ROTORDYNAMICS OF GAS TURBINES USING THE FINITE ELEMENT METHOD

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Abstract. *One of the main requirements in the assessment of integrity of gas turbines is the correct evaluation of its dynamic behavior. The natural frequencies which give the critical speeds must be determined. The complexity of the turbine design and dynamic motion has forced the use of more sophisticated tools such as the Finite Element Method. Computer programs as ANSYS have allowed dynamic analysis of more complex rotating machinery. A preliminary rotor model has been created and evaluated in two distinct environments associated with the software ANSYS. Considering a two degrees of freedom model, in the first case, the APDL environment has been explored. For the second case, the same model has been examined using the "Structural" and "Modal" tools of the Workbench in ANSYS. This study revealed the possibility of different approaches for obtaining the same results. This analysis have also been compared with numerical solutions of an analytical model for the same case. Advantages and deficiencies of these computational tools will be discussed and criticized.*

Keywords*: Rotordynamics; Finite Element Method; Gas Turbine*

1. INTRODUCTION

Mechanical failures in the world of gas turbine result in high costs and can cause serious accidents, so these events should be prevented at early stages of design. Therefore, a great number of investigations have to be performed as for example analysis of structural and thermal tensions, developing of new materials and cooling techniques, studies of vibration responses of all components, and others. For the purpose of illustrating the importance of developing the correct model and analysis of mechanical vibrations, in the diagnostic of failure of a certain industrial gas turbine, carried out by Farrahi *et al.* (2010), the main conclusion is that the probable source for this failure was the increasing of the vibration amplitude of the rotor near the second natural frequency causing the initiation of multiple cracks in the interface of the disks and shaft by the fretting fatigue.

For the study of mechanical vibration a complete rotordynamics analysis is fundamental, and a key method for this type of analysis is the Finite Element Method (FEM). Recently, Hutton (2004) brought a very detailed approach of this method, discussing not only the application in rotordynamics, but many other areas of interest.

The FEM has been studied for many years. According to Chiang *et al*. (2004), in the beginning of 1970 only the bending vibration for a linear displacement were considered. However, during the last two decades several researches have been applying this method for modeling rotor systems. Lalanne and Ferraris (2001) have worked on very detailed analyses of tipical rotordynamics problems, developing equations for each element of the rotor (disks, shaft and bearings) using the numerical method of Rayleigh-Ritz and the Lagrangian equations. Based on this study the authors Chiang *et al*. (2004), cited previouly, use a model with single and dual rotor system through an inertial coordinate system, with gyroscopic moment, rotary inertias, bending and shear deformations, axial loads, and internal and external damping, obtaining satisfactory results.

More recently, Young *et al*. (2007) investigate the behavior of random axial forces in the lateral vibration in a disk-shaft system supported by a pair of ball bearing, featuring the dynamic stability of this system. Making use of computational tools, more realistic dynamic models have been created.

The Finite Element Method can be implemented according to several aproachs. Creci *et al.* (2009) used the Ansys® to analyze a single spool gas turbine modeled with beam elements, which obtained the whril speed, unbalance response, modal orbits, Campbell diagram and trasient analysis. Mingjian Lu and Baisong Yang (2010) also used Ansys® creating a 3D solid model which allowed the analysis of the depedence of rotor natural frequency on road preload and vibration modes. One may notice that Ansys® applicatives offer different alternatives for the solution of rotordynamic problems. In order to investigate the applicability of this software in rotordynamics, in the present work, the Parametric Design Language (APDL) and the Workbench environments were visited.

Given the wide range of modeling possibilities, the present work has the purpose of exploring some of the most important functions of these computational tools to developed the rotor model. Campbell diagrams are plotted to find natural frequencies and critical speeds, and the results are compared. Moreover, the computational time spent for each case also is also reported.

2. MODEL

The rotor system studied is the same analyzed by Lalanne and Ferraris (2001), which consists in a single rotor-bearing system, with three rigid disk and two linear bearing supports as show in Fig. 1. The authors use the pseudo-modal method and compare results with the direct method.

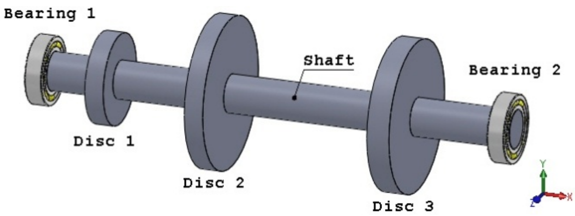


Figure 1: Rotor model.

In this model, the disk and shaft are made of steel with Young´s modulus of 200 GPa, Poisson´s ratio of 0.3, and density of 7800 kg/m^3. Table 1 shows the main geometric properties of the rotor-discs and Tab. 2 shows the technical features of the two identical bearings. The shaft cross-section radius is 0.05 m.

Table 1. Disk data

	Disc 1	Disc 2	Disc 3
Thickness (m)	0.05	0.05	0.06
Inner radius (m)	0.05	0.05	0.05
Outer radius (m)	0.12	0.2	0.2

Table 2. Technical features of the bearings

Stiffness	Damping
$K_{zz} = 5x10^7$ N/m	$C_{zz} = 5x10^2$ Ns/m
$K_{yy} = 7x10^7$ N/m	$C_{yy} = 7x10^2$ Ns/m
$K_{zy} = K_{yz} = 0$	$C_{zy} = C_{yz} = 0$

As previously stated, the present work uses the finite element method implemented by the software Ansys® and compared with the results obtained by Lalanne and Ferraris (2001). Analysis of the model was divided into three cases which are identified below.

Case I: Direct generator of rotor-bearing

The direct generator is one method of generating the model in APDL environment. In this type of resolution it is necessary to determine the location of every node and all other parameters of elements such: size, shape and connectivity. Figure 2 illustrates the model configuration for Case I. The elements types chosen for this model are Beam189, Pipe16 and Combi214.

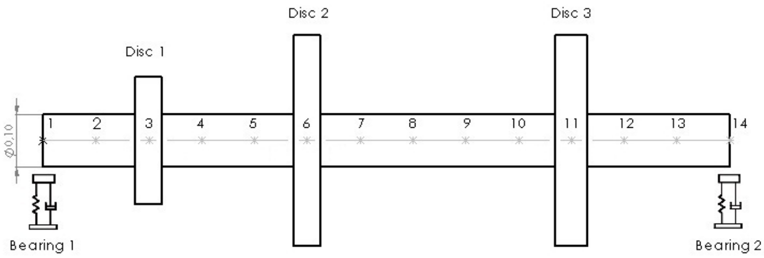


Figure 2. Rotor model using beam elements.

The shaft is modeled using 13 Beam189 elements. This element is based on Timoshenko beam theory considering six degrees of freedom at each node, translation in the x,y and z direction and rotation in x, y and z direction. The rotor has three discs modeled with 6 Pipe16 elements. Finally, to model the bearings, 2 Combi214 elements are used, considering stiffness (K_{11}, K_{22} , K_{12} and K_{21}) and/or damping characteristics (C_{11}, C_{22} , C_{12} and C_{21}) in 2-D applications, as can be seen in Fig. 3.

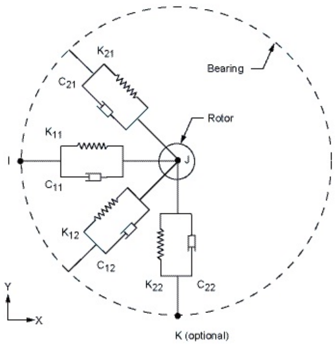


Figure 3. Combi214 element used for the bearings modeling.

Case II: Solid rotor-bearing.

Solid modeling is another possibility for generating the model in APDL. This method requires a geometric boundaries description of the model and a control over the size and desired shape of elements. After defining the geometry and mesh size, all nodes and elements are generated automatically.

In this case the rotor is modeled with Solid273 elements which are used to model axisymmetric solid structures. This type of element is defined by eight nodes on the master plane and nodes created automatically in the circumferential direction based on the eight master plane nodes.

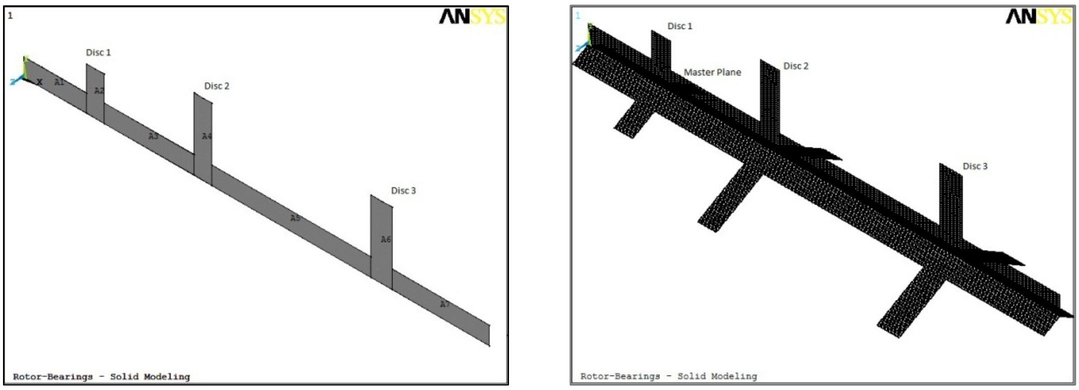
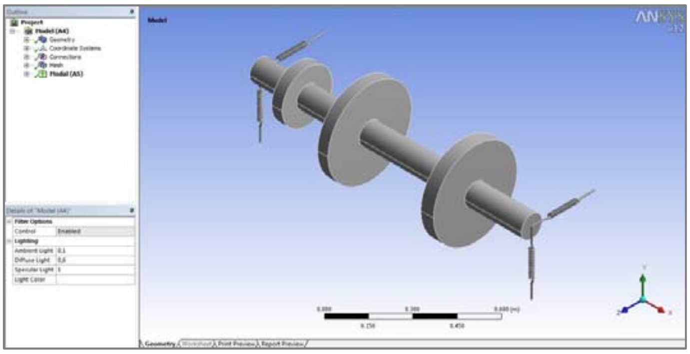


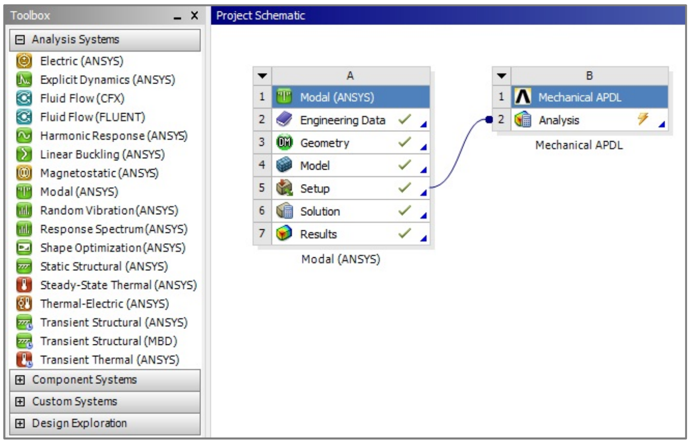
Figure 4. Solid model developed in APDL; (a) Construction of model; (b) Model Mesh

Case III: Solid rotor-bearing imported from CAD software.

The drawing is created in a CAD software and imported to Ansys ® Workbench. In this environment the model setup can be accomplished. The bearing stiffness and damping of the bearing, as well as, displacement constraints are defined, as shown in Fig. 5a. Taking this model, including all boundary conditions discussed before, a preliminary modal analysis has been performed in Workbench and the results were transferred to APDL to obtain the corresponding Campbell diagram and other results as illustrated in Fig. 5b.



(a)



(b)

Figure 5. (a) Solid model imported by CAD software; (b) Transferring the model to the APDL

The elements are created automatically when the model is transferred from Workbench to APDL. For this model the elements are Solid187, Combin14, Conta174 and Targe170. The first element is used to model the rotor as it considers a quadratic interpolation function for the displacement behavior and is appropriate to model irregular meshes produced from CAD software. The bearings are modeled with Combin14 elements that are uniaxial tension-compression elements with up to three degrees of freedom at each node, considering only translations in the directions x, y and z. Conta174 and Targe170 are used to make connection between parts of the rotor.

2.1. Rotor-Bearing analysis

In the case of rotating machinery, for the successful design and operation, is important to determine the natural frequencies as a function of the speed of rotation, this relationship is obtained through the Campbell diagram. Other important parameter is to determine the critical speeds that may vary with the type of excitation frequencies. According to Creci *et al.* (2009) the most important is caused by the mass unbalance of the rotor that given origin to excitation frequency ω. Others excitation are due the coupling misalignments in $2{*}\omega$ and aerodynamics stimulus in $0.5{*}\omega$.

Therefore, for the three cases described before, the Campbell diagram were obtained to predict the natural frequencies and the critical speeds caused by the mass unbalance of 200 g.mm situated on Disc 2. The speed range was set from o to 30000 rpm. A harmonic analysis was also performed to compare the cases.

All analyses were performed in the APDL environment in a computer processor Intel Core 2 Duo 2.4 GHz with 4GB of RAM.

3. RESULTS AND DISCUSSIONS

Figure 6 shows the Campbell diagram with the first seven bending modes and Tab. 3 reveals the results of frequencies for each case obtained at 25000 rpm. It can be observed that the results of the first case are the closest to the reference model results with maximum difference of 0.4%. However, with a maximum difference of 3.9%, the others cases may also be considered acceptable.

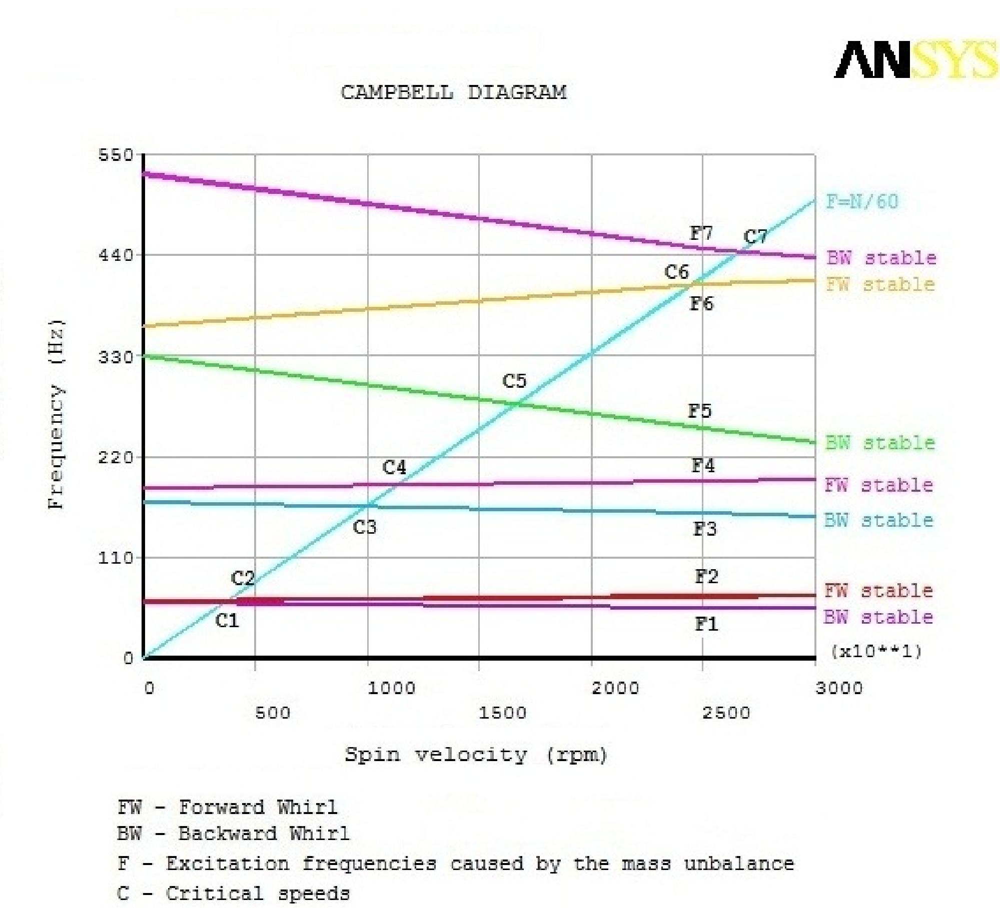


Figure 6. Campbell Diagram.

Table 3. Frequencies in hertz at 25000 rpm.

Frequency	Reference Value	Case I	Case II	Case III
F1	55.41	55.417 0%	55.973 1.0%	56.767 2.4%
F2	67.2	67.203 0%	66.955 -0.4%	68.643 2.1%
F3	157.9	157, 861 0%	151.76 -3.9%	159.619 1.1%
F4	193.6	193.577 0%	190.235 -1.7%	198.593 2.6%
F5	249.9	250.835 0.4%	259.556 3.9%	244.064 -2.3%
F6	407.5	408.165 0.2%	401.591 -1.5%	418.026 2.6%
F7	446.7	446.807 0%	444.428 -0.5%	456.813 2.3%

The accuracy of the frequencies results, error rate less than 5%, evidence that the Campbell diagram is being developed properly by the proposed models. Thus, it is possible to find the critical speeds according to each excitation type. Table 4 shows the critical speeds caused by the mass unbalance. As for the frequencies values, the direct generator had an advantage in the results obtained for the critical speeds. The first case had an average difference of 0.7% with a maximum of 2%, the second had an average of 1.1% and maximum of 3.8% and the third had average of 2% and maximum of 3.3%.

This mapping performed is very important to ensure that the operating speed range is situate in a safe region and/or define which regions are safe for the rotor spin.

Table 4. Critical speeds in RPM caused by the mass unbalance.

Critical Speed	Reference Value	Case I	Case II	Case III
C1	3620.4	3591.68 -0.8%	3606.73 -0.4%	3671.92 1.4%
C2	3798.1	3819.31 0.6%	3829.74 0.8%	3911.63 3.0%
C3	10018	9819.31 -2.0%	9635.04 -3.8%	10052.56 0.3%
C4	11279	11349.43 0.6%	11122.68 -1.4%	11649.92 3.3%
C5	16785	16631.90 -0.9%	16793.64 0.1%	16977.86 1.1%
C6	24408	24423.85 0.1%	24605.32 0.8%	25086.37 2.8%
C7	26615	26636.45 0.1%	26495.98 -0.4%	27164.98 2.1%

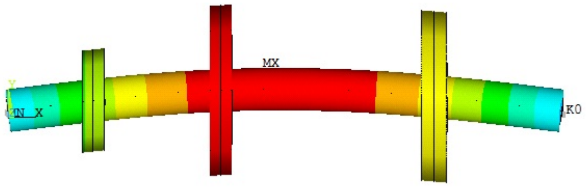
It is possible to observe that the frequencies and critical speeds results obtained with the three cases are satisfactory. Thus, other parameters can be analyzed to try to determine which model is the most advantageous. Table 5 shows the general features for each case such as node number, elements number and computational time to develop the “Campbell diagram” by modal analysis and develop the “Mass unbalance response” by harmonic analysis. The first case prevailed and proved to be more effective, because the computational time spent in both analyses (modal and harmonic) was much smaller than the other cases, principally in the harmonic analysis.

Besides the computational time, the difficulty to develop each case can be other important and decisive issue for choosing the kind of model. The direct model which revealed good results, can be easily implemented for a simple project like this, but for a more complex model the expert labor time demand may became the most critical issue.

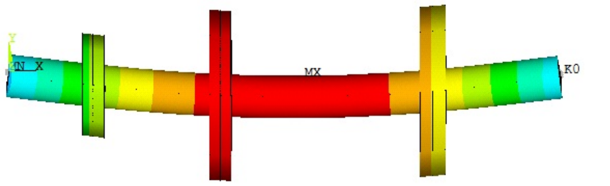
Table 5. General features of each case for the mapping Campbell diagram.

	Case I	Case II	Case III
Nodes	22	31627	28951
Elements	21	3402	14760
Calculation time (s) “Modal”	5	523	255
Calculation time (s) “Harmonic”	182	34920	32323

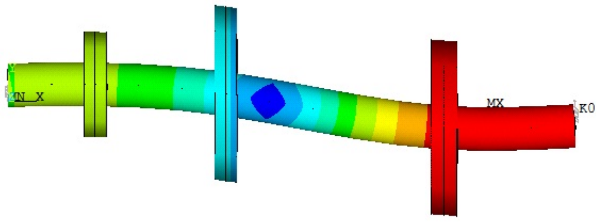
The harmonic analysis allowed the representation of the bending modes and modal orbits occurring along the shaft. Figure 7 illustrates the six most fundamental bending modes of the rotor. Figure 8 shows the corresponding modal orbits for these six modes. For this analysis the direct generator has been used due to the reduced computational time.



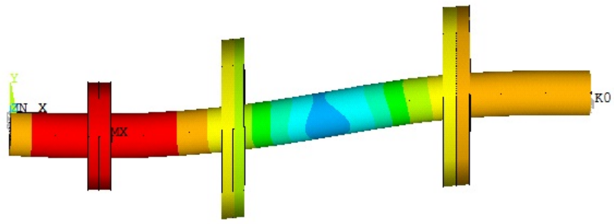
a) The first mode



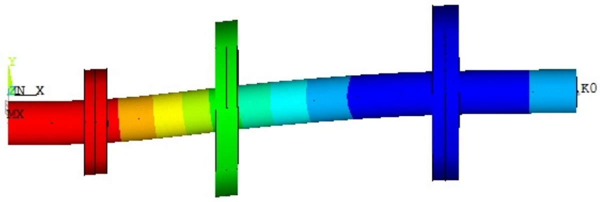
b) The second mode



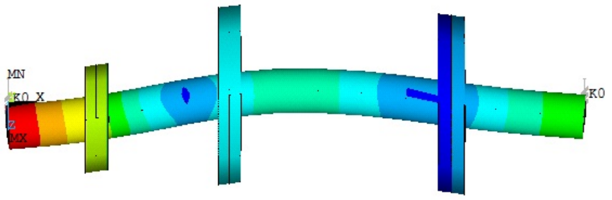
c) The third mode



d) The fourth mode

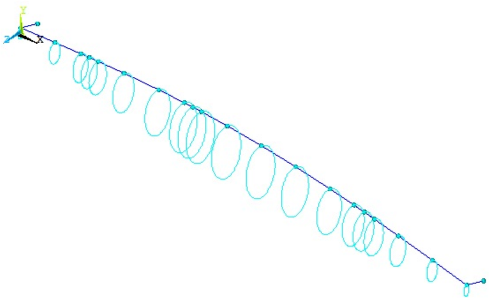


e) The fifth mode

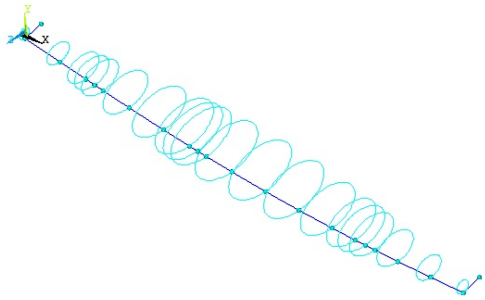


f) The sixth mode

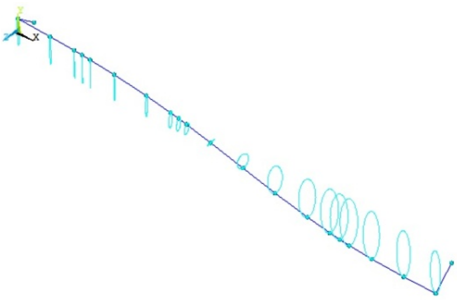
Figure 7. The first six bending modes of the disc rotor using the direct generator.



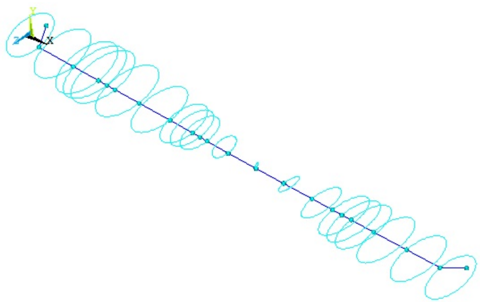
g) The first modal orbit



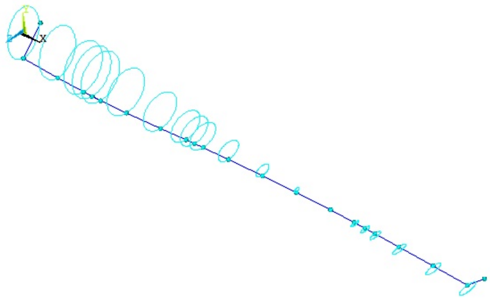
h) The second modal orbit



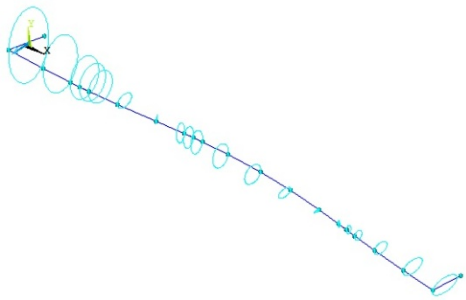
i) The third modal orbit



j) The fourth modal orbit



k) The fifth modal orbit



l) The sixth modal orbit

Figure 8. The first six modal orbits using the direct generator.

4. CONCLUSION

A rotordynamic model was successfully implemented using the Finite Element Method considering three different approaches. According to the literature review, this method has been increasingly adopted for the more complex models solutions. For the present cases, solutions were obtained using the commercial software Ansys® which considers the rotatory effects and gyroscopic inertia. Comparison with a published and well known reference case revealed the accuracy of the model and very close agreement has been observed. Three different cases were considered to explore the several tools offered by the Ansys® environment. One can observe that the results obtained considering the beam elements has shown the better agreement with the Lalanne's reference case. One also has to consider that Lalanne's case was based on a beam element model.

However, with an error of less than 5%, results obtained from the other two cases can be considered very satisfactory.

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