts
0  initial steady-state condition value
c  compression, compressor
cc  combustor
f  fuel

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INTRODUCTION

A large number of different dynamic gas turbine models have widespread utilization in research and in the technical world (DeHoff and Hall, 1978; Szuch, 1978; Schobeiri, 1987; Krikelis and Papadakis, 1988; Blotenberg, 1993; Bettocchi et al., 1996). These models differ from one another on the basis of lower or greater accuracy, as well as on the dynamic phenomena considered as follows:

- dynamic balance of rotating masses;
- dynamic phenomena due to the gas mass contained in the gas turbine component volumes;
- dynamic phenomena due to flow mixing;
- dynamic phenomena due to thermal inertia of the metal mass and heat transfers.

The importance of the various dynamic phenomena depends both on the type of gas turbine and on its use. For example, the transient behaviour of a multi-shaft gas turbine aircraft engine is very different if compared to that of an electrical drive single shaft gas turbine. As a consequence, the same dynamic phenomena considered in the model will weigh differently in the two cases.

In this paper, in particular, an analysis was made of the transient responses performed by different dynamic models of a generator drive single shaft gas turbine, with variable compressor IGV angle, working in parallel with electrical mains.

CONSIDERED MODELS

In order to evaluate the influence of the various dynamic phenomena on the transient response of a single shaft gas turbine with variable compressor IGV angle, two different models were considered.

The first one (MOD. 1), developed by Bettocchi et al. (1996), uses the SIMULINK tool (MathWorks, 1991) of MATLAB software (MathWorks, 1990).

The dynamic phenomena considered in this model are related to the dynamic balance of rotating masses and to the working gas masses contained in the gas turbine component volumes.

The dynamic balance of rotating masses is expressed by means of the following usual differential equation:

\[ \frac{2\pi}{60} \frac{dN}{dt} = C_t - C_c - C_r \]  

The dynamic phenomena related to the working gas masses are described by the mass and momentum balance equations. These equations, in the hypothesis of assimilating each gas turbine component to a constant section duct, have the following forms:

\[ \frac{\partial p}{\partial t} + (pv) \frac{\partial}{\partial l} = 0 \]  

mass balance (2)

\[ \rho \frac{\partial v}{\partial t} + pv \frac{\partial v}{\partial l} = -\left( \frac{\partial p}{\partial l} + F + pg \frac{dz}{dl} \right) \]  

momentum balance (3)

The dynamic phenomena due to flow mixing, the thermal inertia of the metal and gas masses and the heat transfers between gas and shell and between shell and environmental are instead, neglected.

The effect on the thermodynamic cycle of turbine nozzle and blade row cooling flows is assessed by splitting the total cooling flow appropriately into two parts. It is assumed that one is mixed upstream and the other downstream from the turbine, causing a reduction of the main flow total temperature and then, a reduction of the available enthalpy drop (Benvenuti et al., 1993).

The second gas turbine model considered (MOD. 2) is a module of MMS program (Framatome Technologies, 1996). In this model, differently from the previous one, the dynamic phenomena related to thermal inertia of the metal mass and to heat transfers between gas and shell are taken into account, along with the dynamic balance of rotating masses.

The dynamic phenomena related to the working gas masses contained in the gas turbine component volumes and to flow mixing are instead neglected.

Concerning the effect of cooling flow, it is considered that it mixes downstream from the turbine.

The dynamic balance of rotating masses is expressed by the equation (1), while the thermal inertia and the heat transfers are described by means of the following equation:
\[ Q = \frac{dT_s}{dT} = K (T_o - T_s) \tag{4} \]

where \( K \) is the heat transfer coefficient. In the case of conductive heat transfer, \( K \) is a constant, while in the case of convective heat transfer it is:

\[ K = K_1 \left( \frac{M}{M_0} \right)^{0.8} \tag{5} \]

The thermal power is considered equal to the maximum between conductive and convective heat transfer.

The eq. (4) is used for both the compressor and turbine, assuming that, for each component, the overall metal mass is located in an energy storage node at its exit. In eq. (4), temperatures \( T_o \) and \( T_s \) represent the gas temperature at the component node exit and the shell average temperature through the component, respectively.

In each component, the amount of thermal power between gas and shell is expressed by:

\[ Q = M c_p (T_i - T_o) \tag{6} \]

where \( T_i \) is the temperature at the node inlet due to the compression or expansion transformation. In particular for a cooled turbine, it represents the temperature downstream from the mixing point between main and cooling flows.

For both the models:
- the equations representing the thermodynamic transformation are used in a stationary form;
- the compression and expansion transformations are computed by means of non-dimensional map;
- the mass flow rate at the compressor inlet can be performed by using IGV angle variation.
- the flow at the turbine inlet is considered in a choking condition, thus making possible the use of the following equation:

\[ \frac{M_i \sqrt{T_i}}{P_i} = \text{const.} \tag{7} \]

**INFLUENCE OF THE VARIOUS DYNAMIC PHENOMENA**

The simulations considered in this study are relative to the case of a single shaft industrial gas turbine working in parallel with electrical mains. As a consequence, the torque offered by the electric generator to the gas turbine adapts almost instantaneously to the torque delivered by the machine, keeping the gas turbine rotational speed almost equal to the synchronism speed. Therefore, the first term of eq. (1) is always very close to zero.

The schematic lay-out of the gas turbine considered is shown in Fig. 1 and in Table 1 the main relative features in ISO design condition.

![Fig. 1: Gas turbine schematic lay-out.](image)

**Table 1: Gas turbine main features.**

<table>
<thead>
<tr>
<th>Pe [MW]</th>
<th>5.4</th>
<th>Bc</th>
<th>9.2</th>
</tr>
</thead>
<tbody>
<tr>
<td>T_H [K]</td>
<td>1228</td>
<td>M_01 [kg/s]</td>
<td>24.6</td>
</tr>
<tr>
<td>T_TO [K]</td>
<td>785</td>
<td>M_F [kg/s]</td>
<td>0.3923</td>
</tr>
</tbody>
</table>

The analysis of the influence of the various dynamic phenomena on the gas turbine transient response was performed considering, in the first phase, the effect of each single phenomenon evaluated separately and in the absence of the gas turbine control system. In particular, the influence of dynamic phenomena due to the working gas mass contained in the gas turbine component volumes was evaluated using the MOD. 1. MOD. 2. was instead used to evaluate the influence of thermal inertia of the metal mass and of the heat transfers.

Once the most significant phenomena were determined, the machine behaviour was evaluated considering these remarkable phenomena in the case in which the gas turbine power plant is equipped with a temperature control system at the gas turbine discharge. More in detail, consideration was given to gas turbine working conditions relative to constant temperature at the gas turbine outlet (type of control generally employed when the gas turbine works in combined cycle or cogeneration plants) and to outlet temperature varying with an assigned law to maintain a constant fire temperature.

**Transient responses of the gas turbine without control system**

The first step was to compare the gas turbine responses of the two models, when no dynamic
phenomena was considered (static response) and for the same variation of each input variable. In particular, the fuel valve position and the IGV angle were taken into account as input variables.

Once it was verified that the static responses obtained by means of the two models were similar, the effect of each single phenomenon was evaluated separately.

As mentioned before, the phenomena related to working gas mass contained in the gas turbine component volumes were considered using the MOD. 1 (Bettocchi et al., 1996). The phenomena related to thermal inertia of the metal mass and to heat transfers were evaluated using the MOD. 2 (Framatome Technologies, 1996).

In both cases of static and dynamic simulations, the following variations in fuel valve position and IGV angle were performed:

- step variation of fuel valve position with a constant IGV angle;
- step reduction of IGV angle with a constant fuel valve position.

Fig. 2 shows the $M_f$ variations due to a step closing of fuel valve from 100% to 50% of full opening position.

Fig. 3 shows the IGV variation from the full opening position (IGV=1) to the full closing position (IGV=0). In the same figure, the fuel mass flow rate variation resulting from IGV angle reduction is shown.

The fuel mass flow rate is not constant since a reduction of compressor inlet mass flow rate determines a reduction of the pressure at the compressor outlet (considering eq. 7). This means an increase in the pressure drop across the fuel valve and finally, an increase in the fuel mass flow rate.

**Results**

In order to represent the gas turbine dynamic response, the parameters that were considered significant for this study are:

- shaft power, $P_{sh}$;
- turbine inlet temperature, $T_{it}$;
- turbine outlet temperature, $T_{ot}$;
- compressor outlet temperature, $T_{oc}$;
- compressor outlet pressure, $p_{oc}$;
- gas mass flow rate at the turbine outlet, $M_{ot}$.

It should be noted that, as previously mentioned, the shaft speed was not considered since, in the case of a gas turbine working in parallel with electrical mains, the rotational speed is practically constant.

The reported parameters are instead of greater relevance because they can be used by the control system, and are of particular interest in the case of gas turbine in cogenerative or combined cycle application.

As mentioned before, the first comparison made was between the static responses of the two gas turbine models considered (MOD. 1 and MOD. 2).

Table II shows the Mean Square Deviation (MSD) on the main variables due to $M_f$ and IGV step variations in Figures 2 and 3 respectively. The

<table>
<thead>
<tr>
<th></th>
<th>MSD for $M_f$ variation</th>
<th>MSD for IGV variation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{sh}$</td>
<td>$7.3x10^{-3}$</td>
<td>$1.9x10^{-5}$</td>
</tr>
<tr>
<td>$T_{it}$</td>
<td>$4.0x10^{-4}$</td>
<td>$3.7x10^{-5}$</td>
</tr>
<tr>
<td>$T_{ot}$</td>
<td>$3.4x10^{-5}$</td>
<td>$1.2x10^{-5}$</td>
</tr>
<tr>
<td>$T_{oc}$</td>
<td>$1.3x10^{-5}$</td>
<td>$9.3x10^{-7}$</td>
</tr>
<tr>
<td>$p_{oc}$</td>
<td>$2.0x10^{-6}$</td>
<td>$1.3x10^{-6}$</td>
</tr>
<tr>
<td>$M_{ot}$</td>
<td>$2.5x10^{-7}$</td>
<td>$6.0x10^{-7}$</td>
</tr>
</tbody>
</table>
maximum MSD value is relative to the Psh variable and is mainly due to the different cooled expansion models adopted by MOD. 1 and MOD. 2 (see also MSD on T1d).

At this point, the static and dynamic responses of each model were analyzed and compared separately. In this way, it is possible to take into account each dynamic phenomenon separately, avoiding errors due to differences in response between the MOD. 1 and MOD. 2.

Table III shows the Mean Square Deviation between the static and dynamic responses obtained using MOD. 1 for the Mf and IGV step variations in Figures 2 and 3 respectively.

The analysis of the values in Table III shows a negligible influence of the dynamic phenomena due to the working gas mass contained in the gas turbine component volumes.

This result is valid in the particular analyzed case of a single-shaft gas turbine working in parallel with electrical mains. Different results would be obtained, for example, in the cases of two-shaft gas turbines with rotational speed controller (Blotenberg, 1993) or of gas turbine aircraft engines (DeHoff and Hall, 1978; Szuch, 1978). In these cases, the dynamic phenomena due to the working gas mass contained in the gas turbine component volumes affect the rotational speed transient response.

Table III: MSD between the MOD. 1 static and dynamic responses.

<table>
<thead>
<tr>
<th>MSD for Mf variation</th>
<th>MSD for IGV variation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Psh</td>
<td>3.0x10^-5</td>
</tr>
<tr>
<td>Tft</td>
<td>1.8x10^-7</td>
</tr>
<tr>
<td>Toc</td>
<td>4.9x10^-7</td>
</tr>
<tr>
<td>Tot</td>
<td>7.8x10^-7</td>
</tr>
<tr>
<td>Poc</td>
<td>6.1x10^-6</td>
</tr>
<tr>
<td>Moc</td>
<td>8.5x10^-6</td>
</tr>
</tbody>
</table>

Concerning the static and dynamic responses obtained using MOD. 2, the effect of the thermal inertia of the metal mass and of heat transfers are significant only on the compressor and turbine outlet temperatures. Figures 4 and 5 show the static and dynamic responses on Toc and Tot due to the Mf and IGV step variations in Figures 2 and 3 respectively.

The dynamic response on turbine outlet temperature should be especially taken into account since this temperature is often used by the gas turbine control system.
Transient response of the gas turbine with control system

The influence of the most significant dynamic phenomena (thermal inertia of the metal mass and heat transfers) was evaluated in the case of presence of turbine outlet temperature control system, using MOD. 2.

Two different control logics were adopted. The goal of the first one (logic A) is to maintain a constant temperature at the gas turbine outlet (type of control generally used for gas turbine in combined cycle or cogeneration application).

The aim of the second one (logic B) is to maintain the turbine inlet temperature at a constant value, imposing that the turbine outlet temperature follows a variation law that links the values of turbine outlet temperature and compressor outlet pressure.

In both cases, the control system is a PID controller (Fig.1) that adjusts the IGV angle on the basis of the difference between the turbine outlet temperature and the imposed set-point.

Results

For both the control logics analyzed, the following parameters were reported:
- inlet guide vane position, IGV;
- fuel mass flow rate, M_f;
- turbine inlet temperature, T_{it};
- turbine outlet temperature, T_{ot};
- shaft power, P_{sh};
- gas mass flow rate at the turbine outlet, M_{ot}.

The simulations are relative to the step variation of fuel valve position from full opening to about 90% of full opening position.

Logic A (Constant turbine outlet temperature)

Fig. 6 shows the IGV angle curve obtained by means of the first control logic, along with the fuel mass flow rate curve due to the imposed fuel valve position variation.

Fig.7 shows the T_{it} and T_{ot} curves; note that the control logic A provides a constant value of turbine outlet temperature.

Fig.8 shows the P_{sh} M_{ot} curves obtained by means of the control logic A.

It should be observed that, even if a step reduction is made of the fuel valve position from full opening, the fuel mass flow rate trend is not a step (Fig.6). This is essentially due to the fact that the PID controller introduces a delay in the IGV angle adjustment and, as a consequence, in the gas turbine inlet mass flow rate (Fig.8).
**Logic B (Constant turbine inlet temperature)**

Fig. 9 shows the IGV angle curve obtained by means of the second control logic, along with the fuel mass flow rate curve due to the imposed fuel valve position variation.

Fig. 10 shows the $T_{it}$ and $T_{ot}$ curves; note that the control logic B provides a constant value of turbine inlet temperature.

Figure 10 reveals that the gas turbine outlet temperature finishes its transient at a value greater than that in steady state. This derives from the fact that, if the $T_{it}$ is set to be constant, a step variation of fuel valve position from full opening leads to a reduction of compressor inlet mass flow rate and consequently of compressor outlet pressure (considering eq. 7). Therefore, the expansion starts from a lower pressure than in steady state and thus finishes at a greater temperature.

Fig. 11 shows the $P_{sh}$ $M_{ot}$ curves obtained by means of the control logic B.
CONCLUSIONS

The analysis developed in the paper allowed evaluating the influence of the various dynamic phenomena on the transient response of a single shaft gas turbine working in parallel with electrical mains and consequently at constant rotational speed.

This analysis was carried out using two different gas turbine dynamic models: the first one considering the dynamic phenomenon effect of gas masses contained in the gas turbine components and the second, the effect of metal mass thermal inertia and heat transfers.

The simulations were performed using step variations both in fuel valve and IGV position.

It resulted that, for the gas turbine considered, the response is not significantly affected by dynamic phenomena due to the gas masses contained in volumes. Concerning the thermal inertia and heat transfers, these only affect particularly the response in temperatures. More in detail, for the kind of gas turbine analyzed, the turbine outlet temperature and compressor outlet temperature arrive to the transient end after about 300 seconds.

Since the turbine outlet temperature is used in feedback by the gas turbine control system, the delay in this temperature should be taken into account in the gas control system setup phase.

The analysis was then developed considering the presence of the gas turbine control system. In particular, the behaviour of the gas turbine with two different logics was studied taking into account the effect of the inertia metal mass and heat transfers.

The first control logic maintains constant the turbine outlet temperature, while the second maintains constant the turbine inlet temperature.

The simulations were performed using a step closing of fuel valve while the IGV position is regulated by the PID controller.

It resulted that, for the two control logics, the turbine outlet temperature and turbine inlet temperature arrive at the constant value, in the transient end, after about 15 seconds and about 30 seconds respectively. This highlights the relevance of control logic and of PID constants on the overall gas turbine transient response.

Finally, it should be observed that this analysis permitted to determine the most important dynamic phenomena that should be taken into account in the particular case of gas turbine working in parallel with electrical mains. Furthermore, these non linear dynamic models may be used:

- to analyze the transient response of different gas turbine plants;
- to setup the machine control system;
- to compare different control systems;
- to generate time series of transient condition data, necessary to setup, by means of system identification techniques, models used for the diagnosis of gas turbine sensors.

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