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ANALYSIS OF THE BEARINGS' INFLUENCE ON THE DYNAMIC BEHAVIOR OF A ROTATING MACHINE

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Abstract. *This work is about the analysis of dynamic behavior of rotating machines employing the finite element method. It aims on evaluating the bearings model influence in the rotor's performance. A computational code - that allows testing several simplifying hypotheses such as bearing model simplification - was written in order to do all the tests. Shaft and disc elements are modeled based on Timoshenko beam theory. Bearings are modeled with both rigid and hydrodynamic options. The hydrodynamic bearing is modeled by the solution of Reynolds equation for short bearing, in order to obtain approximations of damping and stiffness coefficients that represent the oil film. Analyzes are done in frequency domain (FRF), time domain (orbits) and modal domain (Campbell diagram). Two types of rotors are studied: rotor with centralized and decentralized disc - supported at both ends. The results showed the considerable effect of bearing modeling on the dynamic behavior of the rotor, by changing the natural frequencies, orbits dimensions and the vibration amplitudes.*

Keywords: *Rotordynamics, Hydrodynamic Bearings, Vibration Analysis, Finite Element Method*

1. INTRODUCTION

With increasing prominence in the industry and hence in the academic world, vibration analysis of rotating machines is driven by the growing variety of systems in which they can be employed. By having usually large dimensions and high complexity, their production is expensive, emphasizing the concomitant importance of their modeling and optimization, reducing, in this way, many operational risks.

The study of rotating machines has aroused the interest of many researchers in the field of machines and structures due to the significant amount of typical phenomena that occur during their operation. This type of rotating system represents the largest and most important class of machinery, used for transport of fluids, machining and conformation of metals, power generation, naval and aeronautical propulsion, among many other applications. The existence of a rotating component supported on bearings and transmitting power creates a family of problems, found in the most diverse machines: compressors, pumps, motors, and turbines in a large range of sizes.

Dynamic analysis of rotating machines involves many parameters and therefore must take into account the interaction between components, such as bearings. Consequently, a critical issue in rotor dynamics is the modeling of the bearings, which directly affects the dynamic behavior and stability of the rotating system.

In order to obtain the characteristics of a rotating system, there are several approaches. Among these, is noteworthy the finite element method, since it presents relative ease of implementation and has good scalability, allowing increase of algorithms' complexity according to the need of precision. In this context, this work presents an analysis of the effect of the characteristics of the bearings on the dynamic behavior of rotors, being the computational part developed through the finite element method in an algorithm implemented in MATLAB®.

2. METHODOLOGY

Modeling a rotating system involves the analysis of its components such as shafts, bearings, discs, foundation, and their interaction with each other. The numeric analysis of the rotor model, in this work, uses the finite element method (FEM). Therefore, each component, and its respective dynamic response, must be discretized taking into account the interaction with the adjacent nodes.

2.1 Finite Element Method

Analytical solutions for complex equations are computationally costly and often not even possible with currently available algorithms. The impossibility of solving a system through mathematical manipulation may require a simplification of the model to make an analytical solution possible. However, simplification will lead to loss of accuracy. Since the accuracy of the answer constitutes a great advantage of the analytical method, the simplification of a problem to solve it analytically is contradictory. Thus, the most feasible way to perform the analysis of systems with several coupled equations is through numerical methods.

To solve engineering problems, several discretization methods can be used, such as the Finite Element Method, Finite Differences Method, Finite Volume Method, among many others. In the analysis of rotors, the most used method is the finite element method (FEM). Due to its dissemination in this area, this method was chosen to be used in this work.

The main assumption of the method is dividing the continuum into a finite number of elements, which can be characterized by a finite number of parameters. The solution of the complete system as a set formed by its elements follows the same impositions applied to general discretized problems.

The model used in the simulations of this work consists of a rotating system with typical elements such as discs, shaft elements and bearings, as shown in Fig. 1, along with the coordinate systems used to describe the movement of the system (local and inertial).

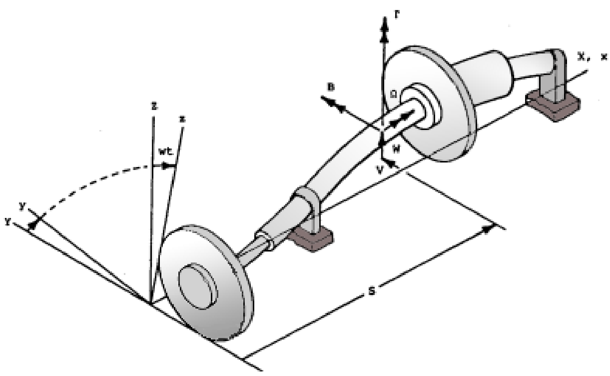

Figure 1: Typical rotor configuration (Adapted from Nelson and McVaugh, 1976)

The complete rotor model includes the modeling of the rotor shaft, the bearings that connect it to the foundation structure and the foundation itself. For this work, the foundation is considered rigid; the shaft is modeled by Timoshenko's beam and disc elements, as presented by Nelson and McVaugh, 1976; and the bearings is modeled in two ways: either simulating a bi-clamped shaft and a shaft supported by hydrodynamic bearings.

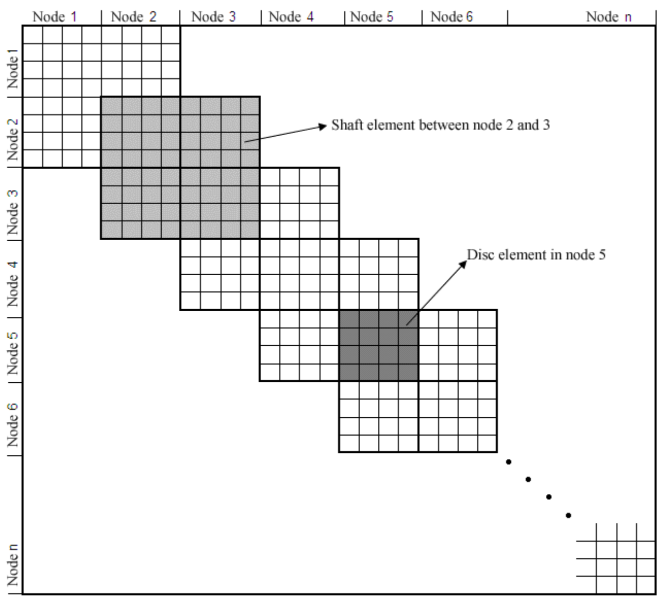

Figure 2: Elements' matrices superposition (Machado, 2014)

Thus, it is possible to obtain the global equations of the system with each modeled component. The matrices of each element are grouped in a global matrix and their positions in the global matrices are related to degrees of freedom, as shown in Fig. 2.

2.2 Bearings Modeling

The bearing is responsible for restricting the displacement of the shaft in certain directions, as well as reducing friction in the directions in which the movement is free. Among the various types of bearings, the most used are: ball bearings, magnetic, dry friction and hydrodynamic. Since hydrodynamic bearings is widely used in rotating machinery, this work is concentrated in type of bearing.

For the purposes of comparison with the hydrodynamic bearing, and due to the simplicity offered by the model, the idealized bi-clamped bearing is also used in the simulations. It offers infinite stiffness in all directions, allowing only the rotation in the axis parallel to the rotor.

For the hydrodynamic bearing, its behavior is obtained through the Reynolds equation. Since this complete equation does not present an analytical solution, and even the complete numerical solution presents high computational cost, the use of simplifying hypothesis is necessary to make a solution feasible, albeit it lowers the precision of the results (Daniel, *et al*.,2016).

Equivalent springs and dampers characterize the hydrodynamic bearings. Their equivalent coefficients of stiffness and damping can be inserted directly into the appropriate elements (nodes) of the respective matrices for both the time domain and the frequency domain analyzes. This approximation is equivalent to a model of shaft supported on springs and dampers, as shown in Fig. 3. For this work, the expressions for the coefficients of stiffness and damping are those given by Krämer, 1993.

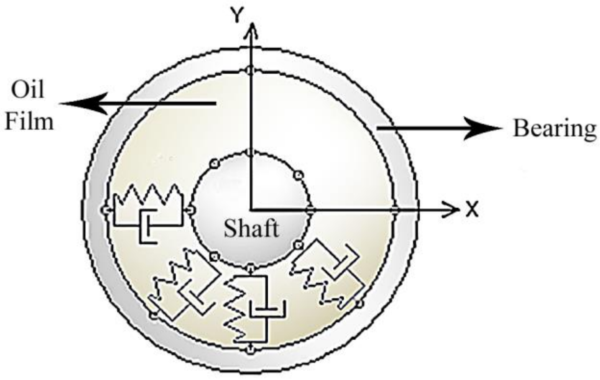

Figure 3: Hydrodynamic bearing, with oil film modeled as springs and dampers (adapted from Machado and Kavalca, 2009)

2.3 Computational Algorithm

The computational algorithm was written in the form of a main program divided into blocks and auxiliary functions. The use of blocks is encouraged by the recognition of blocks by MATLAB® as parts of the code that can be executed one-*by*-one easily. In addition, independent block construction favors parallel processing of the blocks, taking advantage of the increasingly common processing parallelism. The general functioning of the algorithm is shown in the diagram of Fig. 4. The block of inputs and initial calculations is the least complex and runs in a very short time. The block of matrix assembly presents the most complex calculations of the code, but since they are performed only once, they take only a fraction of a second. The following blocks are already independent and can be executed in parallel.

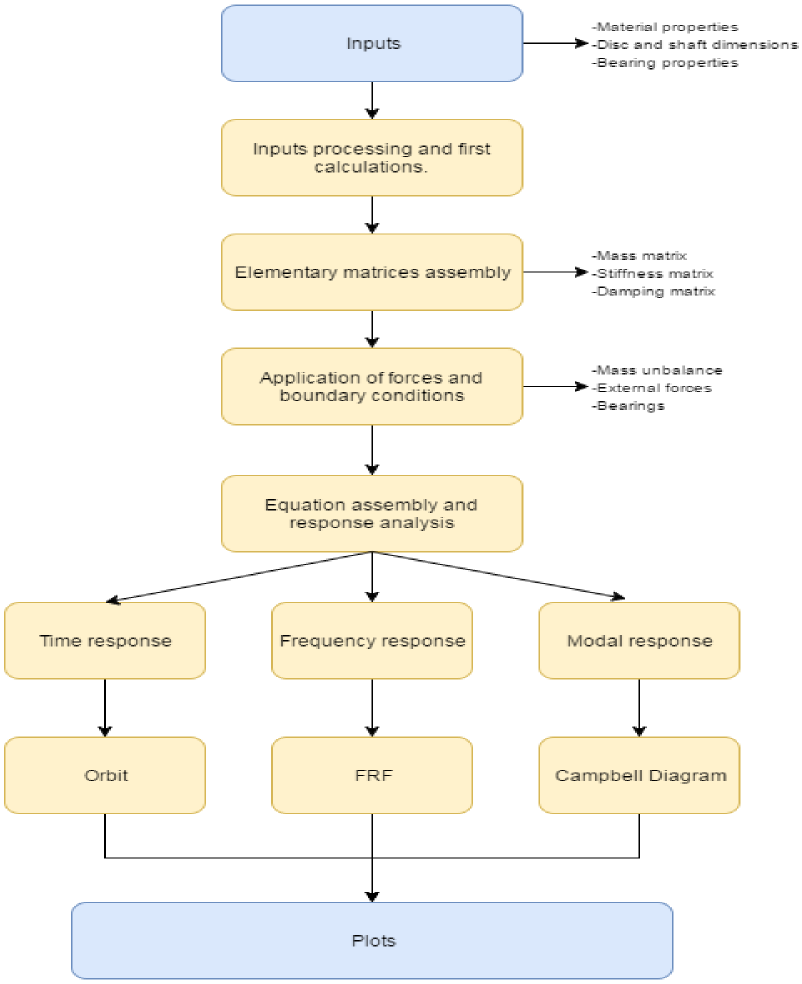

Figure 4: Algorithm's workflow chart

3. RESULTS AND DISCUSION

For the simulations, a model with the dimensions similar to the real rotor present in the Laboratory of Rotary Machinery (LAMAR-FEM-UNICAMP) was used, in order to provide an experimental verification of the results in the future. The dimensions of the real rotor are similar to those of Fig. 5, while the model of Fig. 6 is used to simulate the condition with the decentralized disc.

For each of the cases will be presented and analyzed results in time domain - disc node orbits; in frequency domain - disc node FRF due to mass eccentricity and in modal domain - Campbell's diagram. For the time domain analysis, two frequencies will be shown: one before and the other after the crossing of the forward whirling line in Campbell's diagram. This way it is possible to visualize the change in whirling direction.

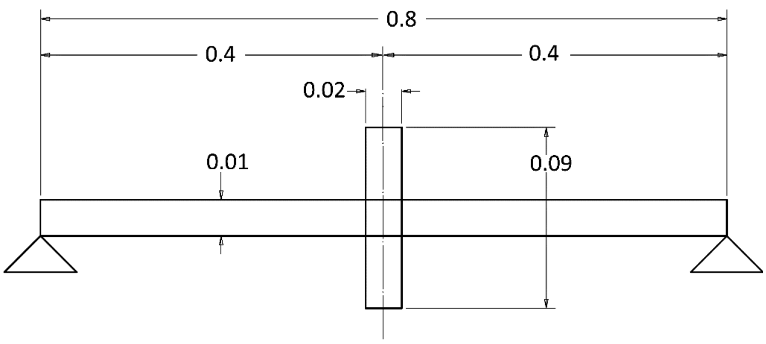

Figure 5: Centralized disc rotor (dimensions in meters, out of scale for best visualization)

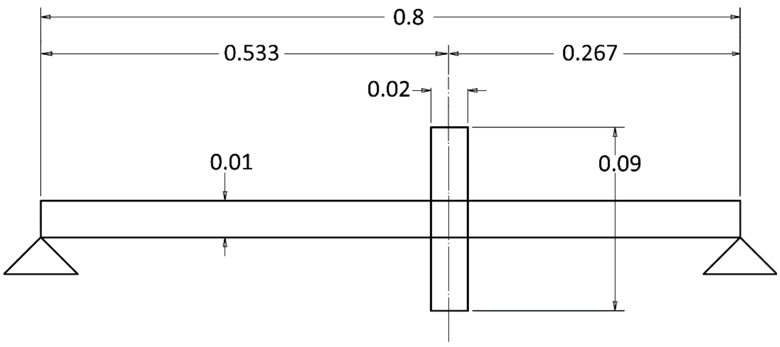

Figure 6: Decentralized disc rotor (dimensions in meters, out of scale for best visualization)

3.1 Bi-Clamped Rotor

3.1.1 Centralized Disc

Analysis starts with the centralized disc bi-clamped rotor. It can be seen in Fig. 7 that the first three modes of vibration - corresponding to the first three natural frequencies of the system – can be seen in the Campbell diagram. There is only distinction in forward and backward whirl frequencies in the second natural frequency. This behavior is due to disc in the center of the axis. Therefore, for the odd modes, the movement is restricted to translation in a plane perpendicular to the z-axis and there is no influence of the gyroscopic effect. In the even modes, there is an inverse behavior - there is no translation movement, only rotation in an axis perpendicular to the z-axis, which explains the absence of the peak of the second natural frequency in the FRF and the frequencies of whirling are distinguishable for forward and backward whirl. In Fig. 8 is visible the change of whirling direction as the rotor frequency crosses the forward whirl frequency of the second mode.

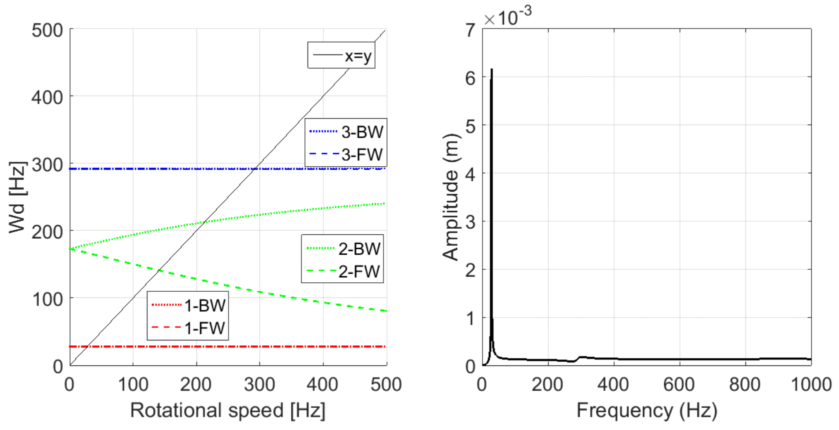

Figure 7: Campbell diagram (left) and FRF (right) for centralized disc clamped rotor

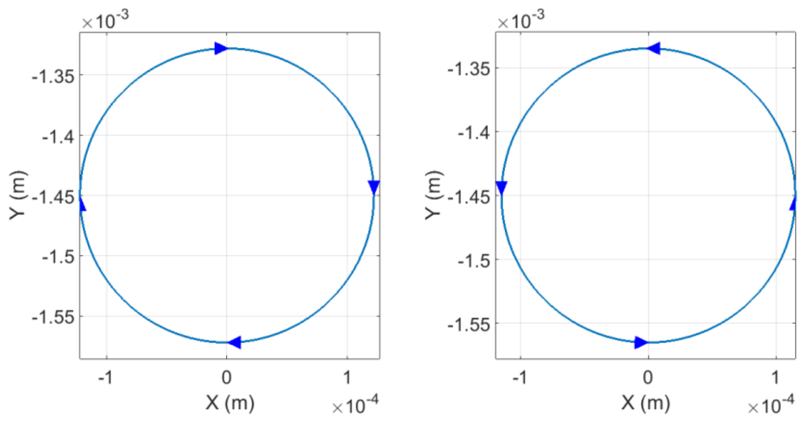

Figure 8: Rotor orbits for 110 Hz (left) and 160 Hz (right)

3.1.2 Decentralized Disc

Differing from the centralized disc rotor, where the forward and backward whirling frequencies are the same for the odd modes, the decentralized rotor, because of its geometry, exhibits distinguishable whirling frequencies for both odd and even modes of vibration. In spite of this separation cannot be seen in the first natural frequency, because of the algorithm's low frequency resolution, starting from the third natural frequency its effect is clear. Because for the decentralized rotor forward and backward whirling frequencies are distinguishable, Fig. 10 shows the reversion of whirling direction as the system crosses the third mode forward whirling frequency.

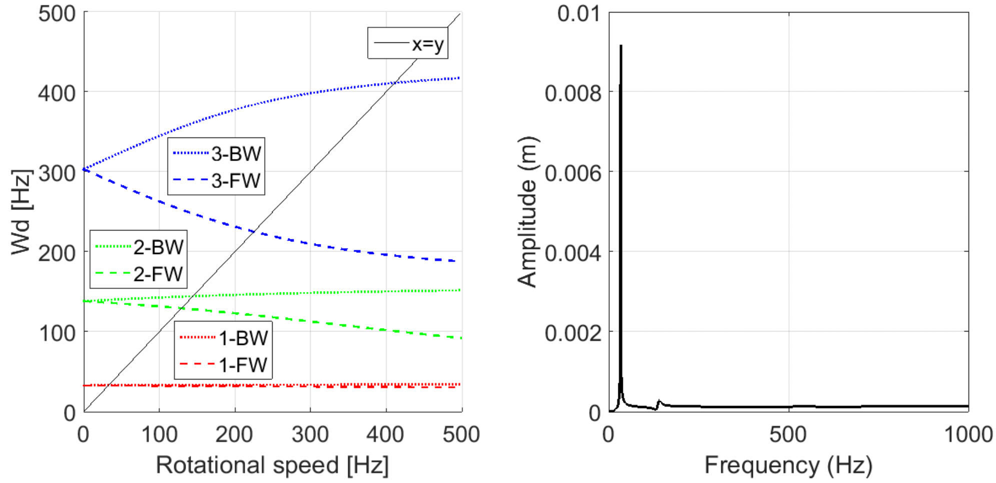

Figure 9: Campbell diagram (left) and FRF (right) for decentralized disc clamped rotor

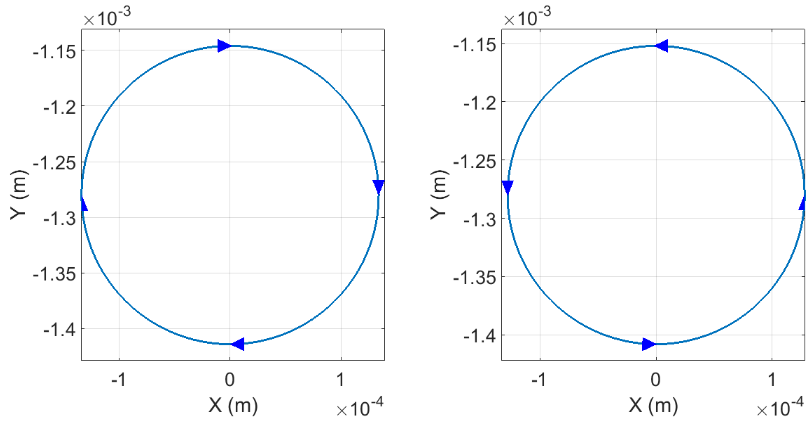

Figure 10: Rotor orbits for 200 Hz (left) and 240 Hz (right)

3.2 Rotor with Hydrodynamic Bearings

3.2.1 Centralized Disc

Through analysis of the results shown in Fig. 11, adding hydrodynamic bearings to the model lowered the natural frequency, as expected, since the oil film reduces the overall stiffness of the system without changing its mass, resulting in a lower natural frequency.

Besides the reduction in the natural frequency, is noteworthy the reduction in the peak amplitude to less than a half of the peak encountered in the clamped rotor for the centralized configuration. There is now a peak in response related to the second mode of vibration that was negligible in the clamped rotor configuration, but can affect the behavior of the rotor supported by hydrodynamic bearings. Comparing Fig. 12 with Fig. 8 it can be noticed that orbits' dimensions were reduced due to part of vibration energy being dissipated on the bearings.

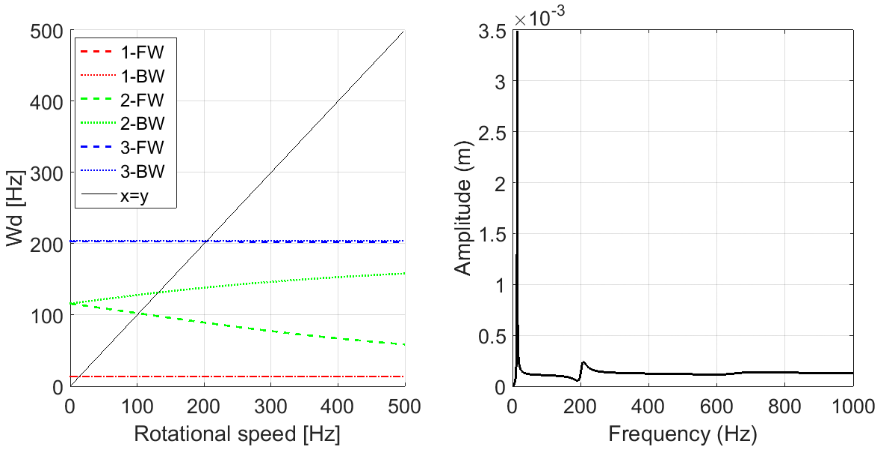

Figure 11: Campbell diagram (left) and FRF (right) for centralized disc rotor supported on hydrodynamic bearings

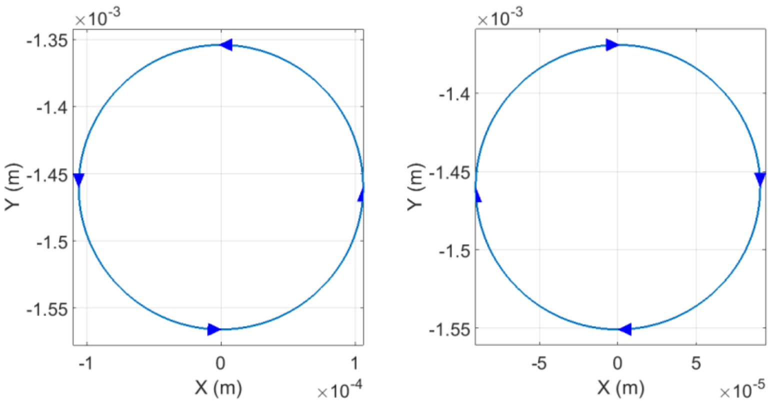

Figure 12: Rotor orbits for 110 Hz (left) and 160 Hz (right)

3.2.2 Decentralized Disc

The decentralized disc showed a similar behavior to the centralized disc when hydrodynamic bearing were added to the model. The peak of amplitude in the first mode, shown in Fig. 13, was reduced to half, compared to the peak in Fig. 9. However, unlike was observed in the centralized disc rotor, the higher modes of vibration have no increase in vibration amplitudes. It is noteworthy the reduced split between the forward and backward whirling frequency for the third mode of vibration compared to the bi-clamped rotor. Because of this, the orbits on Fig. 14 could not be obtained in the same frequencies of the Fig. 10 and still show the reversion of whirling direction.

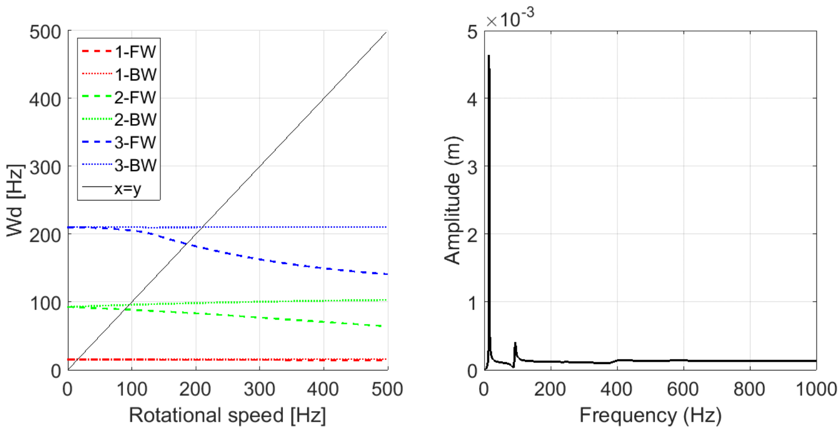

Figure 13: Campbell diagram (left) and FRF (right) for decentralized disc rotor supported on hydrodynamic bearings

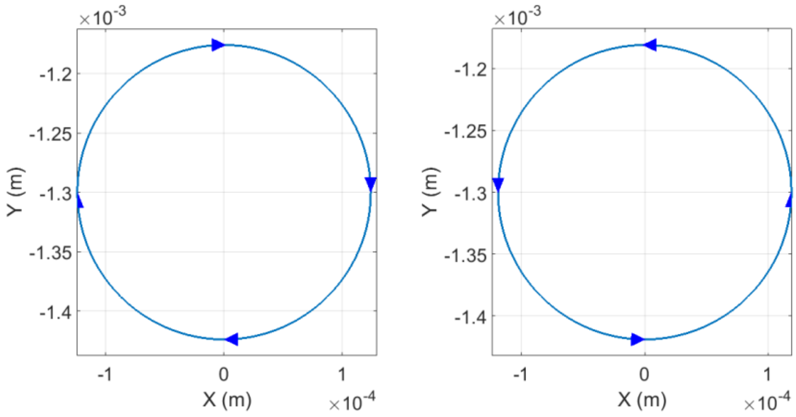

Figure 14: Rotor orbits for 160 Hz (left) and 200 Hz (right)

4. CONCLUSIONS

For the analyzed rotor configurations, the replacement of the rigid bearing (bi-clamped rotor) by the hydrodynamic bearing presented an effect of reduction in the natural frequencies, since it reduced the stiffness of the system without any change of mass. The main change occurred in the first natural frequency, which was reduced to approximately half with the addition of hydrodynamic bearings. It is notable that the presence of hydrodynamic bearings causes a reduction in maximum amplitude of the FRF peaks while widening them. In the Campbell diagrams, besides of the reduction in natural frequencies, the rotors with hydrodynamic bearings displayed less spread between the whirl frequencies of the same mode. This behavior can be explained by the decrease of vibration of the rotor with the inclusion of the hydrodynamic bearings. As the bearings dissipate part of this vibration, the rotor, in particular in the disc position, has a lower vibration, and consequently, there is a decrease in the gyroscopic effect, causing the backward and forward frequencies to be less dispersed.

With what was presented, it can be inferred that there is a great influence of the bearings in the total damping of the system, showing the considerable effect of bearing modeling on the dynamic behavior of the rotor and justifying the need for a careful analysis of this component of the rotating system. Following the results of this paper, future research will be made using this methods and topological optimization to design shafts that can compensate bearings compliance and/or foundation structure influence with minimum mass increase.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Daniel, G.B, Machado, T.H., Cavalca, K.L. – Investigation On The Influence Of The Cavitation Boundaries On The Dynamic Behavior Of Planar Mechanical System With Hydrodynamic Bearings – Mechanism and Machine Theory 99 p. 19-36, 2016.

Krämer, E. – Dynamics of rotors and foundations", Springer-Verlag, New York, 1993.

Machado, T.H. Identificação do desgaste em mancais hidrodinâmicos através do efeito de anisotropia. Campinas: Faculdade de Engenharia Mecânica, Universidade Estadual de Campinas, 2014.

Machado, T.H.; Cavalca, K.L. – Evaluation of dynamic coefficients for fluid journal bearings with different geometries. In: 20TH INTERNATIONAL CONGRESS OF MECHANICAL ENGINEERING - COBEM 2009. Proceedings of the 20th International Congress of Mechanical Engineering. Rio de Janeiro: ABCM. v. 1. p. 1-11, 2009.

Nelson, F.C., McVaugh J.M. – The dynamics of rotor bearing systems using finite elements. Journal of Engineering for Industry, Vol. 98, p. 593-600, 1976

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