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# DESIGN OF A MULTI-SPAN ROTOR TEST RIG FOR INVESTIGATION OF ROTATIVE PHENOMENA

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***Abstract***. *This work presents the design of a prototype for a reduced scale multi-span rotor. An analytical method, which involves the principle of linear and angular momentum, was applied for the modeling. For the simulation of the system response we did not chose a single value for the parameters of the system, but we varied them in an appropriate range. This was necessary for the evaluation of the sensitivity of the systems response with respect to parameters, such as shaft, bearing or rotor (mass and inertia) characteristics. With this prototype it is possible to study the influence of joints in the drive line and misalignment of bearings during the operation. This procedure is helpful towards a better understanding of the dynamic behavior of this type of rotating machines (multirotor-bearing system).*

***Key words:*** *Dynamics of Rotation, Mechanical Project*

## 1. INTRODUCTION

Rotating machinery is found in most industrial processes. Depending on the application, this mechanical equipment presents a wide variety of physical characteristics, as well as a wide operation range in the processes of the industrial plants they are in. As an example we cite: gas turbines, steam turbines, bombs, compressors, turbogenerators, etc... These equipment are very common in petroleum industries, and energy generation plants.

The technological progress has changed the development of rotating machines. The result are more efficient and sophisticated units, although more complex.

This progress created new procedures for diagnose and organization of the maintenance. Diagnose involves concepts in the field of statics, dynamics, mechanics of fluids, rotordynamics and heat transfer. Furthermore, knowledge about experimental measurement techniques is necessary.

The study of rotating machines involves computational and experimental analysis to indicate the several possible sources of problems that induce an incorrect operation. To continue some publications in this area will be mentioned. Swarnamani (1995) studied the effects of a flexible joining between the driving and the driven parts in a machine. He analyzed how the misalignment gives origin to periodic forces and moments, which act on the shaft of the machine. Xu (1994) studied the misalignment and unbalancing of a rotating machine with universal coupling. For the validation of the theoretical results he compared their power spectrum with that of the measured signals. Hirano (1997) studied a turbogenerator system and presented an experimental confirmation of changes in the critical speeds and in the vibration amplitude due to the misalignment of the bearing. Lee studied (1997), theoretically and experimentally, the effects of misalignment in a system with ball bearings.

## 2. DEVELOPED SYSTEM

The prototype test rig to be studied shall represent a part of some plant of electric power generation. These machines are indispensable in an industrial process. A theoretical development of the simplified model of a turbogenerator will be considered. The model consists of two mounted elastic shafts on rigid selfaligned bearings, each shaft carrying rigid disk(s). The shafts are united through the joining with a rigid or flexible coupling. This last one provides an angular misalignment.

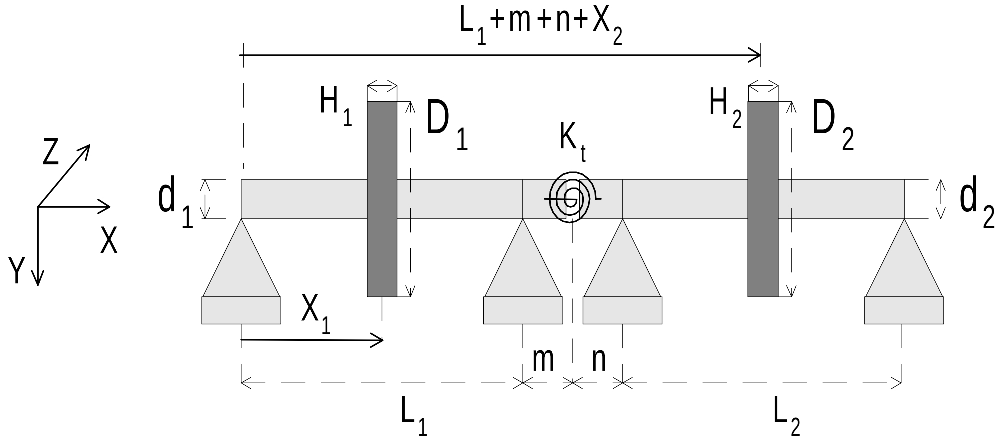


Figura 1: Model studied

This analysis neglects the influence of the masses of the shaft and of the joining in respect to the mass value of the disk. The coordinates of motion of the system will be written in terms of the rotation and translation of the disks in the perpendicular plan to the rotation axis. In this work the equations of motion were obtained according to the Figure 1.

### 2.1. Dynamics of the Rigid Body

The rotation matrix was formed by three sequential rotations in relation to the coordinate axes (Cardan Angles). The sequence of rotations are; $A \xrightarrow[3]{\varphi_Z} B$ rotation $\varphi_Z$ in the axis Z that takes the System of the A for another B, $B \xrightarrow[2]{\varphi_Y} C$ rotation $\varphi_Y$ in the axis Y that takes the System of B for another C, $C \xrightarrow[1]{\varphi_X} D$ rotation $\varphi_X$ in the axis X that takes the System of C for another D. The first two angles will be considered small and the equations are linearized.

Next, these equations of motion are written from the angular momentum conservation in the inertial system A:
Euler:

$$\begin{aligned} \mathbf{M}_G = & \quad \dot{\mathbf{H}}_G \\ \mathbf{H}_G = & \quad I_G \; {}_A\boldsymbol{\omega}_D \end{aligned} \tag{1}$$

Newton:

$$\begin{aligned} \mathbf{F}_G = & \quad \dot{\mathbf{L}}_G \\ \mathbf{L}_G = & \quad M\,\mathbf{V}_G \end{aligned} \tag{2}$$

In the two previous equations:

$\mathbf{M}_G$: the moment resulting in the system.
$\mathbf{H}_G$: angular momentum of the system.
$\mathbf{I}_G$: moment of inertia of the system.
$\boldsymbol{\omega}$: absolute angular speed.
$\mathbf{F}_G$: the force resulting in the system.
$\mathbf{L}_G$: linear momentum system.
$\mathbf{M}_G$: Mass of the system.
$\mathbf{V}$: linear speed.

From equations (1) and (2), the forces and the moments applied in the mass center (S) are writen next:

$$\left\{ \begin{array}{c} F_Y \\ M_Z \\ F_Z \\ M_Y \end{array} \right\}_S = \underbrace{\left[ \begin{array}{cccc} M & & & \\ & I_a & & \\ & & M & \\ & & & I_a \end{array} \right]}_{\text{Inertia Matrix}} \left\{ \begin{array}{c} \ddot{Y} \\ \ddot{\varphi}_Z \\ \ddot{Z} \\ \ddot{\varphi}_Y \end{array} \right\}_S + \underbrace{\left[ \begin{array}{cccc} & & & -I_p\Omega \\ & & & \\ & & & \\ & I_p\Omega & & \end{array} \right]}_{\text{Gyroscopic Matrix}} \left\{ \begin{array}{c} \dot{Y} \\ \dot{\varphi}_Z \\ \dot{Z} \\ \dot{\varphi}_Y \end{array} \right\}_S \tag{3}$$

## 2.2. Stiffness matrix and matrix of external damping

The investigated system represents the joining of shafts in a simple turbo-machine. The stiffness matrix will be obtained through the inverse of the flexibility matrix. Their elements will be calculated one by one, considering that each force assumes an unitary value and all others are null. The displacements will be measured in all degrees of freedom, forming the flexibility matrix. There are defined 8 degrees of freedom to determine the equations of motion, 4 for each disk, two of rotation and two of displacement, considering rigid the bearings.

The dissipation of energy is modeled as viscous damping forces. Its determination in practice is difficult, but not impossible if some simplifications are considered and the correct measurements are made. The viscous damping matrix is symmetrical looking like the stiffness matrix (Weber, 1992). Writing the elastic and viscous forces applied on the shaft (W), we get:

$$\{\mathbf{F}\}_k = [\mathbf{K}]\{\mathbf{Q}\}_W \tag{4}$$

$$\{\mathbf{F}\}_c = [\mathbf{K}]\{\dot{\mathbf{Q}}\}_W \tag{5}$$

where

$\mathbf{F}_k$: elastic forces acting on the shaft.
$\mathbf{K}$: stiffness matrix of the shaft, (Chavez, 1999).
$\mathbf{Q}_G$: displacements of the shaft center.
$\mathbf{F}_c$: viscous forces acting on the shaft.
$\mathbf{C}$: damping matrix.
$\dot{\mathbf{Q}}_G$: speed of the shaft center.

## 3. NUMERIC SIMULATION

The Campbell diagram of the system presented in the Figure 1 has its parameters defined in the following tables, as well the loads, the materials and the coupling.

Tabela 1: Geometric characteristics

| Semi-Shafts | | | | | | Discos | | | |
|---|---|---|---|---|---|---|---|---|---|
| $d_{1(\mathrm{m})}$ | $L_{1(\mathrm{m})}$ | $\mathrm{m}_{(\mathrm{m})}$ | $d_{2(\mathrm{m})}$ | $L_{2(\mathrm{m})}$ | $\mathrm{n}_{(\mathrm{m})}$ | $D_{1(\mathrm{m})}$ | $H_{1(\mathrm{m})}$ | $D_{2(\mathrm{m})}$ | $H_{2(\mathrm{m})}$ |
| 0.008 | 0.315 | 0.035 | 0.008 | 0.315 | 0.035 | 0.12 | 0.08 | 0.12 | 0.08 |

Tabela 2: Shipments

| $X_{1(\mathrm{m})}$ | $X_{2(\mathrm{m})}$ | $E_{1(\mathrm{GPa})}$ | $\rho_{1(\frac{\mathrm{kg}}{\mathrm{m}^3})}$ | $E_{2(\mathrm{GPa})}$ | $\rho_{2(\frac{\mathrm{kg}}{\mathrm{m}^3})}$ | $\kappa_{t(N.\frac{\mathrm{m}}{\mathrm{rad}})}$ |
|---|---|---|---|---|---|---|
| 0.25 | 0.10 | 200E+9 | 7863 | 110E+9 | 8860 | $\infty$ |

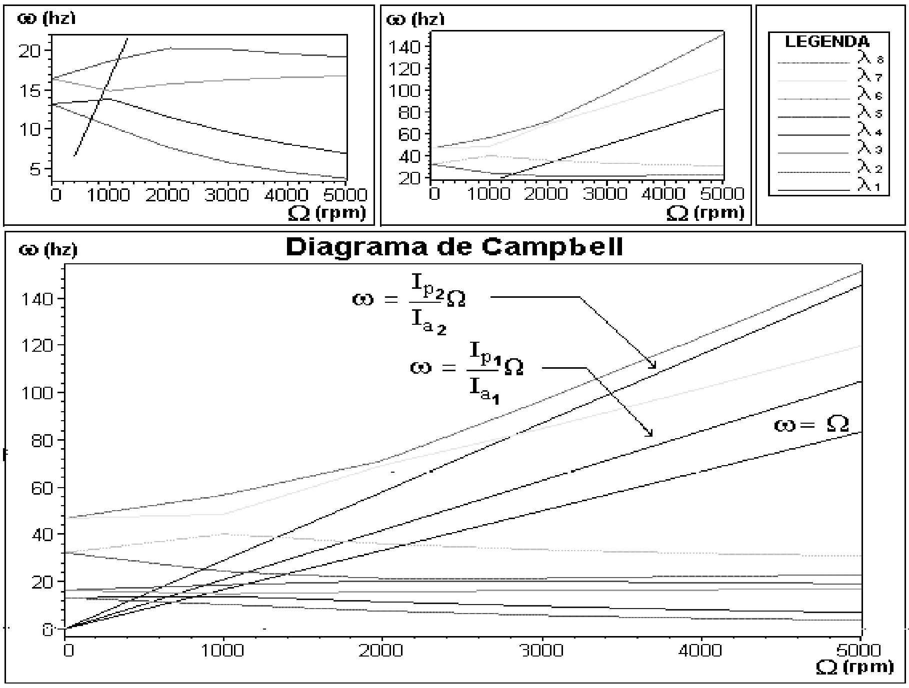


Figura 2: Campbell Diagram

The modal shapes corresponding to the Campbell diagram of the simulated system are presented in the following drawing.

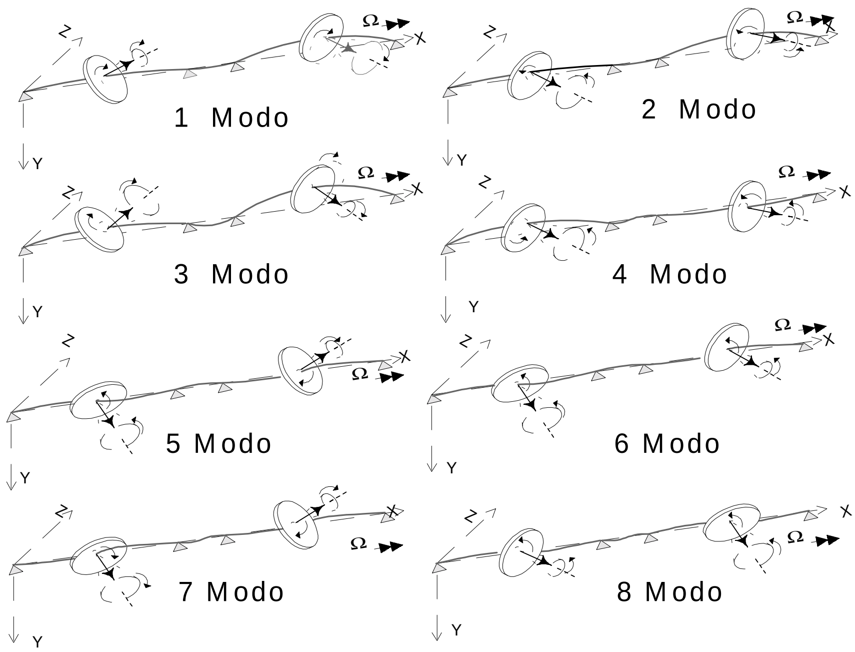


Figura 3: Modal shapes

## 4. MULTI-SPAN ROTOR

The objective in this prototype development, corresponding to a particular rotating machine, is to cross the largest possible number of critical speeds. This will supply interesting information on the dynamic characteristics of this machine.

Figura 4: Prototype of tubogenerator

The design parameters were chosen in a way that it is possible to operate and measure unbalances, shaft bow, bearing anisotropy, misalignment, etc. Results from simulation of the designed prototype allow an estimate of the resulting dynamic behavior.

The multi-span Rotor is a two-stage rotor, each stage with three disks (rotors). A coupling joins the two stages. An electric motor (CA), which is controlled by frequency inverter, supplies the power to the system.

This prototype allows the experimental study of the influences on the system response of couplings and bearing misalignment in operation, and may be helpful towards a better understanding of the dynamic behavior of this type of rotating machinery (multirotor-bearing systems)

## 5. IDENTIFICATION OF THE MULTI-SPAN ROTOR

In this part it is included the measurements of the first two natural frequencies corresponding to the first two modal shapes without rotation. In the experiment the sensors 3 and 4 are positioned in the vertical direction and the sensors 1 and 2 the horizontal direction.

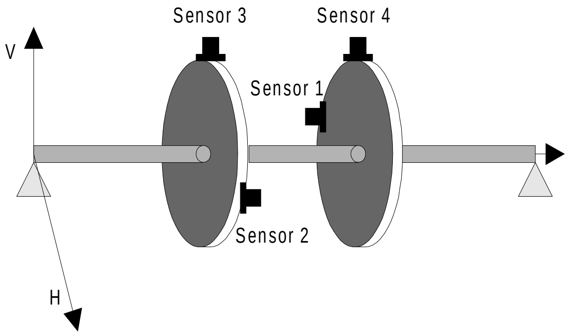


Figura 5: Disposition of the test

The system is excited with the impact hammer. This Figures represents the value and the phases for the two mentioned cases.

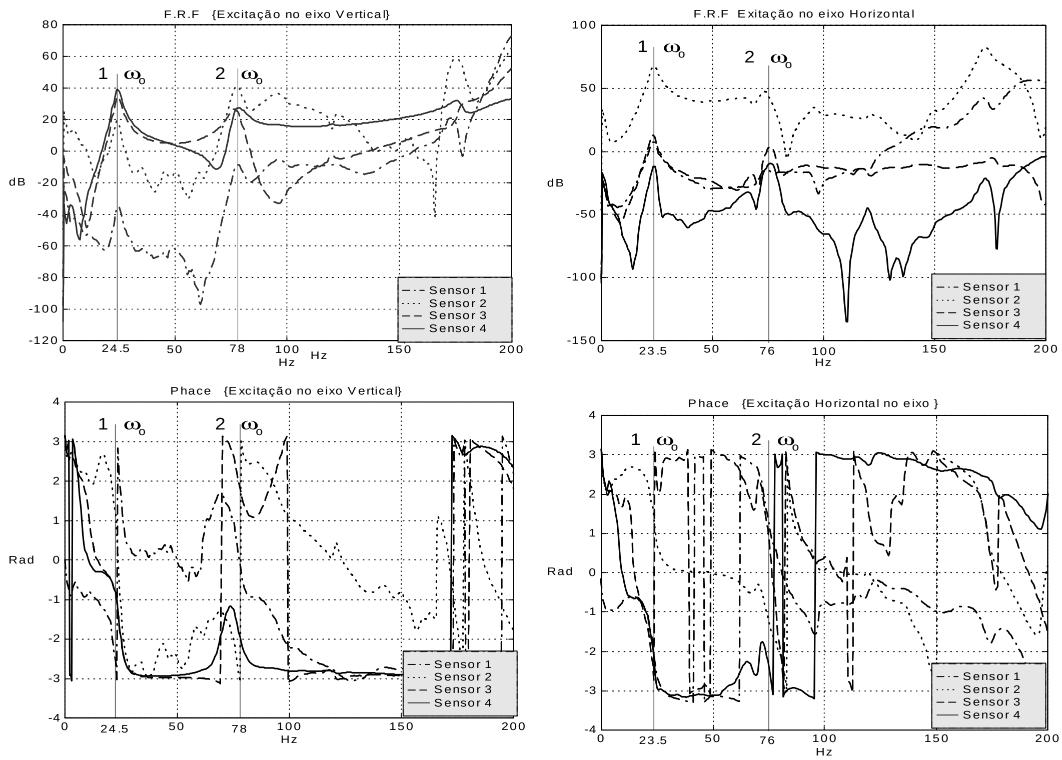


Figura 6: Vertical and horizontal impulse on the Shaft

## 5.1. CONCLUSIONS

The project of a prototype in reduced scale of a turbogenerator was developed with the purpose of a basic study of the dynamics of this type of machines. This was based in the analytic modeling to obtain the value of the parameters of the system and finally the project of the multi-rotor span was developed. A difference in the value of the first two natural frequencies was observed, in the two directions due to anisotropy in the real bearing). The validation of the experimental results with the one obtained from the model, provides a first contact with experimental techniques. A complete experimental evaluation of the whole system was not in the scope of this work.

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