Abstract. – This paper concerns the set-up of a Finite Element axisymmetric model of the 701F whole machine, i.e. the ensemble of the whole axis line (the compressor/turbine group and the ABB alternator) and the casing. The model has been used for computation of critical speeds and harmonic responses to given unbalances, in order to evaluate the clearance variation between rotor and casing at some critical location. Another aim of the work was to investigate the influence of the casing on the dynamical behaviour of the machine, comparing the critical speeds computed for the rotor alone, for the whole machine and the experimental measured ones.

INTRODUCTION

During the last years the market of heavy duty gas turbines for power generation has been targeted towards increased power outputs and improved performance. The development and application of larger sizes with better thermal efficiencies implies that the engines parts must satisfy more severe and challenging requirements in terms of life and reliability.

Therefore there is a strong need of mathematical models which use is an essential tool all over the product life, from design phase to service.

For the industrial gas turbine applications the development of rotor-dynamic models is finalised to represent the engine and its interactions with the other components of the shafting line either at design or at off-design conditions; to support the definition of the project, the set up during commissioning, and the diagnosis in case of problems during service.

PURPOSE OF THE STUDY

• To build an axisymmetric model of the gas turbine rotor and casing joined with the generator to represent the whole shaft line.

• To calculate the critical speeds and compare them with the experimental data.

• To plot and analyse the mode shapes of the first critical speeds.
To do sensitivity analysis of the harmonic response to residual unbalance on the turbine disks.

To analyse the vibration response of the rotor to balancing weights put on the balancing planes of the gas turbine. The outputs, plotted as polar diagrams for each balancing plane, provide the so-called “Effect Diagrams” used for the balancing activity.

To investigate the effect of gas turbine casing modelling over the rotor dynamic behaviour.

To investigate the relative displacement between the gas turbine rotor and casing.

**MACHINE DESCRIPTION**

The longitudinal section of the gas turbine is shown in the figure. The main characteristics are:

- Total mass: 330000 [kg]
- Total length: 17.3 [m]

The generator drive flange is at the cold end of the rotor.

The horizontal split casings are connected to the ground by means of two fixed supports at the compressor inlet and by means of two hinged (to allow thermal growth) supports at the turbine end.

The bearing casings are joined to the casings by means of radial struts at the compressor end and by means of tangential struts at the turbine end.

**ROTOR:**

- Speed: 3000 [rpm]
- Total mass: 89000 [kg]
- Total length: 11.6 [m]
- Journal bearing centre lines span: 8.91 [m]
- Maximum diameter (bladed): 3.25 [m]

The rotor (see fig. 1) is built by the compressor and the turbine sections joined one to the other by means of an intermediate shaft.

The rotor is sustained by means of 2 radial journal bearings with tilting pads located at the rotor ends. The axial position of the rotor and the reaction to the residual axial thrust is ensured by one thrust bearing with tilting pads located at the compressor side end.
COMPRESSOR CONFIGURATION:

- 17 stages plus 1 Variable Inlet Guide Vane (axial flow)
- Stacked disks with tension bolts.

TURBINE CONFIGURATION:

- 4 stages (axial flow, reaction type)
- Stacked disks with Curvic coupling and tension bolts.

Particular assembly procedures are enforced in order to minimise the residual unbalance of the rotor during the assembly phase. At site, if needed, for the balancing purpose of the gas turbine and of the shaft line, weights can be applied in three sections (see also figure): first compressor disk, intermediate shaft and last turbine disk.
SHAFT LINE DESCRIPTION

The present study refers to a shaft line layout described by the figure below. The Generator is joined to the gas turbine by means of an intermediate shaft. The rotor is kept in place by means of 1 thrust bearing and 5 journal bearings. 3 on the generator and 2 on the gas turbine.
In the field the absolute movement of the rotor is monitored by means of: devices capable of:

- seismic measurement of the casing
- relative displacement of the rotor with respect to the casing.

Typically these measuring devices are installed over each one journal bearings of the shafting line which can provide information about the journal position, amplitude and phase of the vibration. These are the locations where it is possible to make direct comparison between experimental and computed behaviour.

**MODELS AND ASSUMPTIONS**

**STRUCTURE**

Both rotating parts and casing have been modelled, together with bearings, struts and supports.

SAMCEF allows to represent the rotating parts by two different kinds of models:

- A beam model, in which disks are represented as lumped masses. This kind of model doesn’t take in account disk stiffness, and is well suited especially for preliminary analyses.

- An axisymmetrical model, in which the displacement field is developed in Fourier series. The model must be reduced to a superelement by component modes synthesis – via the DYNAM module – and then it can be passed to the ROTOR module for rotor dynamic analysis.

Only the axisymmetrical model has been considered in this work. The 701F turbine/compressor group and the alternator have been modelled together, so dealing with the whole axis line.

The casing, too, has been represented by a DYNAM created superelement, obviously considered as non-rotating.

Analyses have been carried in a speed range between 0 and 3000 RPM (50 Hz).

Figures 2 and 3 show the models as well as the lumped masses applied to the alternator.
SAMCEF

Rotor + stator model

Geometric scale
1000.

Numerical scale
1/1205.879516

Fig. 2

SAMCEF

Model details

Geometric scale
1000.

Numerical scale
1/556.062622

Turbine & compressor group

Fig. 3
In this kind of model, a special attention must be paid to the blade zones, in which an equivalent density must be used to keep into account the circumferential discontinuity. For each blade stage the equivalent density has been computed as follows:

a) Generation of the axisymmetric volume by a simple revolution of the blade section;

b) Computation of the axisymmetric theoretical weight \( g_{ax} \);

c) The equivalent density is given by the relation:

\[
\rho_{eq} = \frac{g}{g_{ax}} \rho
\]

where:

\( \rho \) = real material density

\( g \) = real blade stage weight.

**BEARINGS**

Each bearing, strut and external support of this machine can be considered – in each direction – as a spring/damper link between two nodes.

The following links have been represented:

- alternator bearings oil films;
- alternator supports to ground;
- TG bearings oil films;
- TG struts between bearings and casing;
- TG casing supports to ground.

Some of the stiffness and damping coefficient values are shown in figures 4 and 5.

Considerations about the first part (horizontal) of the curves will be done in results analysis.
Fig. 4

SAMCEF - BACON : V 7.1-3
701F turbogas - whole axis line with casing
Compressor side casing support damping

Fig. 5

SAMCEF - BACON : V 7.1-3
701F turbogas - whole axis line with casing
Turbine side casing support damping
LOADS

Only unbalance loads have been considered.

Three unbalances have been separately loaded at three different locations in the rotor:

- 300 gcm at the 1st turbine stage;
- 300 gcm at the coupling flange;
- 300 gcm at the middle of the alternator.

These loads are supposed to act separately, i.e. a single analysis has been performed for each unbalance.

Fig. 6 shows the location of the unbalances.

Fig. 6
RESULTS

CRITICAL SPEEDS

Fig. 7 shows the Campbell’s diagram of the whole structure, i.e. of the assembly between rotor and casing.

Fig. 7

Fig. 8 ÷ 10 show the deformed shapes at the first three critical speeds.

Fig. 8
Fig. 9

Fig. 10
The following table lists the critical speeds (RPM) computed by ROTOR – both for the rotor alone and for the whole machine – compared to experimental values.

<table>
<thead>
<tr>
<th></th>
<th>Computed (rotor alone)</th>
<th>Computed (whole machine)</th>
<th>Experimental</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>720</td>
<td>700 ÷ 750</td>
<td>700 ÷ 750</td>
</tr>
<tr>
<td>993</td>
<td>890</td>
<td>790 ÷ 900</td>
<td></td>
</tr>
<tr>
<td>1079</td>
<td>1450</td>
<td>1280 ÷ 1350</td>
<td></td>
</tr>
<tr>
<td>1376</td>
<td>1680</td>
<td>1700 ÷ 1800</td>
<td></td>
</tr>
<tr>
<td>2581</td>
<td>2200</td>
<td>2300 ÷ 2400</td>
<td></td>
</tr>
</tbody>
</table>

The following considerations can be done:

a) The influence of gyroscopic effect on critical speeds seems to be quite small, as shown by the Campbell lines which are almost horizontal;

b) The two first critical speeds (about 100 an about 500 RPM, see Campbell diagram) have to be carefully considered, because the stiffness and damping curves of TG oil films are not defined at very low speeds; the values are supposed to be constant from 0 to 700 RPM (see figs. 4 and 5), but this assumption seems to be quite arbitrary.

c) The influence of the casing is unnegligible: the first critical speed, which is usually very important, is not shown by the rotor alone, and all other critical speeds are better identified by the whole model with considerable differences with respect to the rotor alone ones.

UNBALANCES

The aim of this part of the study was to investigate the effect of unbalances located at different points of the shaft line (see fig. 6), taking in account the presence of the casing and the related supports to ground. Useful information about the relative radial displacements of a rotor point and the related casing point, as well as about the relative phases, can arise from this simulation.

Measurement points are the casing supports, while the machine locations supposed to be critical are the 17th compressor stage and the 1st turbine stage (see fig. 11).
Fig. 11

Fig. 12 and 13 show the displacements induced by the turbine unbalance, while figs. 14 up to 16 show some of the results of the analysis, in terms of clearance modifications.

Fig. 12
17° compressor stage and measurement points - radial displacements

Fig. 13

Radial clearance variation

Fig. 14 – clearance variation with alternator unbalance
The 900 RPM critical speed is only excited by the alternator unbalance, while the remaining ones concern the whole axis line.

The alternator unbalance induces a maximum displacement of about 1.4E-4 mm.
The flange unbalance induces a maximum displacement of about 1.E-3 mm.  
The turbine unbalance induces a maximum displacement of about 6.E-4 mm.  
The maximum clearance reduction (2.38E-4 mm) is achieved in correspondence of 
the compressor bearing for the condition of flange unbalance at 12 Hz (720 RPM),
i.e. the first critical speed.  
This value of clearance reduction seems to be reassuring about the risk of rubbing 
between turbine and stator, at least for the considered amount of unbalances.