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**EVALUATION OF ROLLING BEARING STIFFNESS TO IMPROVE
CRITICAL VELOCITY CALCULATION**

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Abstract. *Electric motors play a prominent role in modern industrial production. By converting electrical energy into mechanical energy, they enable the operation of most of equipment present in manufacturing processes. Despite the numerous standard lines available on the market, many customers have special applications, which must be analyzed in detail. One of the steps present in the customized design of these machines is the calculation of the critical velocity of the rotor, which includes the shaft, the set of plates and the other components it supports. One of the parameters with great influence on the mathematical model used to predict the resonance frequencies of the rotor is the stiffness of the bearings, especially the radial one. However, there is still no consensus in the literature as to how to determine this value. Although the finite element method has gained great prominence in recent years, it is still very costly in terms of setup and computational effort. In order to increase the accuracy of this model, this paper evaluates a method to estimate the stiffness of rolling bearings from an analytical equation, which considers geometric aspects of the bearing and the load applied to it. The stiffness values obtained with this equation were compared with those provided by the main bearing manufacturers, which use different methods to calculate the stiffness, for a series of cases with different boundary conditions. Furthermore, experimental tests were conducted in order to validate the stiffness of bearings used in a case study. To carry out that, a run-up test was performed, correlating the machine vibration velocity with the rotor rotation frequency, which allows the estimation of its critical velocity. It is concluded, from the analyzed data, that the proposed equation underestimates the stiffness values of the analyzed bearings. In this case, the model used indicates a critical velocity value lower than that measured experimentally*

Keywords: *Mechanical vibration. Rotor dynamics. Critical velocity. Rolling bearing stiffness.*

1. INTRODUCTION

Currently, it is estimated that more than 300 million electric motors are used globally in the infrastructure, large buildings and industrial sectors (“Top 20 Electric Motor Manufacturers in 2019”, 2019). And that number tends to grow at a rapid pace. More than 30 million electric motors are sold annually only for the industrial sector. No wonder they are responsible for most of the energy consumed in factories, being used to operate the most diverse equipment, from machine tools to devices for lifting and moving loads. In fact, the systems driven by them, such as pumps, fans and compressors, represent around 30% of global electricity demand (INTERNATIONAL ENERGY AGENCY, 2019).

The biggest increase in electricity consumption is likely to come from the development of developing countries, as their per capita income is expected to increase, as is the consumption of household appliances. Thus, Africa and India are expected to be the main drivers of growth in the global electric motor market, which is expected to reach a value of USD 136 billion in 2025, with a compound annual growth rate of 4.5% (“Electric Motor Market Size, 2025”, 2018).

To meet this high demand for products and still meet increasingly high levels of efficiency, it is necessary to make sure that the design of an electric motor is adequate. Although there are already standardized motor lines, with numerous options available, many customers request customized designs for their application. They require, for example, that the electric machine operates in rotation continuous superiority of the catalog, as the use of a speed multiplier is not adequate in some situations, such as those in which there is a need for low maintenance or a reduced number of components, for example. Another practical case is the request for a product with reduced dimensions, mainly when the motor must replace another unit that is already operating in a system with strict restrictions.

Thus, when it comes to mechanical aspects, the most common specialties are related to: rotation frequency, applied load, operating condition, degree of sealing and dimensioning of components. In many cases, to speed up the process and standardize the format of the provided solution, analysts usually follow a standardized routine for checking these items.

The rotation frequency of a three-phase electric motor depends on its polarity and operating frequency. If the requested rotation is higher than the values available standardized motors, a series of aspects must be analyzed, such as: bearing rotation limit, grease viscosity factor, tensions generated in the rotor set due to centrifugal force and so on.

Either way, one of the first factors to consider is the rotor critical velocity. As every mechanical part, it also has a set of natural frequencies, which are determined by its geometry, composition and supports. When its rotation frequency coincides with one of its natural frequencies, small disturbances can produce intense vibrations due to the phenomenon of resonance. In this case, the disturbance is the residual mass of the rotor, inherent to its manufacturing process, and the high vibrations can lead to the failure of components, mainly the roller bearings.

To avoid such situation, the separation margin of the first resonant frequency of the rotor in relation to its operating frequency must be analyzed. For this, however, it is necessary to evaluate the rotor vibration modes, and one of the boundary conditions of this calculation is the bearing stiffness matrix.

In this paper, an evaluation of the stiffness of the rotor bearings will be carried out in order to improve the calculation of its critical velocity. To delimit the scope of the project, only two types of bearings are considered: deep groove ball bearings and cylindrical roller bearings.

2. NUMERICAL MODEL

This section describes the mathematical model used to calculate the natural frequencies of the rotor and the necessary theory to calculate the value of the rotor critical velocity. In addition to its most important considerations, the equation defined to estimate the stiffness of the bearings, which is one of the parameters with great influence on the desired result.

2.1 Dynamics of the Rotor

To describe the dynamic behavior of the rotor, this paper uses the standard model of Lalanne and Ferraris (1998), which was implemented in a routine in the PTC Mathcad 15 software. The main considerations of the method are described in this section.

In industrial motors, the shafts typically consist of a series of uniformly sized circular sections. The shaft serves as the primary structural element of the rotor, supporting the other components. Figure 1 shows an example of a rotor being analyzed in the horizontal position. The shaft is shown in light gray, the sheet metal packaging is shown in green, the electric motor's external fan is shown in orange, and the bearings are shown in dark gray triangles.

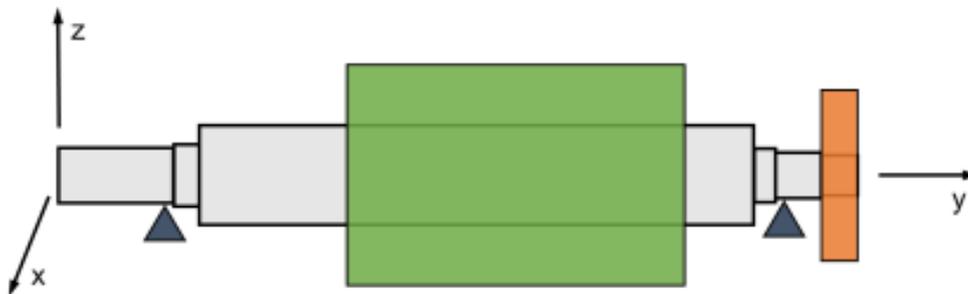


Figure 1 - Rotor example to be analyzed by the model.

By applying the Lagrange equations to a single rotor, such as the one shown in Figure 1, the overall system equation can be written as

$$[M]\{\delta(\dot{t})\} + [C]\{\delta(t)\} + [K]\{\delta(t)\} = \{F(t)\}, \quad (1)$$

where δ is the vector containing the displacement of each finite element, M is the symmetric mass matrix, C is the asymmetric damping matrix, K is the stiffness matrix (usually asymmetric), and F is the force vector.

The mass matrix M is obtained by superposing the mass matrices of the beam elements, formed by two adjacent discrete elements, and the concentrated masses (disks), if any. It also considers a shaft rotation inertia. The damping matrix C is composed of a gyroscopic antisymmetric matrix, defined as a function of the rotation frequency ω and a rolling damping matrix, normally asymmetric. The gyroscopic or Coriolis matrix considers the inertia distributed along the axis of rotation. In most cases, it is customary to consider the damping of the bearings equal to zero. The stiffness matrix K is usually asymmetric due to bearing characteristics. The Timoshenko-Ehrenfest beam model is used to represent the rotor axis, which considers rotational inertia and shear effects (HAN; BENAROYA; WEI, 1999). The axis intensity is influenced by the method of influence coefficients.

Applying the boundary conditions and solving Eq. (1), it is possible to find the eigenvalues of the Eq. (1), being

$$r_i = -\frac{\xi_i \omega_i}{\sqrt{1 - \xi_i^2}} \pm j \omega_i, \quad (2)$$

where $j = \sqrt{-1}$, ω_i is the frequency and ξ_i is the viscous damping factor. More details about the model can be found in Möller (2021).

2.2 Rolling Bearing Stiffness

The stiffness of the bearings, mainly the radial one, is a parameter of great influence in the model. Thus, its value must be carefully defined, as it can lead to the approval or rejection of a project.

The model developed for the rotor has 4 degrees of freedom: the rotor can translate in the x and z directions (radial) and rotate in the x and z directions. The roller bearing model has the same degrees of freedom. Figure 2, presents a view of the main stiffness and damping components of the bearing, as well as the coordinates of the system.

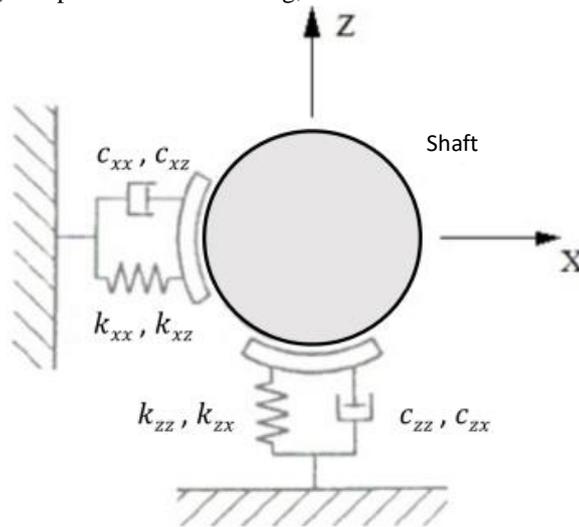


Figure 2 - Degrees of freedom of the rolling bearing.

It is estimated that the viscous damping factor, ξ , of a bearing is less than 0.01 (KRÄMER, 1993). Therefore, the damping matrix is assumed to be completely zero. Thus, as the shaft is symmetric in all directions, the stiffness matrix can be expressed as

$$K_m = \begin{bmatrix} k_r & 0 & 0 & 0 \\ 0 & k_r & 0 & 0 \\ 0 & 0 & k_{\theta r} & 0 \\ 0 & 0 & 0 & k_{\theta r} \end{bmatrix}. \quad (3)$$

However, the angular stiffness of the bearing is usually much less relevant than the radial stiffness; and it is not trivial to estimate its value. Thus, it is customary to neglect the term $k_{\theta r}$.

2.3 Proposed Equation for Bearing Stiffness

To estimate the radial stiffness of the bearings, the equation developed by Gargiulo (1980) is used. The proposed method is relatively simple and depends only on the geometric characteristics of the bearing, in addition to the load applied to it.

It can be used to estimate bearing stiffness in low and medium velocity applications that do not require high accuracy. Despite its restrictions, the permission proposed by this author is often used as benchmarking to verify the results of other methods.

Although Gargiulo (1980) also presents approval for estimating the axial compression of bearings, this is not used, as the model denies this degree of freedom. For a ball bearing, the radial displacement, δ_r , in meters, is given by

$$\delta_r = C_1 \left[\frac{F_r^2}{D \times Z^2 \times \cos^5(\alpha)} \right]^{1/3}; \quad C_1 = 1.28 \times 10^{-7}. \quad (4)$$

where F_r represents the radial force applied to the bearing in newtons, D indicates the diameter of the rolling elements in meters, Z indicates the number of rolling elements and α represents the contact angle of the rolling elements with the bearing races in radians. Algebraically manipulating the Eq. (4), the radial force F_r can be expressed as

$$F_r = C_1^{-3/2} \delta_r^{3/2} D^{1/2} Z \cos^{5/2}(\alpha). \quad (5)$$

The radial stiffness can be expressed as

$$k_r = \frac{\partial F_r}{\partial \delta_r} = \frac{3}{2} C_1^{-3/2} \delta_r^{1/2} D^{1/2} Z \cos^{5/2}(\alpha). \quad (6)$$

Thus, the radial stiffness is a function of the displacement. However, it is customary to know the value of the radial force applied to the bearing. Therefore, substituting Eq. (4) in Eq. (6), the stiffness for a ball bearing, in newton per meter, is expressed as

$$k_r = 1.18 \times 10^7 [D \times F_r \times Z^2 \times \cos^5(\alpha)]^{1/3}. \quad (7)$$

Analogously, the equation for cylindrical roller bearings can be obtained. The radial stiffness for this type of bearing, in newton per meter, is given by

$$k_r = 8.55 \times 10^8 [F_r^{10} \times Z^{10} \times l^4 \times \cos^{19}(\alpha)]. \quad (8)$$

where l is the effective length of the rolling elements in meters.

2.4 Sensitivity Analysis

This section discusses the sensitivity of the parameters used in the proposed equation for the radial displacement and for the radial stiffness of deep groove ball bearings, since this is the type of bearing most used in industrial electric motors.

Bearing model 6205 is defined as a reference for the analyzes carried out. The geometry data considered for this part can be found in Möller (2021). All bearing parameters remain constant, except the variable analyzed in each case.

The radial displacement of the inner race of the bearing is defined by Eq. (4). The radial displacement curve as a function of the radial load applied to the part is shown in Figure 3.

The slope of the curve decreases as the radial load increases, which is explained by the exponent of this parameter. Thus, the greater the radial load applied, the greater the increment required to obtain the same radial displacement.

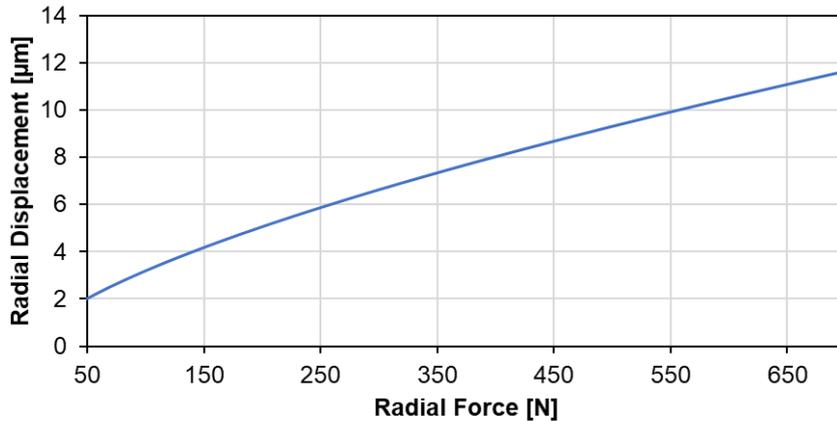


Figure 3 - Radial displacement as a function of the radial force.

Equation (7) describes the behavior of the radial stiffness of the bearing as a function of the radial load, which is shown in Figure 4. As the radial load applied to the bearing increases, its radial stiffness also increases.

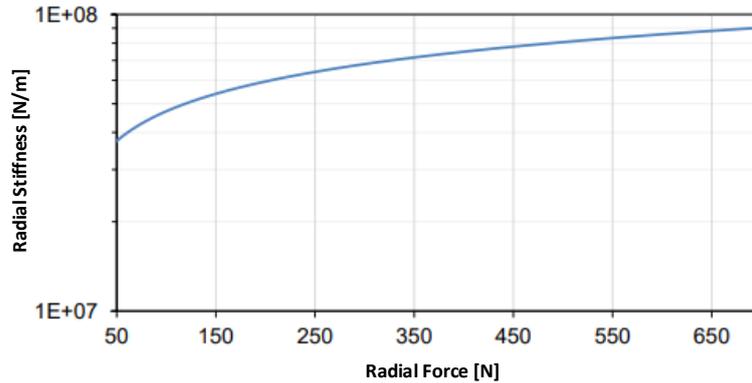


Figure 4 - Radial stiffness as a function of the radial force.

The influence of the contact angle on the radial stiffness of the bearing can also be analyzed by Eq. (7). As the contact angle increases, the radial stiffness of the bearing decreases. Figure 5 presents this relationship visually.

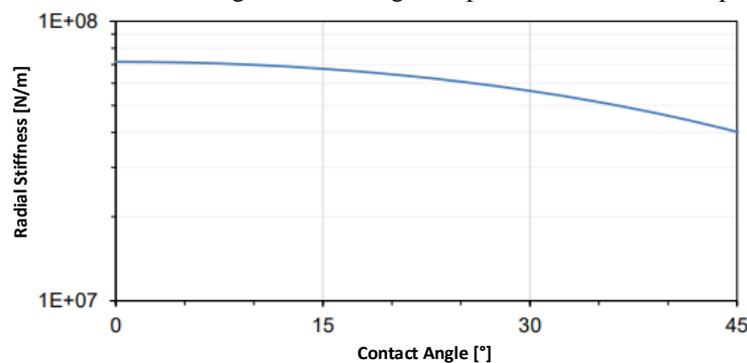


Figure 5 - Radial stiffness as a function of the contact angle.

3. EXPERIMENTAL TEST BENCH

To estimate the critical velocity of a rotor based on experimental data, a run-up test was carried out. Table 1 presents a correlation between the vibration velocity of an equipment and its operating zone. This can be seen as a recommendation as to acceptable vibration limits for a machine. Class I refers, in the case of electric motors, to devices with up to 15 kW of power; while class II consists of machines of intermediate size, electric motors with powers between 15 kW and 75 kW.

Table 1 - Qualitative assessment in relation to the operating zone of a machine.

Vibration Velocity RMS $\times 10^{-3}$ [m/s]	Class I	Class II
0.71	A	A
1.12	B	A
1.8	B	B
2.8	C	B
4.5	C	C
7.1	D	C
11.2	D	D

Source: Adapted from International Organization for Standardization - ISO (2016).

As far as zones are concerned, zone A indicates the expected vibration values for newly installed machines; equipment that vibrates in zone B is considered capable of operating without restrictions for long periods; machines that vibrate in zone C must operate in this condition only for a limited period; finally, vibration values within zone D are considered severe enough to cause damage to the device.

The run-up test basically consists of exciting the natural frequencies of a rotating machine from its own vibration. The excitation frequency of the system is equal to the rotation of the shaft. In this way, as the rotor increases its rotation frequency, the vibration resulting from the rotation of the shaft sweeps the frequencies of interest and thus excites the natural frequencies of the component.

The experimental bench for carrying out this test is mainly composed of the following components: electric motor, frequency inverter (WEG CFM500), vibration analyzer (COMMTTEST INSTRUMENTS VB1000), accelerometer and fan. Figure 6 shows the arrangement of these components.

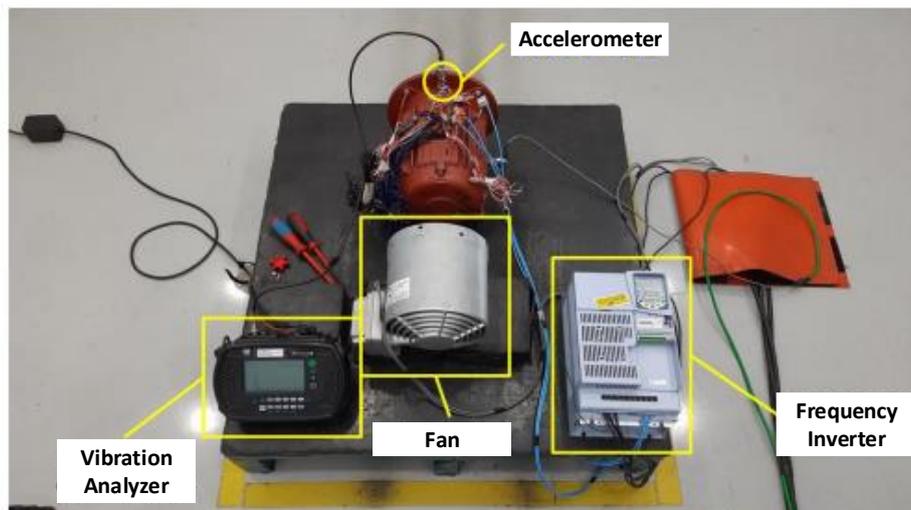


Figure 6 – Run-up test bench.

4. OBTAINED RESULTS

This section describes the main obtained results. The results provided by the proposed equations in Section 2.3 are compared with the values provided by the main bearing manufacturers for different boundary conditions. Furthermore, the experimental data obtained through the tests carried out are analyzed and discussed in a case study, in which the critical velocity foreseen in the project is compared with that calculated experimentally.

The bearing models and boundary conditions shown in Table 2 are used for benchmarking. Models are selected based on their use in electric motors. Therefore, choose to evaluate the most commercially available 62, 63 and NU series options.

Table 2 - Rolling models and boundary conditions defined for benchmarking.

Bearing	Radial Force [N] I	Axial Force [N]
60003	70	40
6205	700	250
6214	3100	700
6218	4800	900
6312	4100	600
6314	5200	700
6319	7650	950
NU-212	4875	0
NU-314	7900	0
NU-322	22500	0

Since radial stiffness has the greatest impact on a rotor's critical velocity, this comparison is especially about it. The radial stiffness values provided by each manufacturer for the reported boundary conditions are shown in Table 3. The Reference column presents the results obtained by the equations presented in Section 2.3. The internal geometry parameters used can be found in Möller (2021). In order to preserve the identity of the contacted manufacturers, they are identified as A, B, C and D.

Table 3 - Radial stiffness calculated by each manufacturer [N/m].

Bearing	Reference	A	B	C	D
60003	3.76×10^7	3.76×10^7	5.89×10^7	6.29×10^7	9.04×10^6
6205	9.02×10^7	9.00×10^7	1.60×10^8	1.62×10^8	7.66×10^7
6214	2.17×10^8	2.06×10^8	3.63×10^8	3.76×10^8	8.84×10^7
6218	2.73×10^8	2.59×10^8	4.24×10^8	4.35×10^8	1.22×10^8
6312	2.14×10^8	2.11×10^8	3.33×10^8	3.28×10^8	1.11×10^8
6314	2.40×10^8	2.39×10^8	3.57×10^8	3.67×10^8	1.43×10^8
6319	2.99×10^8	2.98×10^8	4.02×10^8	5.08×10^8	1.88×10^8
NU-212	7.97×10^8	7.06×10^8	6.00×10^8	5.74×10^8	6.34×10^8
NU-314	1.07×10^9	9.12×10^8	8.09×10^8	7.54×10^8	9.43×10^8
NU-322	1.67×10^9	1.58×10^9	1.40×10^9	1.32×10^9	1.47×10^9

Figure 7 presents the radial stiffness calculated by each manufacturer for bearings of series 62, 63 and NU. For the 62 and 63 series, the results of manufacturer A are very close to those of the Reference. The values of manufacturers B and C are very similar for most of the analyzed cases and represent the highest estimates. The radial stiffness calculated from the curves provided by the D manufacturer is below the average of its peers. For the NU series, manufacturers B and C have the lowest values of radial resistance. Manufacturer D is average among its peers. Manufacturer A indicates values close to the Reference, but between 5% and 15% lower.

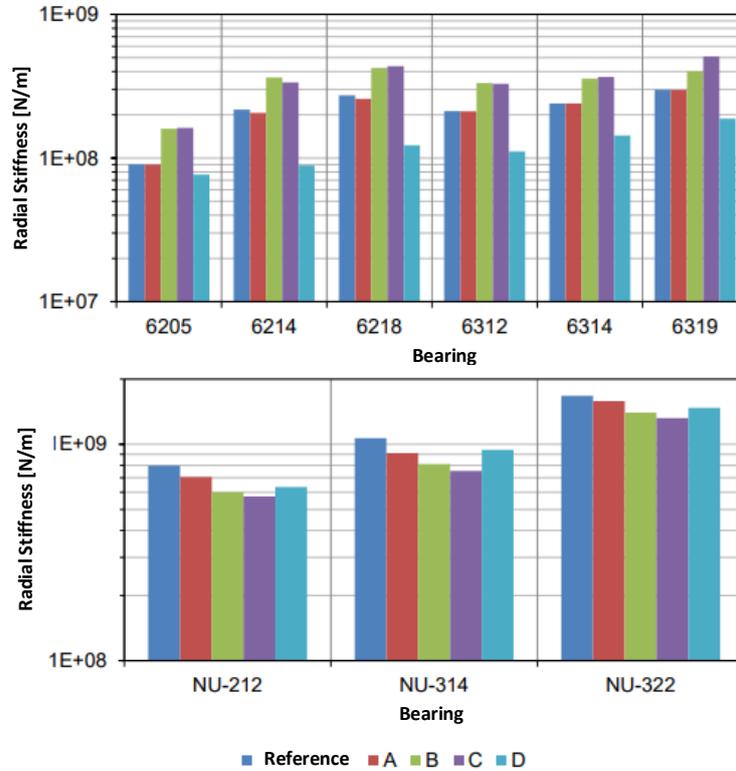


Figure 7 - Radial stiffness for the 62, 63 and NU bearing series.

In order to validate the developed model in a practical way, the present case study is carried out, comparing the predicted results in the project with the experimental ones. To analyze the dynamic behavior of a rotor, two electric motors with the same characteristics are used. The two machines have the same specifications, with the exception of the grease used in the bearings. Shaft geometry, plate pack configuration and bearing design are the same for both cases.

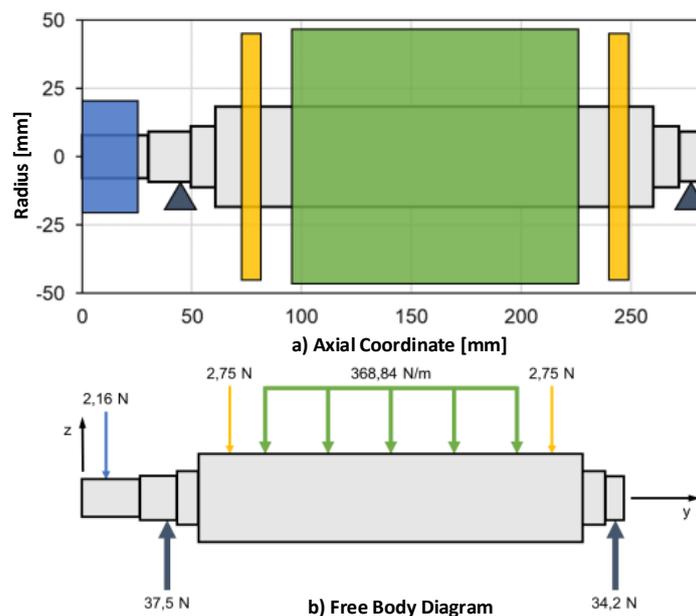


Figure 8 - Representation of the shaft in study. a) Components distribution; b) Free Body Diagram.

The geometry of the analyzed shaft and the free-body diagram are presented in Figure 8, which shows the length of each bar section, in addition to its respective diameter. The components mounted on the shaft also have their positions and shapes indicated, all of which have a circular section. The element in blue represents the coupling used on the motor shaft end, necessary for other tests. The parts in yellow correspond to the balancing discs. The set of plates is represented by the part in green. Finally, bearings are indicated by triangular brackets in gray. More details about the electric motors used can be found in Möller (2021).

To analyze the sensitivity of the radial stiffness for the case under study, Figure 9 presents the critical velocity as a function of the radial stiffness of the bearings. The 1st and 2nd vibration mode are represented. In this analysis, it is considered that both bearings have the same stiffness, which is exclusively radial.

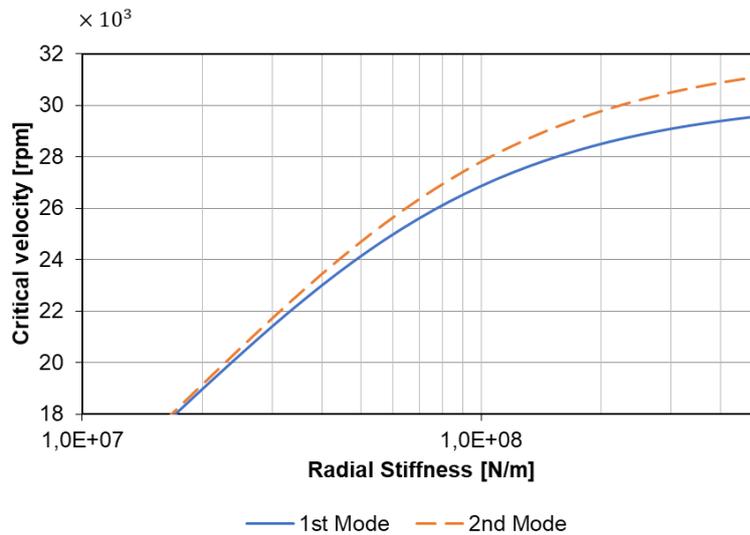


Figure 9 - Critical velocity as a function of radial stiffness for the analyzed case: 1st and 2nd resonance mode.

It is observed, given the slope of the curves, that small variations in the radial stiffness values significantly affect the critical velocity, mainly for the region between 1.0×10^7 N/m and 1.0×10^8 N/m. For example, if a radial stiffness of 5.0×10^7 N/m is considered, instead of the calculated one, the critical velocity is estimated around 24×10^3 rpm, an increase of 11% in relation to the previously calculated value. This indicates that the rigid radial value must be set carefully for a project like this, as the parameter has a relevant influence on the critical velocity. On the other hand, this sensitivity also offers an opportunity to be explored: by measuring the critical velocity experimentally, it is possible to infer bearing stiffness with some precision. The vibration as a function of rotation frequency is measured through the run-up test and the results are shown in Figure 10. The rotor was not accelerated beyond the indicated frequencies due to risks to operator safety.

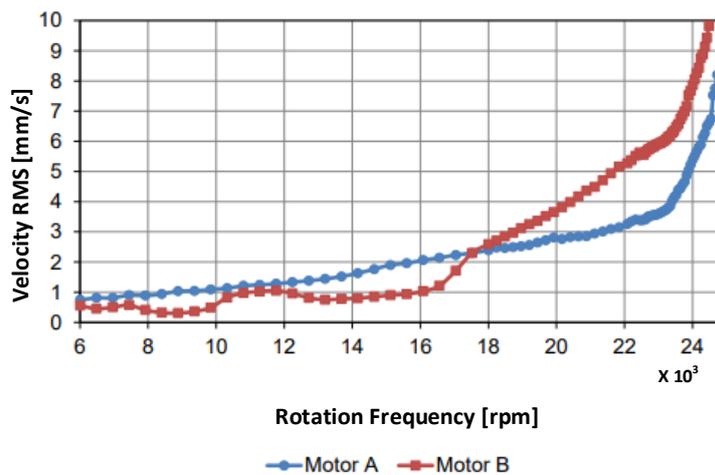


Figure 10 - Results for the run-up test.

Between 6×10^3 rpm and 18×10^3 rpm, Motor A presents a linear behavior, as expected. It is assumed that the oscillations exhibited by Motor B are due to an electromagnetic resonance, which is excited between 10×10^3 rpm and 13×10^3 rpm. Its causes may be related to shape errors, which lead to eccentricity of the rotor in relation to the stator.

Between 18×10^3 rpm and 23×10^3 rpm, both motors show a linear behavior. The difference in the slope of the curves is probably due to the degree of imbalance of each rotor. This indicates that Motor B has a greater residual mass than Motor A, being more sensitive to the increase in rotation frequency.

The measured vibration shows a very high increase after 23×10^3 rpm, when approaching 25×10^3 rpm. Thus, it is estimated that this is its critical velocity. However, since it is not possible to set this value with accuracy, the critical velocity is considered to be between 25×10^3 rpm and 26×10^3 rpm. In this case, the measured critical velocity is higher than the one calculated in the project, which was estimated at 21.5×10^3 rpm. This represents an error greater than 16%, which is significant for special designs that must operate close to their resonant frequency.

Based on the experimental values, it is possible to analyze the radial stiffness as a function of the critical velocity. This time, only the 1st resonance mode is evaluated and the impact of the angular stiffness is considered from its minimum and maximum value, as shown in Figure 11.

Angular stiffness indicates the flexural strength of the shaft. Thus, for the same radial safety value, a higher critical velocity is obtained if the angular safety is significant. This parameter started to be considered due to the great divergence between the estimated value of radial stiffness (approximately 3.0×10^7 N/m) and that experimentally parameter if the angular resistance were neglected (at least 6.0×10^7 N/m).

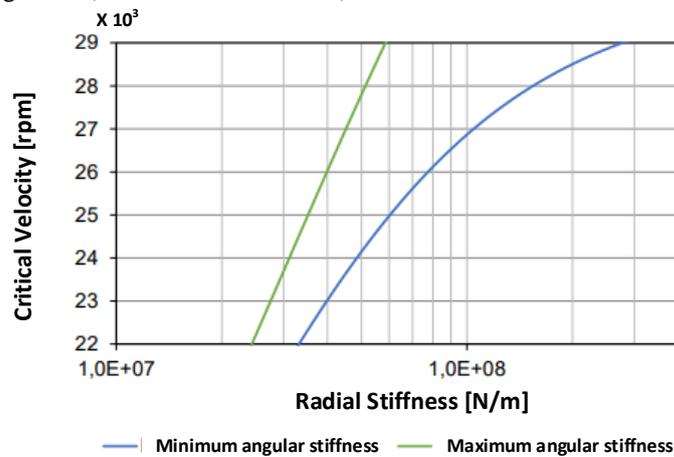


Figure 11 – Critical velocity as a function of radial stiffness for the analyzed case: minimum and maximum angular stiffness.

With the results presented in Figure 11 and the experimental tests, the values indicate that the radial stiffness of the bearings, given the considerations made, is equivalent to a value between 3.5×10^7 N/m and 7.6×10^7 N/m. This means that the calculated stiffness values are lower than the stiffness value measured experimentally, that is, the proposed equation seems to have underestimated the bearing stiffness value for this case. As indicated in Figure 11, a single critical velocity value can be generated by different combinations of radial and angular stiffness, which makes it difficult to determine these values. To overcome this situation, it is proposed the use of a modified equation, which is shown in Figure 12. The graph shows the equation developed by Gargiulo, which was adopted in this paper, for bearing 6003, in addition to the radial stiffness values provided by manufacturers A, B and C. The radial stiffness given by the modified equation is within the range suggested by the experiments. Therefore, it can be considered that the radial stiffness of the bearings used is around 4.67×10^7 N/m.

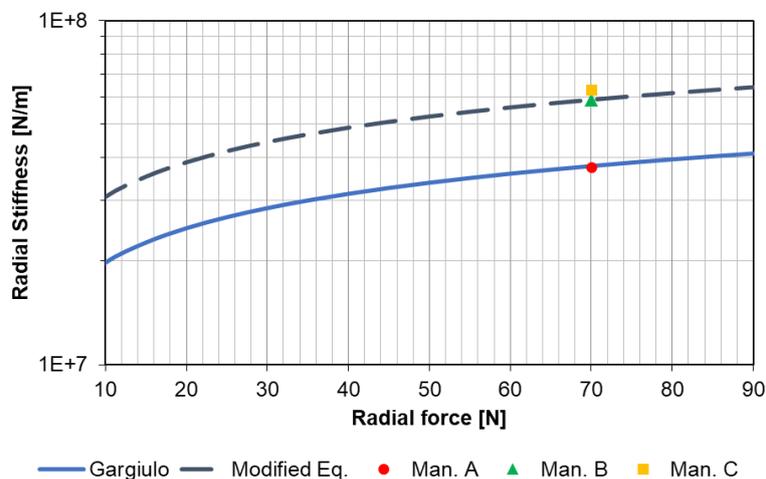


Figure 12 - Radial stiffness as a function of radial force: modified equation.

5. CONCLUSIONS

This paper evaluated the stiffness of the bearings of a rotor in order to improve the calculation of its critical velocity. For this, an equation was presented to analyze the dynamic behavior of the system. The proposed equation was compared with the methods used by the world's main bearing manufacturers. There is no consensus among the different methods for the value of radial stiffness of the listed cases. In the case of deep groove ball bearings, the values provided by the proposed equation are aligned with the median of the too much, although lower than those suggested by manufacturers B and C.

Experimental tests were carried out to validate the critical velocity of a rotor and, consequently, the safety value of its bearings. The experimentally measured critical velocity proved to be a superior development through the developed mathematical model. So, the measured stiffness experimentally is higher than that estimated by the proposed equation. This indicates that the method defined to calculate the radial stiffness of the bearings underestimates its value for the analyzed conditions.

The proposed equation seems to be conservative when estimating the radial stiffness of the bearings from deep groove ball bearings, at least for the analyzed case. By indicating a stiffness value lower than that observed in practice, the method also underestimates the critical velocity of the rotor. In general terms, this is positive, as it culminates in more robust and safer designs, as the rotor must operate with a margin of separation greater than that foreseen in the analysis stage. However, this can limit the development of special, bolder projects that need to operate close to their critical velocities, for example. Another consequence of this would be the use of more material (i.e. shafts with larger diameters) than necessary, for example.

6. REFERENCES

- Electric Motor Market Size, 2025. Disponível em: . Acesso em: 20 jun. 2021.
- Gargiulo, E. P. A Simple Way to Estimate Bearing Stiffness. *Machine Design*, v. 52, n. 17, p. 107–110, 1980.
- Han, S. M.; Benaroya, H.; Wei, T. Dynamics of Transversely Vibrating Beams Using Four Engineering Theories. *Journal of Sound and Vibration*, V. 225, N. 5, P. 935–988, 2 Set. 1999.
- International Energy Agency. *World Energy Outlook*. p. 810, 2019.
- International Organization for Standardization. *ISO 20816: Mechanical vibration - Measurement and evaluation of machine vibration - Part 1: General guidelines*. [s.l: s.n.]. 2016.
- Krämer, E. *Dynamics of rotors and foundations*. Berlin; New York: Springer Verlag, 1993.
- Lalanne, M.; Ferraris, G. *Rotordynamics prediction in engineering*. 2nd ed ed. Chichester ; New York: John Wiley, 1998.
- Möller, R. Evaluation of the Rigidity of the Rotor Bearings to Improve the Calculation of its Critical Velocity. Completion of the Bachelor's Degree In Mechanical Engineering, Federal University ff Santa Catarina. 2021 (Avaliação da Rigidez dos Mancais de Rolamento de um Rotor a fim de Aprimorar o Cálculo de sua Velocidade Crítica, Trabalho de Conclusão do Curso de Graduação em Engenharia Mecânica, Universidade Federal de Santa Catarina) 2021.
- Top 20 Electric Motor Manufacturers in 2019. *Technavio*, 14 jan. 2019. Disponível em: Top 20 Electric Motor Manufacturers in 2019 | Electric Motor Manufacturers List - Market Research (technavio.org). Acesso em: 20 jun. 2021

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