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Study of The dynamics of rotor-bearing using MATLAB and finite elements

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ABSTRACT

The dynamics of the rotors and the stability of the rotating machines, It plays an important role in improving the security and performance of these systems.
Two phenomena in rotor dynamics are particularly dangerous and can lead to unacceptable vibration levels, These are critical rotational speeds and linearly unstable regimes, the consequences of which are often catastrophic.The internal material damping in the rotor shaft introduces rotary dissipative forces which are proportional to spin speed and acts tangential to the rotor orbit. These forces influence the dynamic behaviour of a rotor and tend to destabilize the rotor shaft system as spin speed increases.So, In this paper in the study of the dynamic behavior of rotors, we are interested in:

- study of Hydrodynamic bearing effect on the rotor dynamics.
- Determination of the eigenvalues of the rotor as a function of its rotational speed and diagrams of : Campbell, stability, constant damping....

Keywords:dynamics, rotor, stability, critical rotational speeds, gyroscopic forces.

I. Introduction

Vibration control of turbo machinery is very important for the integrity of industrial plants. In this regard it is very important to predict the dynamic behavior of rotating machinery which operates above the first critical speed accurately. In fact, rotating motion and critical speeds are design criteria of rotating machinery and play an important role in diagnosis and control of rotors.
Harmonic response analysis is a technique used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time and is used to predict the sustained dynamic behavior of structures to consistent cyclic loading. Thus, it can be verified whether or not a machine design will successfully overcome resonance, fatigue, and other harmful effects of forced vibrationsresponses, the influence of asymmetry can be analyzed using a Jeffott model with orthotropic bearings. It is shown (On unbalanced Genta [1], Childs [2]) that at low rotational speeds, the rotor is in direct precession. In the neighborhood and beyond the first critical speed, the behavior becomes retrograde type before becoming direct again for rotational speeds higher than the second critical speed. In comparison with a rotor with isotropic supports, where the orbits are circular for all the rotation frequencies, an anisotropic carrier rotor generates elliptical orbits. The shaft then undergoes normal axial stresses that affect the life of the machine. Also, rotors with anisotropic supports can be used to increase the frequency where instability occurs. Childs [2] shows, for example, that the instabilit originating in the internal damping is shifted thanks to the dissymmetry of the supports. Another phenomenon related to the anisotropy of the supports is the simultaneous manifestation of direct and retrograde precessions along the tree. Muszinska [3] has experimentally observed simultaneous precessions on a vertical rotor whose shaft is flexed. In this study, the

manifestation of simultaneous precessions is affected by the amplitude of the bending of the tree and also by the distribution of imbalances. Identical results (Dias et al [4]) were obtained for a horizontal rotor supported by bearings and bearings. These studies were confirmed by numerical simulations. Similarly, Rao et al. [5], [6] studied experimentally and numerically a horizontal Jeffcott rotor supported by two identical hydrodynamic bearings, showing that the variation of the radial clearance of the bearings (the parameter that controls the anisotropy) is the variable that drives the manifestation of simultaneous precessions: they then observe retrograde precession at the disk level and direct precession elsewhere. Finally,some Rotating machines may exhibit rotor asymmetry. These are, for example, two-pole generators or two-blade propeller systems.

The study of this type of rotor has attracted the attention of several researchers (Sakata et al [7], Kang et al [8], Genta [9]), who showed that this type of system generates a very dynamic rich whose characteristic phenomenon is the existence of a secondary critical speed. This critical speed appears at a speed close to half the synchronous critical speed, more precisely at the intersection of the direct mode frequency evolution curve with the excitation line f = 2F. This is then excited by a constant lateral force, such as the weight of the rotor. From a point of view of the appearance or not of instability, the rotating anisotropies bring up unstable frequency ranges which can be eliminated by the addition of external damping (Genta)[9].
Taplak [10] in his paper studied a program named Dynrot was used to make dynamic analysis and the evaluation of the results. For this purpose, a gas turbine rotor with certain geometrical and mechanical properties was modeled and its dynamic analysis was made by Dynrot program. Gurudatta [11] in his paper presented an alternative procedure called harmonic analysis to identify frequency of a system through amplitude and phase angle plots. The unbalance that exists in any rotor due to eccentricity has been used as excitation to perform such an analysis. ANSYS parametric design language has been implemented to achieve the results.

Sinou[12] investigated the response of a rotor's non- linear dynamics which is supported by roller bearings. He studies on a system comprised of a disk with a single shaft, two flexible bearing supports and a roller bearing. He found that the reason of the exciter is imbalance. He used a numerical method named Harmonic Balance Method for this study. Chouskey[13] et.al studied the influences of internal rotor material damping and the fluid film forces (generated as a result of hydrodynamic action in journal bearings) on the modal behavior of a flexible rotor-shaft system. It is seen that correct estimation of internal friction, in general, and the journal bearing coefficients at the rotor spin-speed are essential to accurately predict the rotor dynamic behaviour. This serves as a first step to get an idea about dynamic rotor stress and, as a result, a dynamic design of rotors.

Whalley and Abdul-Ameer[14], calculated the rotor resonance, critical speed and rotational frequency of a shaft that its, diameter changes by the length, by using basic harmonic response method.

Gasch[15], investigated the dynamic behavior of a Laval (Jeffcott) rotor with a transverse crack on its elastic shaft, and developed the non-linear motion equations which gave important clues on the crack diagnosis.

Das et al [16]. aimed to develop an active vibration control scheme to control the transverse vibrations on the rotor shaft arising from imbalance and they performed an analysis on the vibration control and stability of a rotor- shaft system which has electromagnetic exciters.

Villa et al. [17] studied the non-linear dynamic analysis of a flexible imbalanced rotor supported by roller bearings. They used Harmonic Balance Method for this purpose. Stability of the system was analyzed in frequency term with a method based on complexity. They showed that Harmonic Balance Method has realized the AFT strategy and harmonic solution very efficiently. Lei and Palazzolo [18] have analyzed a flexible rotor system supported by active magnetic bearings and synthesized the Campbell diagrams, case forms and eigen values to optimize the rotor-dynamic characteristics and obtained the stability at the speed range. They also investigated the rotor critical speed, case forms, frequency responses and time responses.

In addition, Łukasz Breńkacz et al. [19] described experimental research. Displacement signals were shown in the bearings and excitation forces used to determine the dynamics of the carrier. The study discussed in this article deals with the rotor supported by two hydrodynamic actuators working in a nonlinear manner. On the basis of calculations, dynamic transaction results were presented for a specific speed.
Fulaj et al. [20] discusses in his research how to obtain critical speeds of the rotor carrier system. A mathematical model for the flexible column was developed with a steel rotor using a specific element Techniques. The limited element model was used to obtain critical speeds in MATLAB.
This paper introduces an alternative procedure called harmonic analysis to determine the frequency of the system through critical speed using MATLAB. The language of matlab parameters design has been implemented to achieve results.
The basic studies of rotary dynamics are related to the Campbell scheme, which represents the evolution of self-repetition as a function of spin velocity and the calculation of unbalance responses mainly during the passage of critical velocity. In using the finite element method, we have seen that gyroscopic moments responsible for the variation of the natural frequency as a function of the speed of rotation or of the circulatory forces which make the movement unstable from a certain speed in the purely linear frame.

II. SOME IMPORTANT PHENOMENA IN ROTOR DYNAMICS

We will now discuss some of the important aspects of rotor dynamics. We will see more particularly the notions of critical velocities, instabilities related to the rotating damping and the role that the dissymmetries can play on the dynamics of the rotors.

A-Critical speeds

The critical speed corresponds to the speed where the unbalance excitation coincides with one of the natural frequencies of the system. For the machines made up of organs with important moments of polar inertia, one notes a strong dependence of the eigen modes vis-à-vis the speed of rotation due to the gyroscopic effects. Thus, we observe the duplication of the eigen modes of the system (for the case of an axisymmetric system) due to the gyroscopic forces as follows:
- A direct precession (FW) where the rotor rotates in the same direction as its precession. Then, under the gyroscopic effects, the associated resonant frequency increases.
- Retrograde precession (BW), in which the rotor rotates in the opposite direction of its precession movement, which gives rise to a softening effect and therefore a fall in the critical speed.

B- Instability due to rotating damping

Depreciation corresponds to one of the determining factors for sizing systems and more particularly for studying the stability of rotating machines. Indeed, the damping (particularly that of the rotating parts) can be responsible for unstable phenomena at high speed, which can lead to the rupture of elements composing the rotor. It is essential to be able to estimate the damping proportions brought on both the fixed parts and the rotating parts, in order to guarantee the integrity of the system.

C-Dissymmetries

Commonly the rotors are designed as axisymmetric. Nevertheless, the stator may have functional characteristics that invalidate this assumption. In this case, the dynamics of the machine will present some differences with respect to the axisymmetric case. The dual modes give way to two simple modes of distinct frequencies leading, as in the case of axisymmetry, to a direct mode and a retrograde mode (figure .1).(Lalanne and Ferraris)[21] .In this case, the retrograde modes are solicited by the synchronous excitation and the number of critical speeds is multiplied by 2, compared to the axisymmetric case.

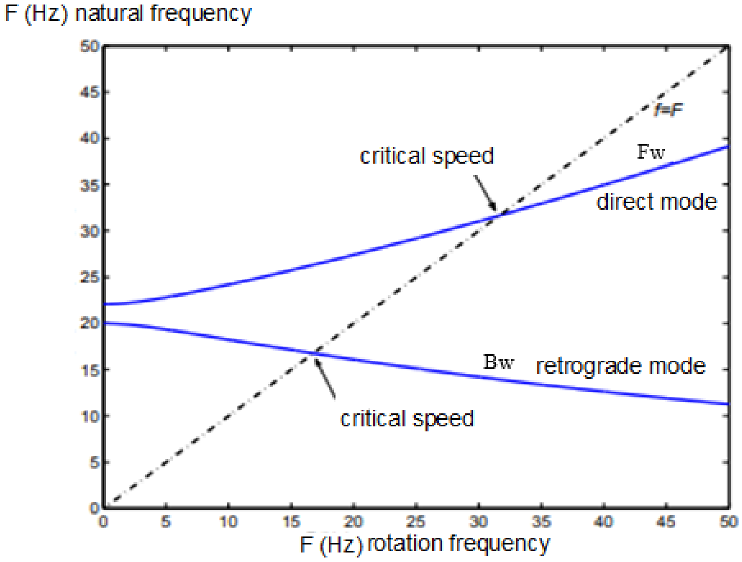


Figure 1.Exemple d'un diagramme de Campbell pour un rotor avec des supports orthotropes.

D- The modal orbit

The points situated in the rotor generator axis describe by the rotor rotation motion and due to the eigenmode of the orbits which have shapes according to the phenomenon to which [22] is envisaged (circular for a symmetrical rotor damped, elliptical name for a rotor asymmetric name depreciated ....) These orbits are generated according to two possible precessions :

- A direct precession where the orbits are described in the same direction as the rotor rotation speed Ω, in which case under the gyroscopic effects, the associated resonant frequency increases.
- A retrograde precession (inverse), where the orbits are described in the opposite direction as the direction of the rotational speed of rotor, from which a relaxation effect and thus a fall of the critical speed.

E-Stability analysis

The stability analysis in the study of vibratory and dynamic behavior of a flexible rotor is necessary since it considered a dynamic system governed by systems of differential equation. The definition of stability covers the definition of Laypunov for equilibrium stability analysis and Poincaré's definition for the concept of orbital stability [23].

We can predict the thresholds of the instability of a dynamic system and in particular in rotor dynamics from the various techniques:

- Sign of the real part of the complex eigenvalues of the equation system in free motion. If the eigenfrequency is given by $s = -a \pm jb$, the only instability is determined when a becomes negative (real positive part), With this criterion one can estimate the frequency as well as the mode for which the system will become unstable.

The Routh-Hrwitz criterion makes it possible to analyze the stability of autonomous systems [24]. The use of this criterion is interesting for systems with a low number of degrees of freedom, for which analytical expressions of the characteristic polynomial associated with the disturbed movement can be deduced. It becomes, however, complex for systems with a large number of degrees of freedom. In addition, this criterion does not provide the frequency of instability.

III. EQUATIONS OF MOTION

By integrating the kinetic energies and the strain energies into the Lagrange equations, we obtain :

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_1}\right)-\frac{\partial T}{\partial q_1}+\frac{\partial U}{\partial q_1}=Fq_1$$

(1)

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_2}\right)-\frac{\partial T}{\partial q_2}+\frac{\partial U}{\partial q_2}=Fq_2$$

(2) The equations of motion ; in matrix form, write:

$$M\ddot{x}+C(\Omega)\dot{x}+Kx=F(t)$$

(3)

M, C and K are the mass matrices, the gyroscopic effect and stiffness, respectively.F: unbalance, bearing, asynchronous force, or other.

$$x=[q_1,q_2]' \quad (4)$$

A- Own frequencies of the rotor

The eigenvalues of the rotor as a function of the speed of rotation are given by:

$$m\ddot{q}_1-a\Omega\dot{q}_2+K_1q_1=0 \quad (5)$$

$$m\ddot{q}_2-a\Omega\dot{q}_1+K_2q_2=0 \quad (6)$$

Or : « a » represents the gyroscopic effect

m : the mass

k1, k2 : stiffness

When stopped (= 0)

$$\omega_{10}=\sqrt{\frac{K_1}{m}} \quad (7)$$

$$\omega_{20}=\sqrt{\frac{K_2}{m}} \quad (8)$$

Turning (≠ 0)

The expressions of the eigenvalues are given according to the speed of rotation in the form:

$$\omega_1=\left[\frac{\omega^2{}_{10}}{2}+\frac{\omega^2{}_{20}}{2}+\frac{a^2\Omega^2}{2m^2}-\sqrt{\left(\frac{\omega^2{}_{10}}{2}+\frac{\omega^2{}_{20}}{2}+\frac{a^2\Omega^2}{2m^2}\right)^2-\omega^2{}_{10}\omega^2{}_{20}}\right]^{1/2} \quad (8)$$

$$\omega_2=\left[\frac{\omega^2{}_{10}}{2}+\frac{\omega^2{}_{20}}{2}+\frac{a^2\Omega^2}{2m^2}-\sqrt{\left(\frac{\omega^2{}_{10}}{2}+\frac{\omega^2{}_{20}}{2}+\frac{a^2\Omega^2}{2m^2}\right)^2-\omega^2{}_{10}\omega^2{}_{20}}\right]^{1/2} \quad (9)$$

IV. METHODOLOGY

The main objective of this work is the study the Campbell scheme, which represents the evolution of self-repetition as a function of spin velocity and the calculation of unbalance responses mainly during the passage of critical velocity. In using the finite element method.We designed a mathematical model under the name of Nelson rotor in a matlab program containing the engineering data of the Nelson rotor element (tree data, disk data, The geometric data of the element carrying the Nelson rotor .In addition to the matrices of stiffness and damping in the form of a set of nodes and elements to calculate the existing stiffness and frequency values in the presence of the speed of 4800-28800 rpm .This calculation mode must be able to give the geometry of the rotor in finite elements. The search for eigenvalues is a fundamental operation in the study of rotor dynamic.

A- Model

The model considered is a Nelson rotor [25]. Fig. 2, which is a 0.355 (m) long overhanging steel shaft of 14 different cross sections. The shaft carries a rotor of mass 1.401(kg) and eccentricity 0.635(cm) at 0.0889(m) from left end and is supported by firstly two bearings at a distance of 0.1651(m) and 0.287(m) from the left end respectively. Six stations are considered during harmonic analysis as shown in Fig.1, where station numbers denote different nodes in themodel (1) Left extreme of shaft, (2) Disc, (3) First bearing node, (5) Between the two bearings, (4) Second bearing node and (6) Right extreme of shaft. A density of 7806 kg/m3 and elastic modulus 2.078E11 n/m2 were used for the distributed rotor and a concentrated disk with a mass of 1.401 kg,polar inertia 0.002 kg.m2 and diametral inertia 0.00136 kg.m was located at station five.

The following cases of bearings were analyzed :

a) Symmetric orthotropic bearings

b) Fluid film bearings.

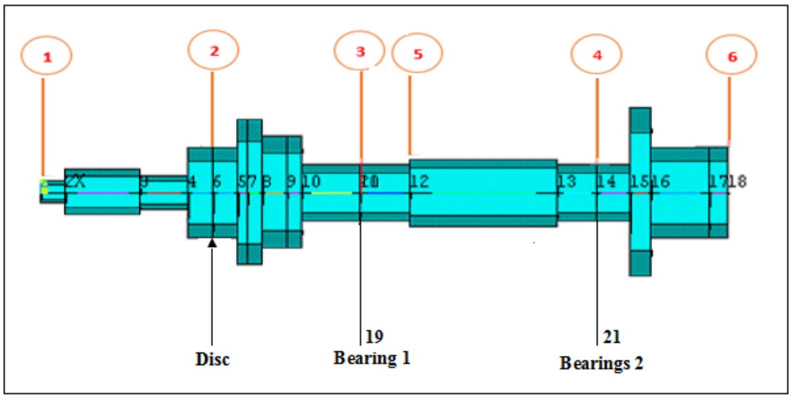


Figure 2. Model of Nelson rotor with various sections, disc and bearings Numbers indicate station numbers

B- Calculation of the Eigen values of a system

We designed a mathematical model under the name Nelson rotor in a Matlab software that contains the engineering data of the Nelson rotor element (shaft data, disk data, bearing data). In addition to the Matrices of the stiffness and damping in the form of a set of nodes and elements to calculate the values of the existing stiffness and frequency in the presence of the speed 4800-28800 rpm. Table 1.

Element Node No	Node Location (cm)	Bearing and Disk	Inner Diameter (cm)	Outer Diameter (cm)
1	0.0		0.0	0.51
2	1.27		0.0	1.02
3	5.08		0.0	0.76
4	7.62		0.0	2.03
5	8.98	Disk	0.0	2.03
6	10.16		0.0	3.30
7	10.67		1.52	3.30
8	11.43		1.78	2.54
9	12.70		0.0	2.54
10	13.46		0.0	1.27
11	16.51	Bearing	0.0	1.27
12	19.05		0.0	1.52
13	22.86		0.0	1.52
14	26.67		0.0	1.27
15	28.70	Bearing	0.0	1.27
16	30.48		0.0	3.81
17	31.50		0.0	2.03
18	34.54		1.52	2.03

C. Resolution algorithm

The algorithm for calculating the eigenvalues of a homogeneous system is described in Figure .3. calculations are done with the Matlab programming code.

• Algorithm for calculating the eigenvalues of a system

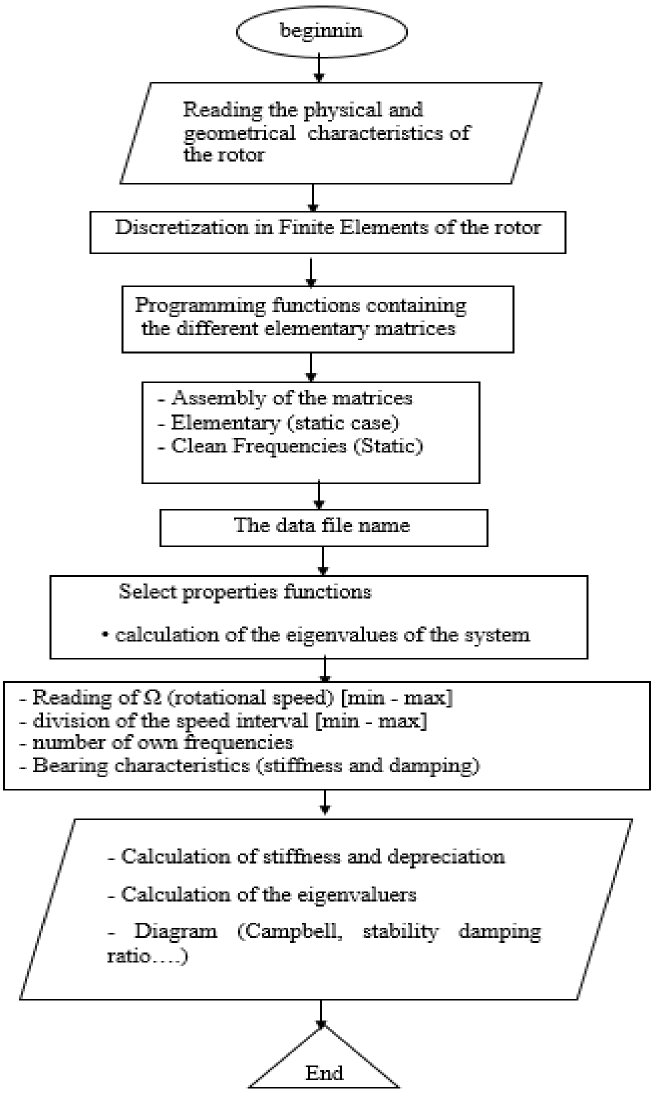


Figure .3. Algorithm for calculating the eigenvalues.

V. RESULTS AND DISCUSSION

A-Data of fluid film bearings

The shaft is supported by two fluid film bearings whose stiffnesscoefficient Table .2 was calculated by Matlab as follows :

Dynamic Coefficients of Hydrodynamic Bearings			
Stiffness coefficients (N/m)			
K_{yy}	K_{yz}	K_{zy}	K_{zz}
7.7539E + 007	2.3381E + 008	-5.4601E + 008	1.3399E + 008
1.3594E +008	5.8365E + 008	-8.412E + 007	1.4718E + 008

Table .2: Dynamic coefficients of hydrodynamic bearings.
Stiffness coefficients (N / m).

While the damping components are = = 1752 (Ns / m). The imbalance response for an eccentricity of the disk center of 0.635 (cm) at the second station was determined for a range of speeds from 4800 to 2800 rpm.

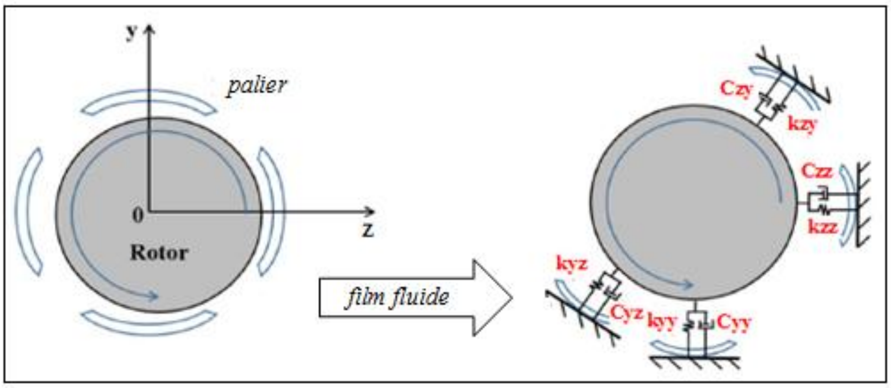


Figure .4. Schematic view of the rotor on the bearing supports and idealization of the fluid film coefficients

The Matlab program also provided us with a model for Nelson rotor with various sections, disc and bearings.Fig.5.

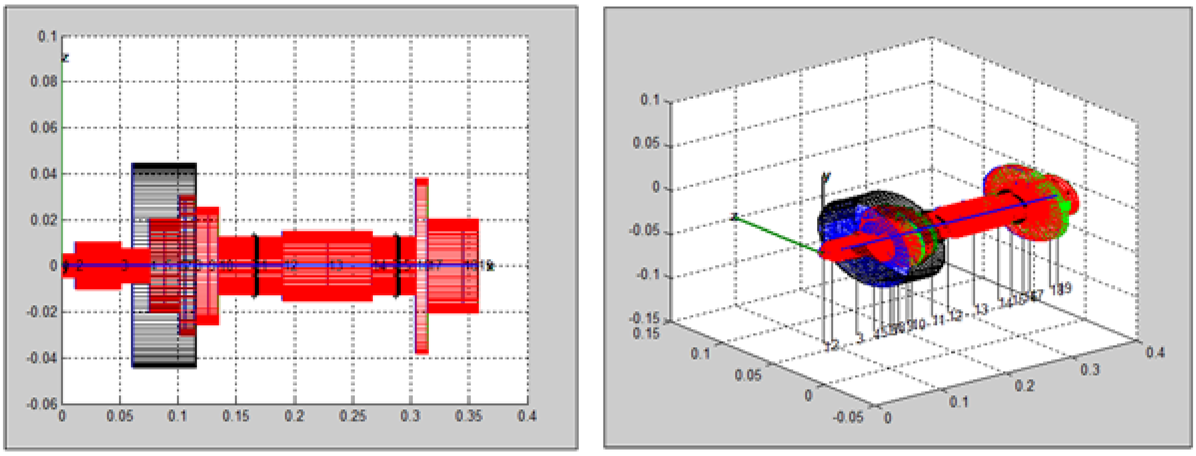


Figure 5. Nelson rotor with various sections.

B- Estimated critical speeds

1-The Campbell Diagram

Figure 6 Shows the Campbell diagram of the rotor-shaft system, when the shafts internal material damping is considered. The graph is plotted by using the whirl frequencies (obtained from the imaginary part of the eigenvalues), and there are two positions, the first position in reverse rotation "BW", where the rotor rotates in the opposite direction. The second position is the rotation "FW", where the rotor rotates in the direction of rotation. The critical speed corresponding to the first position and the critical speed corresponding to the second position appear in Table.3.

Table.3 lists the values of the first critical speeds identified :

Mode	Critical speed (Hz)	Critical speed (rpm)
5	3.3837e+002	2.0302e+004
6	4.0607e+002	2.4364e+004

Table .3. Critical speeds

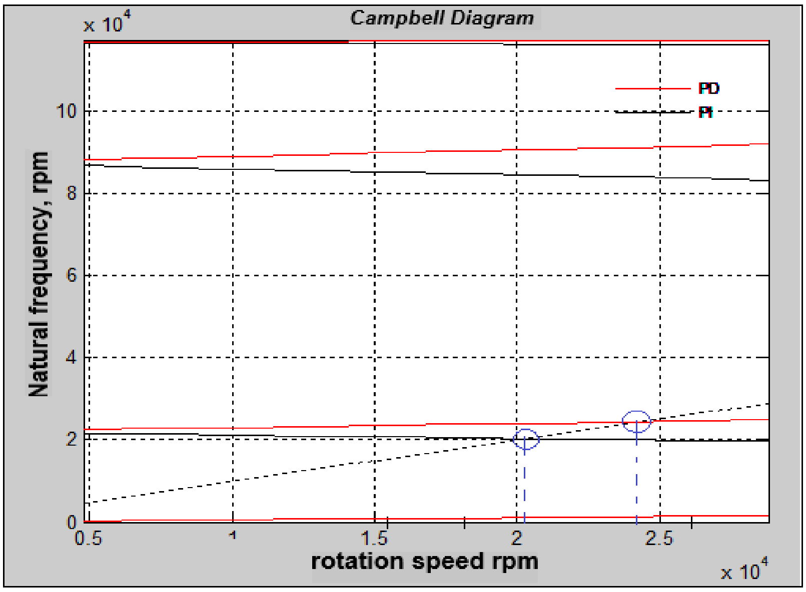


Figure 6. Campbell Diagram.

Results for the Stability Chart, Damping Report Diagram, Mode Shapes, Elliptical Orbits, Mode Shape Precession, Root Locus Diagram. Are presented by matlab program as follows.

2. Stability diagram

The stability diagram (Figure .7) presents the evolution of the damping constant as a function of the rotational speed. We observe through the diagram that the values of the damping coefficient are negative and indicates that the rotor is stable.

Onset of instability speeds

Mode	(Hz)	(rpm)
1	0.0000e+000	0.0000e+000
2	0.0000e+000	0.0000e+000
3	0.0000e+000	0.0000e+000
4	0.0000e+000	0.0000e+000
5	0.0000e+000	0.0000e+000
6	0.0000e+000	0.0000e+000
7	0.0000e+000	0.0000e+000
8	0.0000e+000	0.0000e+000
9	0.0000e+000	0.0000e+000
10	0.0000e+000	0.0000e+000

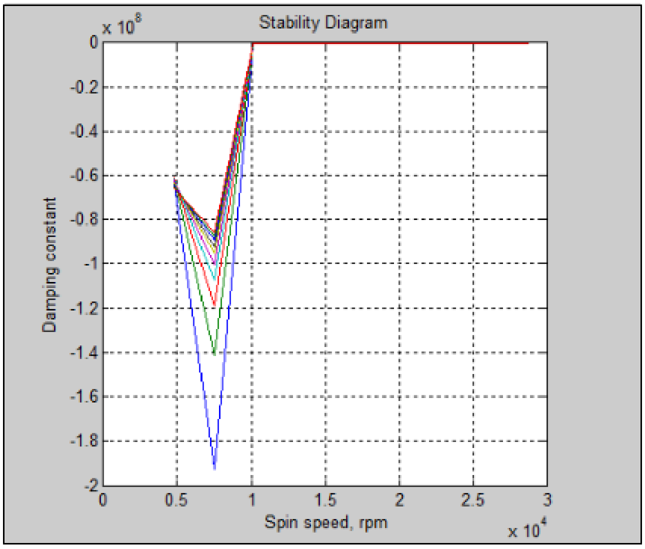


Figure .7. Stability diagram

3. damping ratio

The damping ratio diagram. Figure .8, show the evolution of the damping ratio as a function of the speed of rotation, we note that the system regardless of the speed of rotation (4800-28800tr / min), rotor system operation always stable.

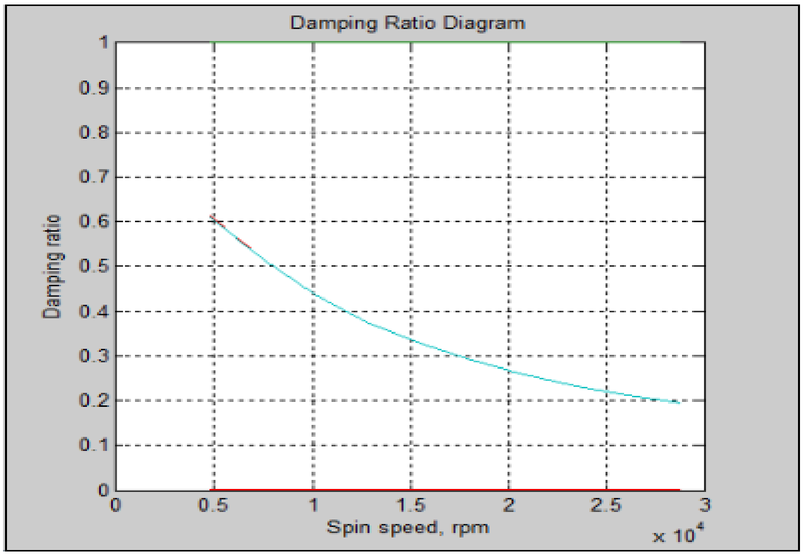


Figure 8. damping ratio diagram.

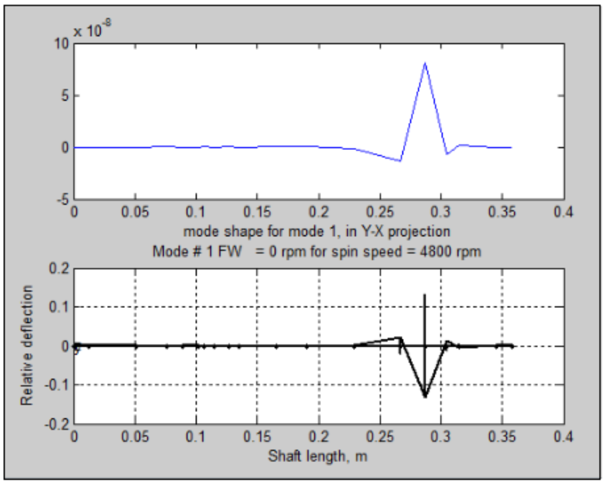
4. Mode Forms and Mode Form Precession

Table.4 gives the result of the shapes of the modes and the shape precession and the rotational speed for the modes. We observe that the modes 1, 2,3, 4, 6, 8, 10 are direct precession (the rotor rotates in the direction of rotation), and the modes 5, 7,9, are inverse precession (the rotor rotates in the opposite direction).

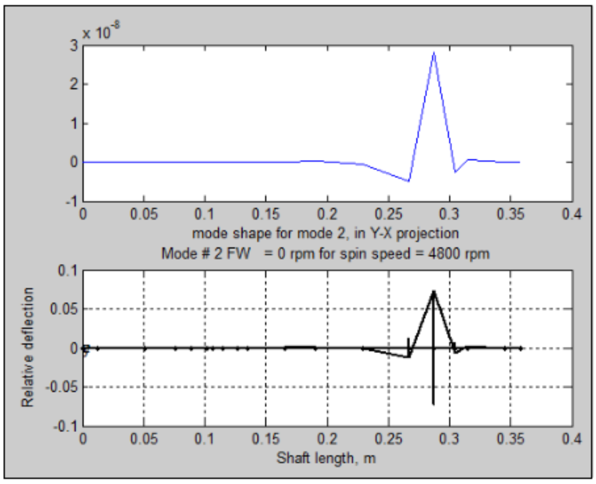
Modes	Precession	Spin speed rpm
1	direct	FW=0
2	direct	FW=0
3	direct	FW=281.8067 rpm
4	direct	FW=281.8176 rpm
5	inverse	BM=21645.764 rpm
6	direct	FW=22513.8662 rpm
7	inverse	BM=86691.2517 rpm
8	direct	FW=88125.162 rpm
9	inverse	BM=116503.6699 rpm
10	direct	FW=116711.5916 rpm

Table.4. Forms of Modes and Precession of Formsat 4800 rpm.

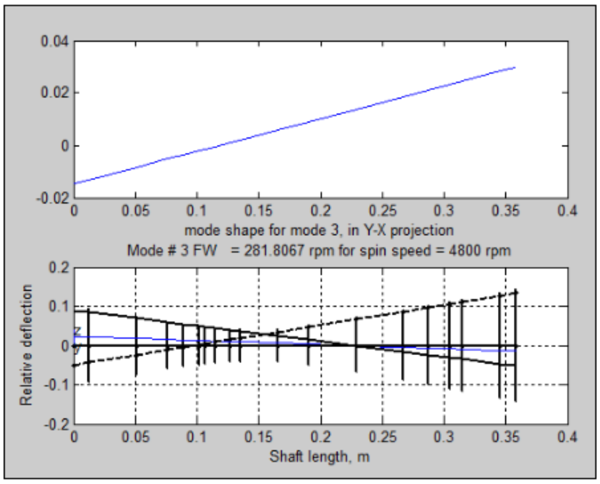
Figure .9 gives relative deviation as a function of tree length and confirms the results obtained in Table 4.



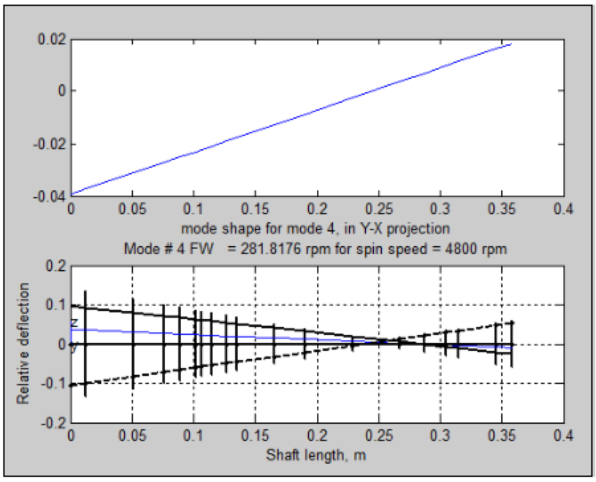
Mode 1



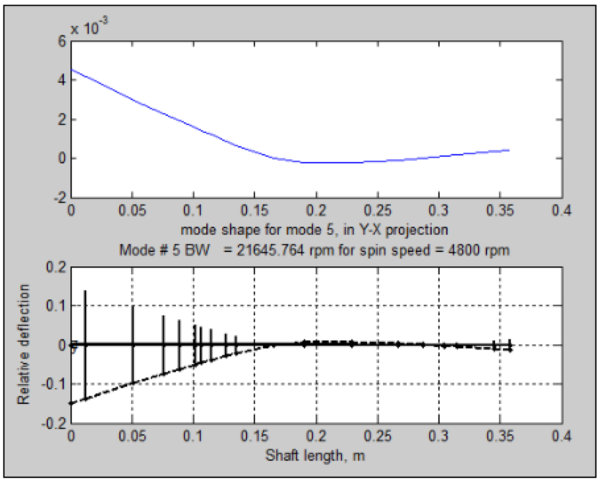
Mode 2



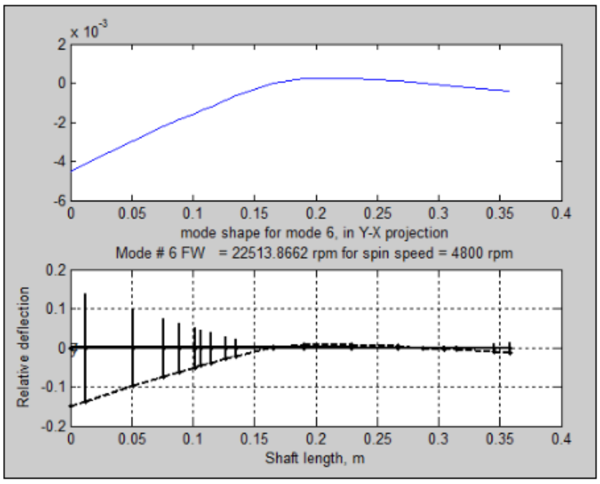
Mode 3



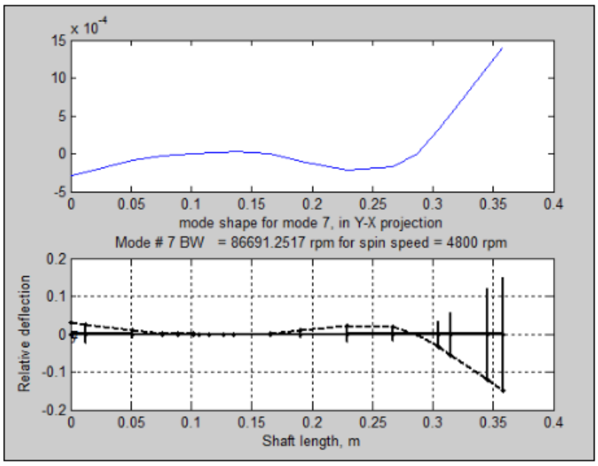
Mode 4

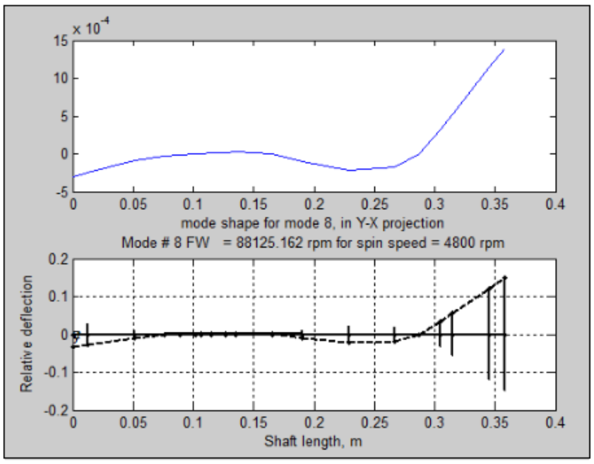


Mode 5

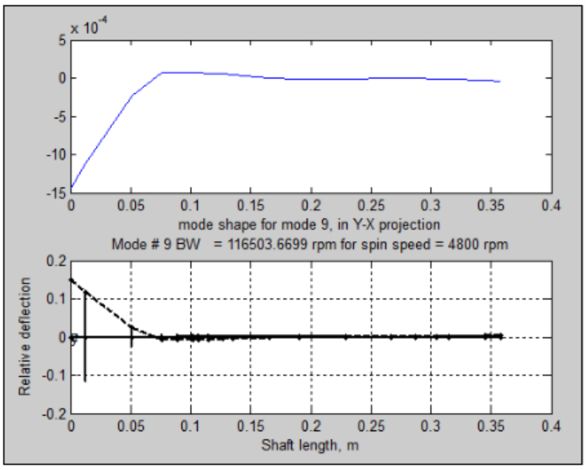


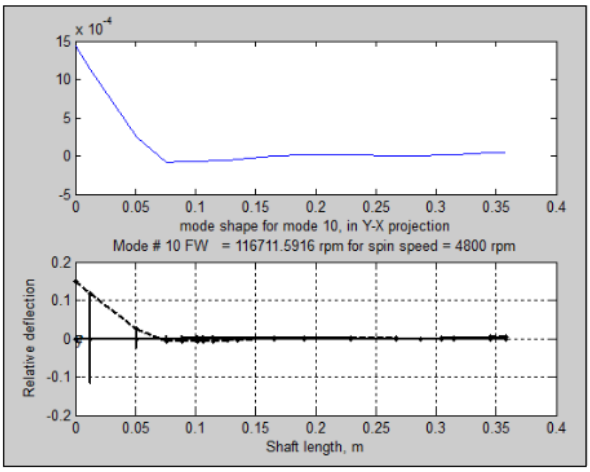
Mode 6





Mode 7 Mode 8





Mode 9 Mode 10

Figure .9. Forms of Modes and Precession of Forms of Modes

at 4800 rpm.

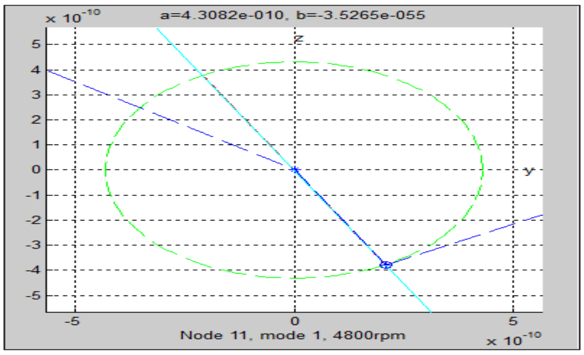
4. Elliptical Orbits

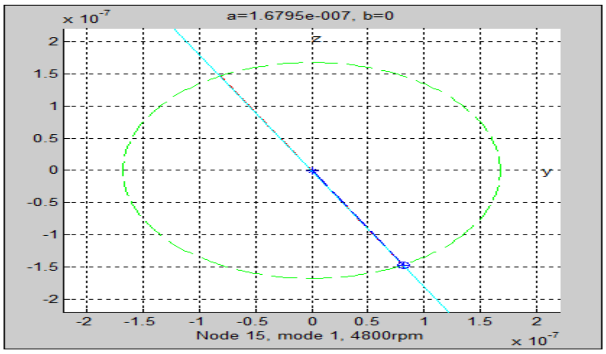
We see the representation of the orbits on the measurement planes, For the rotation speed 4800 rpm, the orbits have forme :

Circular for a symmetrical rotor damped, elliptical name for a rotor asymmetric shape.

The direction of precession of the orbits obtained is represented graphically in Figure .10 with the beginning of the orbit represented by a circle and the end represented by a star. The figure shows that the orbits C1, C2, C6, C10 are described in the same sense as the rotational speed of the rotor Ω, in which case under the gyroscopic effects, the associated resonance frequency increases.

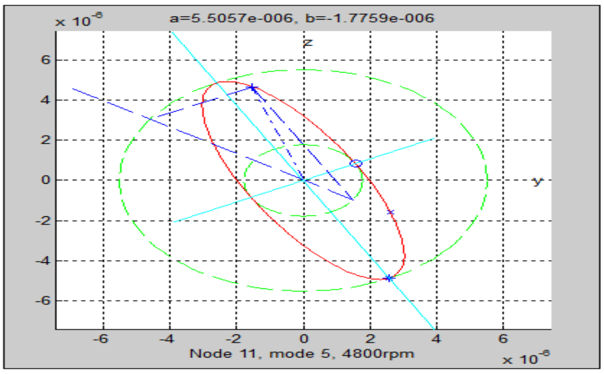
And the C5 orbits are described in the opposite direction as the direction of the rotational speed of rotor, from which generated a softening effect and therefore a fall of the critical speed.

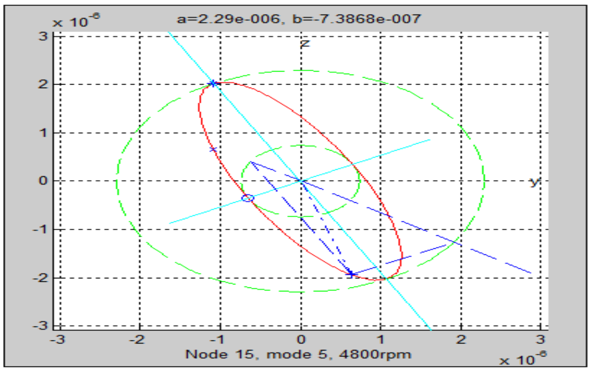




C 1

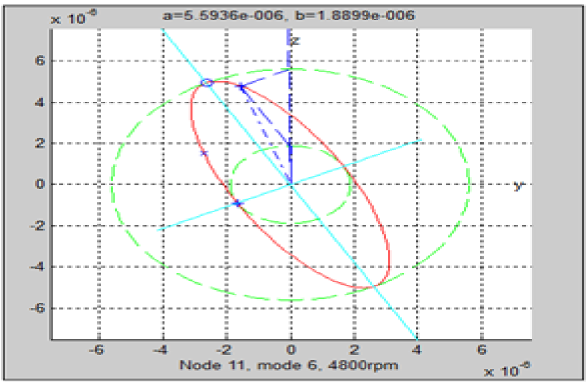
C 2

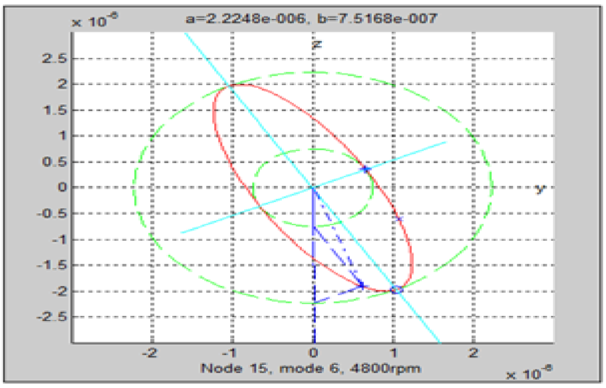




C 5

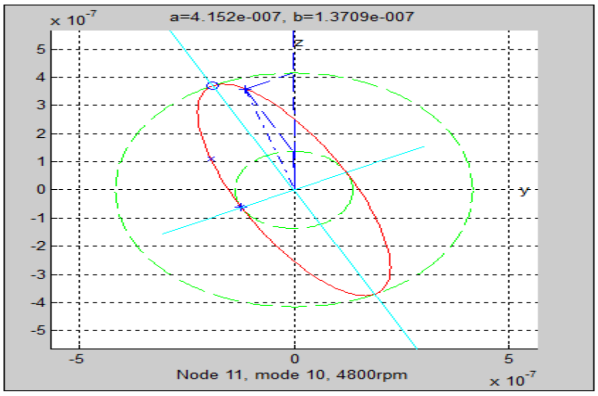
C 5

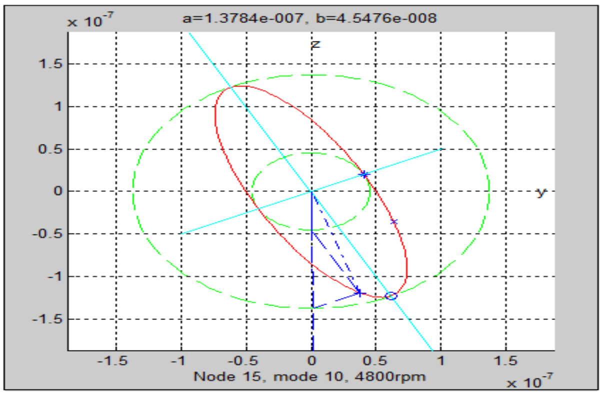




C 6

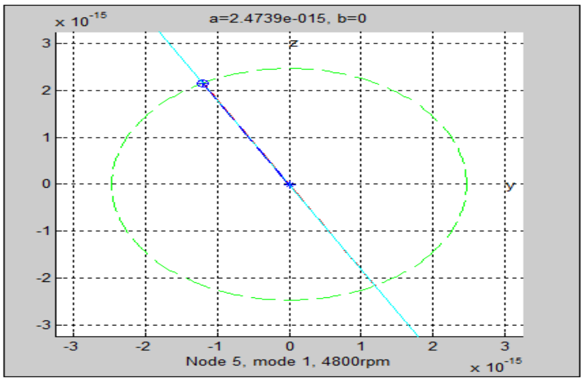
C 6



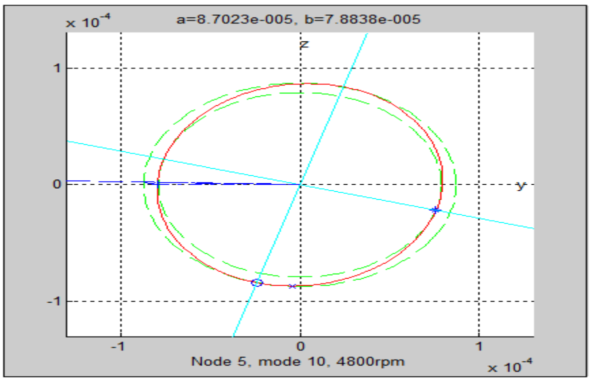


C 10

C 10



C1

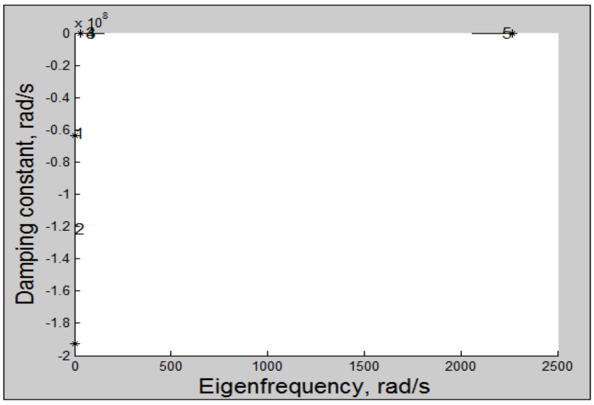


C10

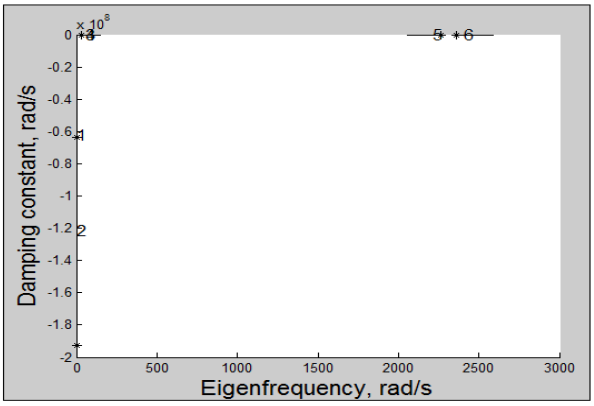
Figure 10. Orbits at 4800 rpm

5. root locus diagram

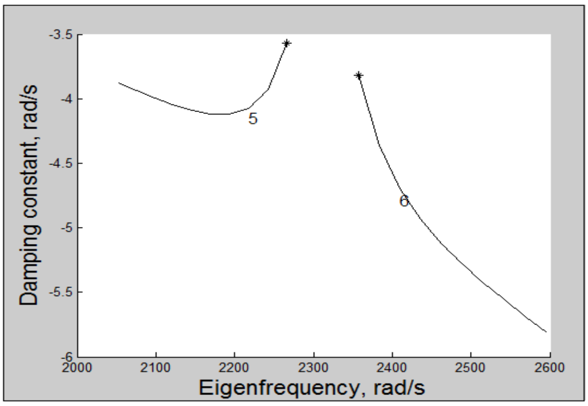
The root locus diagram figure 11. Shows the evolution of the damping constant as a function of the natural frequency. For example, we notice the direction of the modes, 5, 7, 9 from left to right (odd), so the mode a reverse precession. And the direction of the modes 4,6, 8,10 from right to left (even), so the direct precession mode.



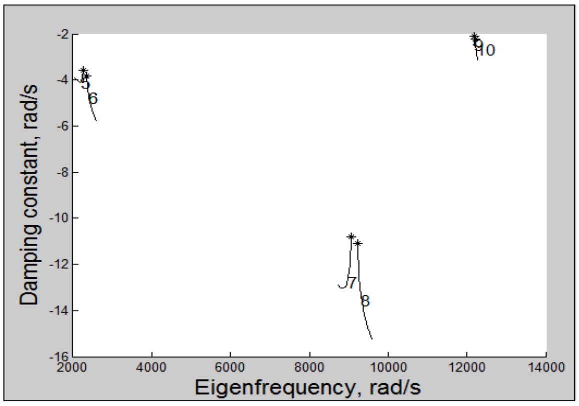
Mode 1 / mode 5



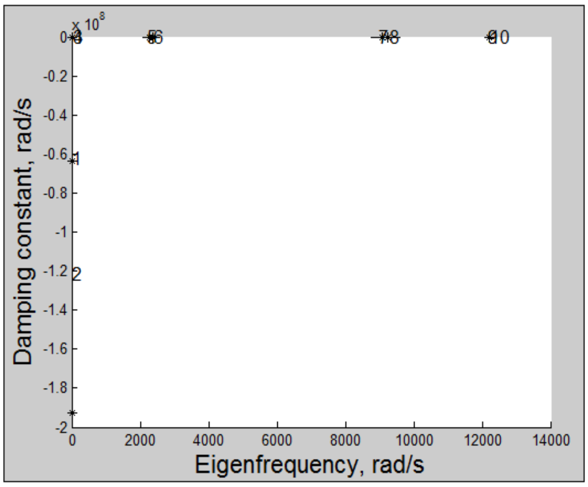
mode 1 / mode 6



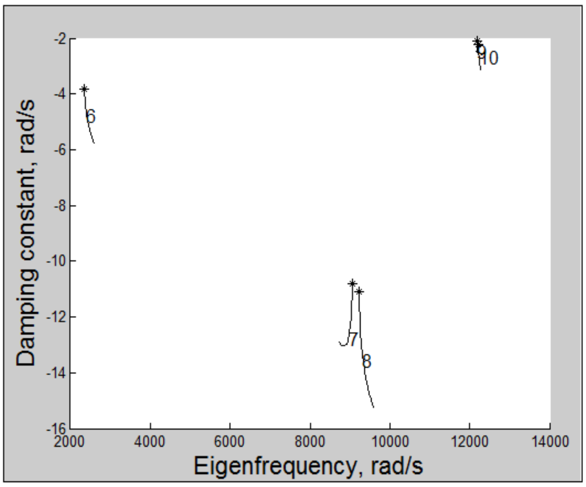
Mode 5 / mode 6



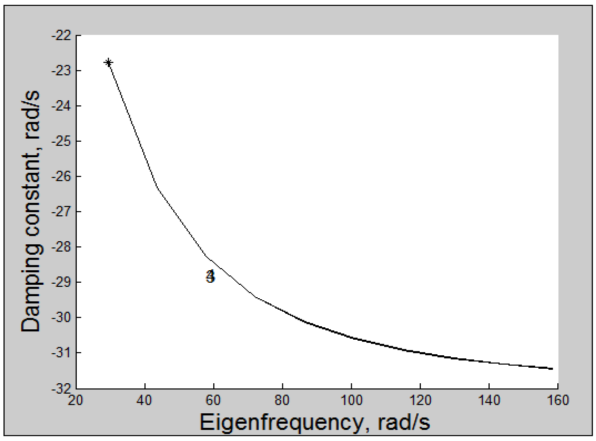
mode 5/ mode 10



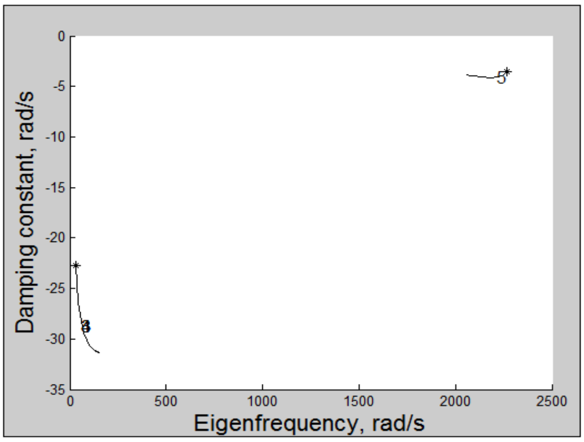
Mode 1/mode 10



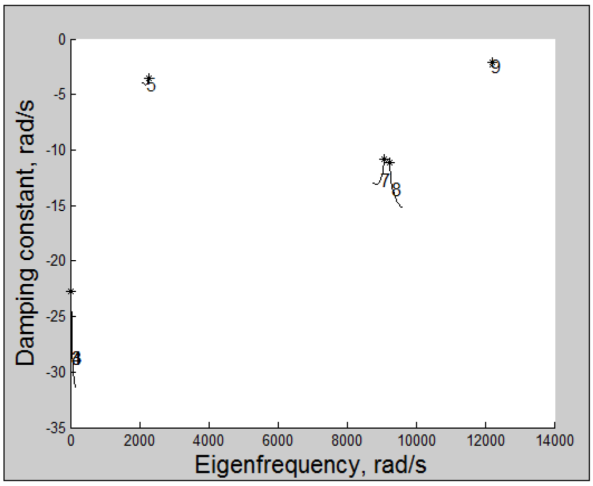
mode 6/mode10



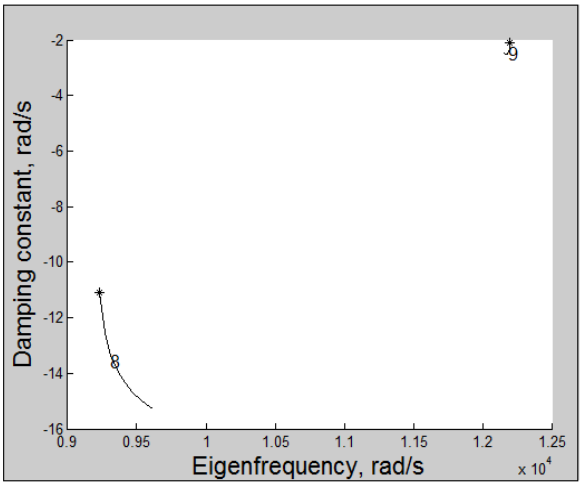
Mode 3/mode 4



mode 3 /mode 5



mode 7/mode 9



mode 8 /mode 9

Figure .11. Root Locus Diagram

VI. Conclusions

The general equations of a uniformly rotated rotor have been developed in this work using the finite element method. It is more adapted to model the real systems insofar as one knows the dynamic characteristics of the bearings for example. It allows the study of all modes of vibration of the rotor. It is also modular because each element of the rotor has its own characteristics.The effects of rotary inertia, and internal damping were included in the analysis. Study also obtained the variation of amplitude response with frequency as it is important for minimizing the noise of the rotor. The increasing amplitude increases the noise of the rotor. This analysis gives the alternate procedure of finding the critical speed that is harmonic analysis.

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NOMENCLATURE

Kyy, Kyz, Kzy, Kzz stiffness coefficients.

Cyy, Cyz, Czy, Czz damping coefficients.

a gyroscopic effect

k1, k2 stiffness

ω The speed of rotation of the shaft (rd / s)
Ω angular velocity of the shaft (rd / s)