



COB-2021-0820

DYNAMIC ANALYSIS OF ROTATING SYSTEMS WITH VIBRATION ABSORBERS

João Henrique dos Santos de Pontes

Tiago Henrique Machado

Gregory Bregion Daniel

School of Mechanical Engineering, University of Campinas. Mendeleyev Street, 200, Campinas - SP - 13083-860

joaohspontes@gmail.com; tiagohtm@fem.unicamp.br; gbdaniel@fem.unicamp.br

Abstract. *In this work several concepts of vibration absorbers were studied and modelled by Finite Element Method from the commercial software Ansys. Then, modal analysis was performed in order to obtain the natural frequencies of these concepts. The geometry of promising concepts was modified to match the absorber natural frequencies with the critical speeds of the rotor. Absorbers were then assembled in the rotor to simulate amplitude levels by unbalance response in the frequency domain, comparing the cases with and without absorber on the rotating shaft. These results were also compared to a case in which the absorber was replaced by its concentrated mass and inertia to ensure that the effect on the original system is due to the absorber dynamic behavior and not caused by the change in the system overall parameters. Results show a promising capacity of amplitude reduction, but with narrow band-gap for some cases. Both the change in position and increase of overall size and mass of the absorber are capable of broadening the band gap but must be analyzed carefully in order to enable its use in real machine due to spatial constraints and change of system characteristics.*

Keywords: *Rotordynamics, Vibration absorber, Dynamic effects*

1. INTRODUCTION

Rotating machines are widely used in innumerable areas of industry and are often seen in their most varied sizes and applications, from huge turbomachines used in power generation plants to small and domestic machine tools. Due to its geometric and manufacturing characteristics, rotors have some inevitable mass unbalance, which causes both lateral and torsional vibration on the rotating system and its effect can be increased by conditions of resonance or lack of damping. Regardless of its cause, such vibrations are undesirable and can cause some serious issues both to the equipment itself and its operator.

There are several types of dynamic absorbers used in rotating machines to reduce vibration. Campos and Nicoletti (2014) applied a viscoelastic damper to a vertical washing machine. Bavastrì *et al.* (2008) proposed a layer of viscoelastic material between bearing housing and system foundation. Liebich *et al.* (2012) evaluated a similar type of vibration absorber using viscoelastic material between the bearing and its housing. Although viscoelastic material presents great results due to its high capacity of energy dissipation (Bavastrì *et al.*, 2008) it shows some serious issues during modelling and simulation due to its nonlinear behavior.

Other type of vibration absorbers is the squeeze film damper, as studied by Shen *et al.* (2005) and Hamzehlouia and Behdinan (2020). However, this type is mainly used to high-speed machinery due to its high cost.

In this context, some authors focus on cheaper dynamic absorbers which are directly assembled to the rotor. This type of absorber consists in the same principle initially proposed by Watts (1883). The main idea is to attach a piece of mass to the primary system using a spring-damper model, being later known as *tuned mass damper - TMD* when Ormondroyd (1928) firstly modelled the mass-spring-damper vibration absorber. Many authors have been studying the application of this type of absorber applied to rotating machine lateral vibration problem, as proposed by Ishida and Inoue (2007) and Prado and Ritto (2020).

Therefore, the aim of this work is to analyze the effect of dynamic vibration absorber when directly coupled to a rotating system. Regarding the traditional absorber that are coupled on the base/foundation, the absorbers evaluated in this paper have the advantage of attenuating the vibration directly on the rotor, before it is transmitted to other components such as bearings and foundation. Thus, it is possible to guarantee better dynamic conditions on these elements that compose the machine what tends to mitigate the incidence of faults. However, these absorbers should have simple geometries, allowing easy manufacturing, good mass balancing and practical assembly on the rotating shaft. Three geometries of absorbers are proposed and modelled by Finite Element Method using the commercial software Ansys. For this, the natural frequency of the vibration absorber should be matched to the critical speeds of the rotor. Once the coincidence of natural frequency and critical speed is achieved, the vibration absorbers are then assembled to the rotor at different

configurations in order to investigate its effect on vibration amplitude in the frequency domain.

2. METHODOLOGY

This section presents the methodology used to model and simulate the rotating system with and without the vibration absorbers proposed, highlighting the different configurations evaluated in this work.

2.1 Unbalance response of rotor

A rotating system is basically composed of shafts, disks and bearings or supports. Its behavior can be modelled by Finite Element Method, in which the geometry model is discretized into a finite number of elements. Each element has its own matrix of mass, stiffness, damping, and for the particular case of rotating machine, also the gyroscopic matrix. Once the matrix of each element is obtained they are superposed in order to form the global matrices of the system.

The equation of motion of the entire system is

$$[M_g]\{\ddot{q}\} + ([C_g] + \Omega[G_g])\{\dot{q}\} + [K_g]\{q\} = \{f\}, \quad (1)$$

where $[M_g]$, $[C_g]$, $[G_g]$, $[K_g]$ are the global mass, damping, gyroscopic and stiffness matrices, respectively, and $\{\ddot{q}\}$, $\{\dot{q}\}$, and $\{q\}$ are the acceleration, velocity and displacement of each degree of freedom of the system. Finally $\{f\}$ represents the external forces vector, which in this case is equal to $F e^{i\Omega t}$ (applied in the central node at the disk).

Expressing the degree of freedom vector as a complex coordinate $q = Q e^{i\Omega t}$, velocity and acceleration are written as $\dot{q} = i\Omega Q e^{i\Omega t}$ and $\ddot{q} = -\Omega^2 Q e^{i\Omega t}$, respectively. Substituting it in Eq. (1):

$$Q = \frac{F}{-\Omega^2[M_g] + ([C_g] + \Omega[G_g])i\Omega + [K_g]} \quad (2)$$

in which can be obtained both the amplitude and phase in the frequency domain. Equation (1) is known as unbalance response of rotor and its results are typically shown in the form of a bode diagram.

2.2 Dynamic vibration absorbers

The idea of dynamic vibration absorbers is to attach a secondary mass to the primary system so that, at desired frequency, most of vibration energy will be concentrated at the secondary mass, and not on the main system, as shown in Fig. 1. Thus, this type of absorber named tuned mass damper must be tuned in order to make it work at a desired operating frequency. However, tuning this type of geometry can be not feasible in many situations. For example, the use of a secondary mass too heavy can change system characteristics instead of absorbing vibration, facing spatial constraints.

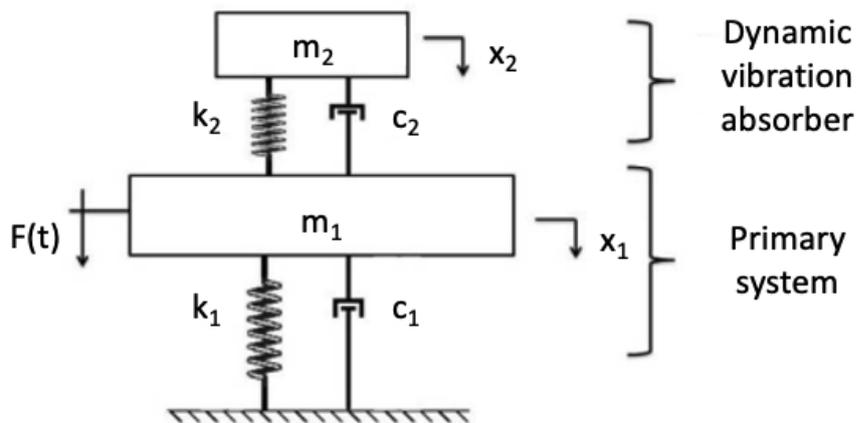


Figure 1: Typical dynamic vibration absorber configuration. From Piccirillo *et al.* (2019)

When working with rotating machine there is a second concern related to the symmetry of absorber, since the primary system will be rotating and thus performing a circular or elliptical orbit. If the absorbers are designed in an asymmetrical way, it may cause even more vibration as a result of its unbalance. Thereby, the absorbers concepts evaluated in this work are based on two ideas, in order to guarantee the absorber symmetry. First idea is to arrange several pendulums around the shaft as secondary masses, while the second idea is to place an external ring as secondary mass. In both cases the main challenge is to perform the parameter tuning without changing the original system response.

The natural frequencies are obtained via modal analysis, which basically consists of an eigenvalue-eigenvector problem. Rewriting Eq. 1 in the state-space form, the system dynamic matrix is:

$$[A] = \begin{bmatrix} 0 & [I] \\ -[M_g]^{-1}[K_g] & -[M_g]^{-1}([C_g] + \Omega[G_g]) \end{bmatrix} \quad (3)$$

Modal parameters such as natural frequency, damping factor and vibration modes are calculated from the eigenvalues and eigenvectors associated with the system dynamic matrix $[A]$. For each eigenvalue λ , natural frequency and damping factor are calculated respectively as:

$$\omega_n = |\lambda| \quad (4)$$

$$\xi = \frac{-Re(\lambda)}{\omega_n} \quad (5)$$

3. RESULTS AND DISCUSSION

3.1 Original rotating system

The rotor used in this work is based on Fontes and Nicoletti (2015) and consists of a thin shaft with two disks equally spaced between the bearings as shown in Fig. 2. In this figure, the right side disk is referenced as *disk 1*, and the left side disk as *disk 2*. In one of the extremities there is a free-end part of the shaft, which is used to assemble the absorbers. Figure 3 shows the main dimensions of studied rotating system. The rotor (shaft and disks) is entirely made of structural steel, whose properties are shown at table 1 and modelled using *SOLID187* elements. This element is defined by a 10-nodes tetrahedral shape, with three degrees of freedom at each node, being x , y and z displacement.

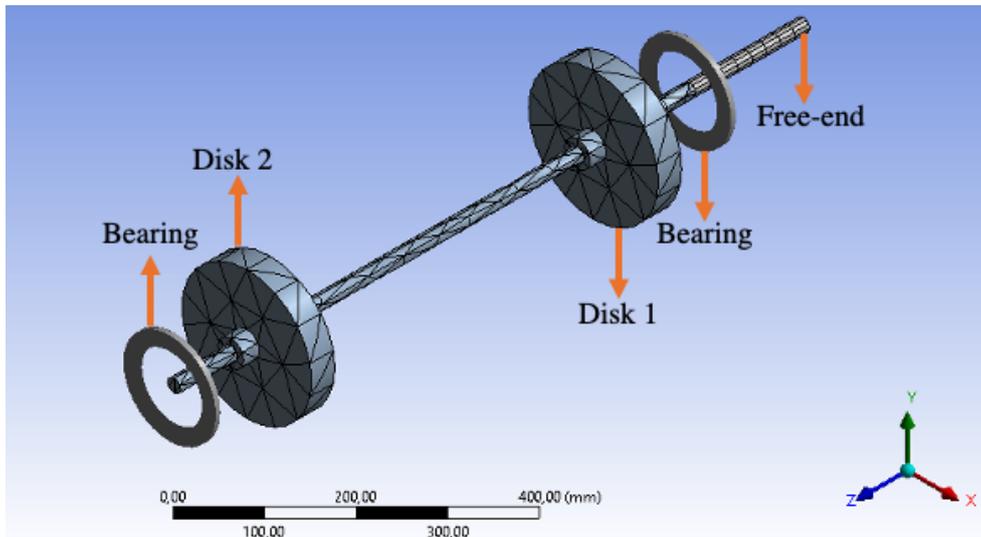


Figure 2: Rotating system studied

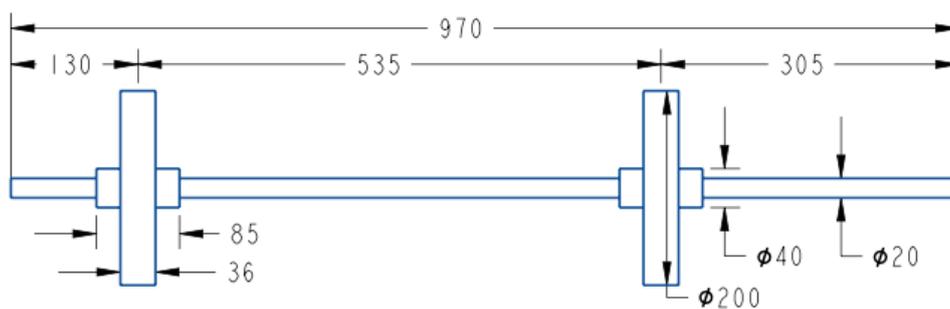


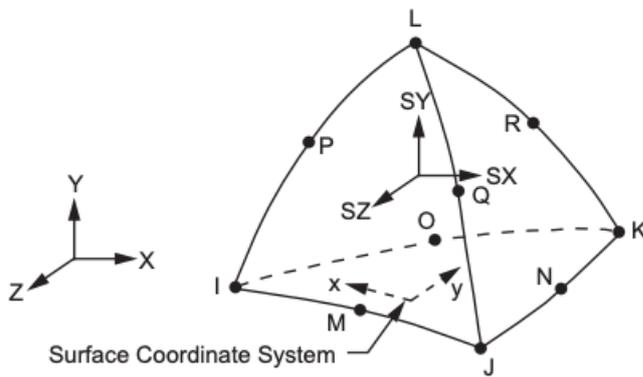
Figure 3: Rotating system main dimensions.

Material	E (N/m)	ρ (kg/m ³)	ν (-)
Structural Steel	200e9	7850	0.30
AISI 1070 Spring Steel	205e9	7850	0.29
Duraluminium	70.3e9	2830	0.33

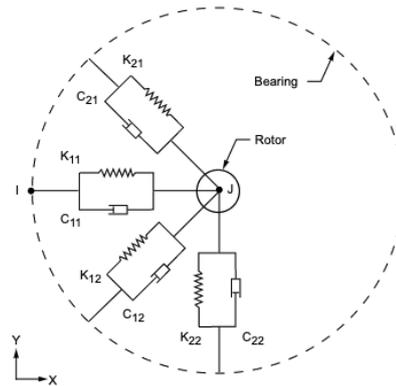
Table 1: Material Properties used in the vibration absorbers

Stiffness (N/m)	Damping (N.s/m)	
K_{11}	1595.0	C_{11} 0.1
K_{22}	2145.0	C_{22} 0.1
K_{12}	0.0	C_{12} 0.0
K_{21}	0.0	C_{21} 0.0

Table 2: Equivalent coefficients of stiffness and damping considered in the bearings



(a) SOLID187 element schematic



(b) Bearing element schematic

Figure 4: Elements used for meshing rotating system. From Ansys Mechanical APDL Element Reference

The behavior of rotating system depends not only from the shaft and disk but also from the bearings. According to Fig 2, gray rings represent the bearings position. In this work, bearings have been included using *COMB1214* element (Fig. 4b) which considers equivalent coefficients of stiffness and damping. The bearings used have only direct terms for both stiffness and damping, whose values are shown in table 2.

The external force is applied on disk 1 and it is equivalent to a rotating mass unbalance of $1e-3$ kg at 100 mm from the geometric center of the disk. The analysis is made to obtain the frequency response for the harmonic unbalance load, in which excitation frequency is equal to the rotational speed of the shaft.

From the graphs of Fig. 5, Fig. 6 and Fig. 7 it is possible to see that the greatest vibration amplitude occurs at $28Hz$ which matches the second critical speed of the system. From the Campbell diagram the second critical speed represents the first vibration mode with forward whirl. Since unbalance load excites only forward whirl, this is the frequency which the absorbers will be tuned to achieve.

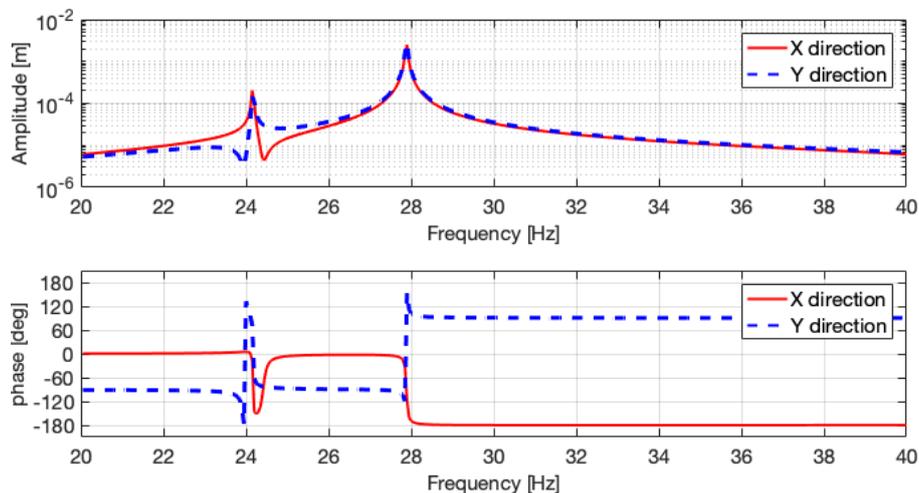
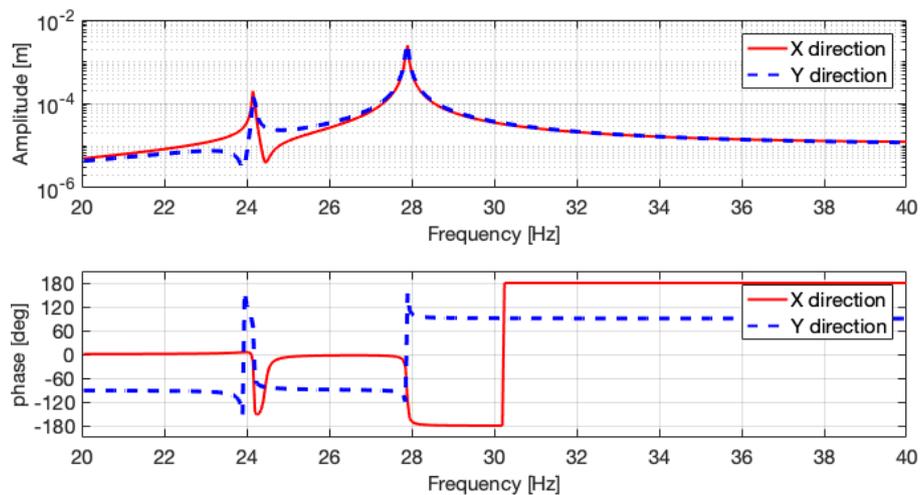


Figure 5: Bode diagram of original unbalance system shown at disk 1



(a) (b) Disk 2

Figure 6: Bode diagram of original unbalance system shown at disk 2

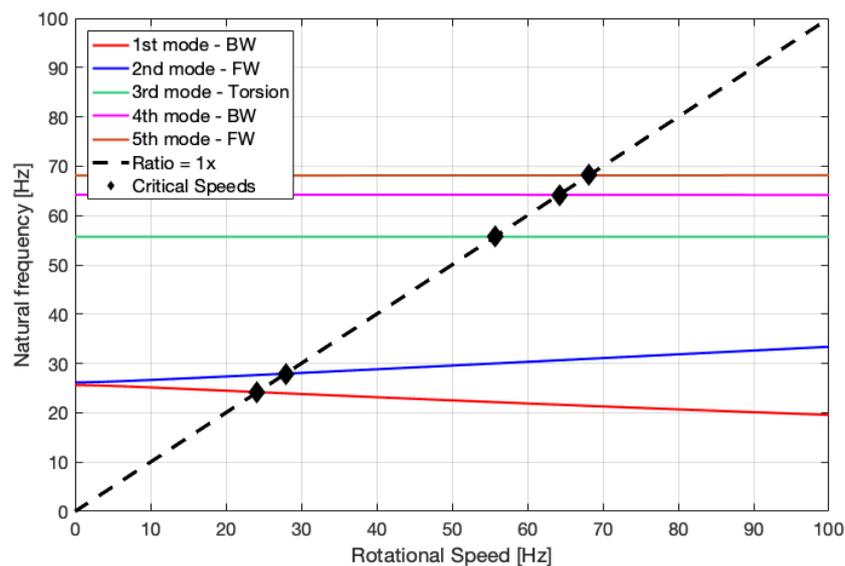


Figure 7: Campbell diagram of rotating system

3.2 Vibration absorbers

Several geometries have been studied and tested via modal analysis using Ansys. The numerical analysis were performed with the inner diameter of the absorber bore with a fixed displacement boundary condition, since it would be attached to the rotor. The three concepts, whose natural frequency is closer to the rotating system were then selected in order to adjust its geometrical parameters so the absorber natural frequency would match the second critical speed of the original system (forward mode). These concepts are presented in Figs 9 and 8.

For all the three concepts, the inner ring was made of duraluminum, the blades which act as spring were made of AISI 1070 spring steel, and the secondary mass (pendulum and/or external ring) was made of structural steel. Properties of all materials are shown in table 1.

With all absorbers concepts tuned for a natural frequency of approximately 28Hz, its resonant mode should occur when the rotating system approaches its second critical speed, thus concentrating its vibration energy on the absorber secondary masses. All three absorber concepts were tested assembled in different locations on the original system. Also, some configurations include two identical absorbers in order to evaluate if the increase of mass destined to absorb vibration energy reflects in greater amplitude reduction.

After adjusting the absorbers to match the natural frequency of the rotor they were assembled on the free-end of the

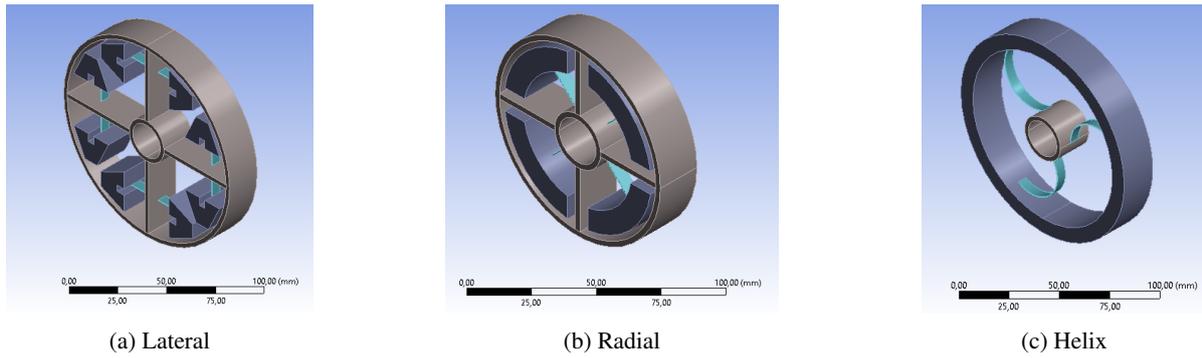


Figure 8: Isometric Views

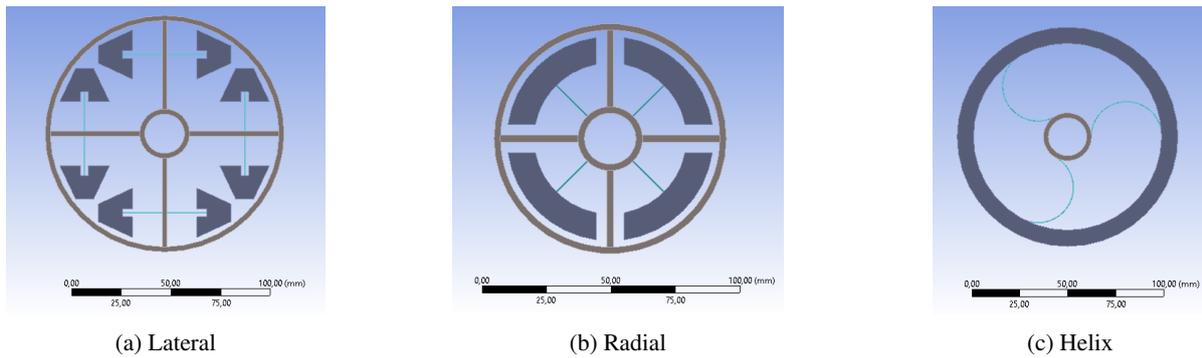


Figure 9: Front Views

rotor as a solid body and as a concentrated point with equivalent mass and inertia representing the absorber. Also it was tested the configurations with one and two absorbers located at the free-end. Therefore, four different arrangements were tested, being:

- 1 concentrated: single concentrated point with equivalent mass and inertia representing the absorber, at free-end;
- 2 concentrated: dual concentrated point with equivalent mass and inertia representing the absorber, at free-end;
- 1 solid: single absorber, at free-end;
- 2 solid: dual absorber, at free-end.

Unbalance response analysis were performed for these four different arrangements and the results of the amplitude in both x and y directions are shown in Fig. 10, Fig. 11 and Fig. 12 for the same disk where there is unbalance force applied (disk 1).

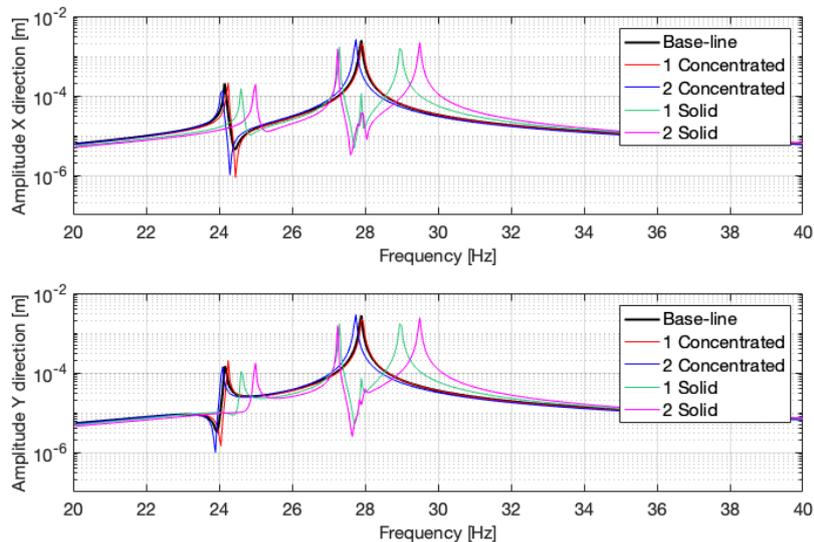


Figure 10: Unbalance response of disk 1 for radial pendulum absorber

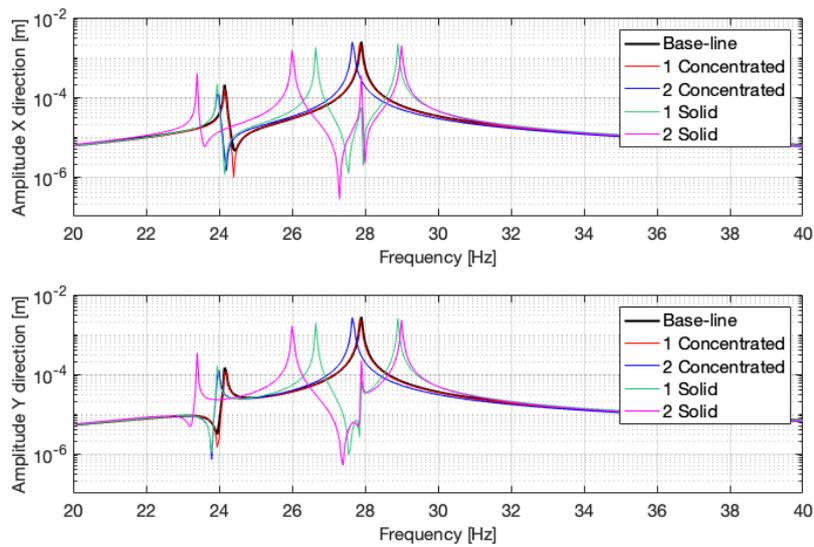


Figure 11: Unbalance response of disk 1 for lateral pendulum absorber

The objective of plotting the results for both real solid dynamic absorbers and also its equivalent mass and inertia concentrated at a single point is to evaluate if the change on the dynamic behavior of the rotating system is due to its variation in total mass and inertia or if it is caused by the vibration absorber effect. For all three absorbers it is possible to see that the concentrated point with equivalent mass and inertia barely affects the response of the rotating system, showing that the change on the dynamic behavior is really caused by the absorber effect.

Another point to be noted is that the arrangement with two concentrated point presents more visible changes on the amplitude peak, although they are still very low in comparison to general behavior of the system. It can be also noted with two real absorbers. This statement can be explained by the fact that with more absorbers there is more mass available to concentrate vibration energy in itself, thus having more impact on the system, as expected.

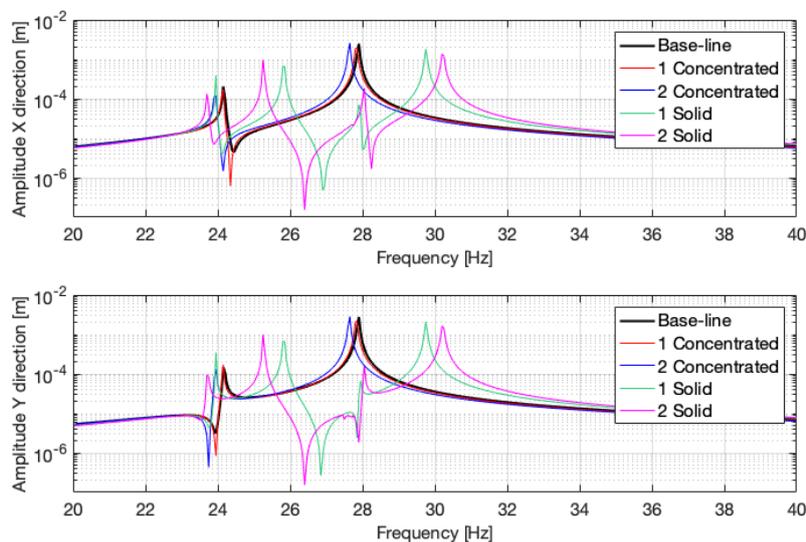


Figure 12: Unbalance response of disk 1 for helix absorber

In order to compare the absorber concepts between itself, the results of real solid absorbers have been overlapped for case with single absorber at free-end as showed in Fig. 13 and case with dual absorber at free-end in Fig. 14.

In general it is possible to verify that all three geometries proposed were capable of reduce vibration amplitude on the desired frequency. However, the band-gap is quite narrow for radial pendulum absorber, specially when compared with the helix absorber, since it represents the best result overall. For presenting the most promising results, the helix absorber was then chosen to further analysis, in which the absorber is evaluated in different assembly position, being:

1. 1-FE: Single absorber at free-end;
2. 2-FE: Dual absorber at free-end;

3. 1-FE 1-C: Single absorber at free-end and single absorber centralized between the disks;

4. 1-C: Single absorber centralized between the disks.

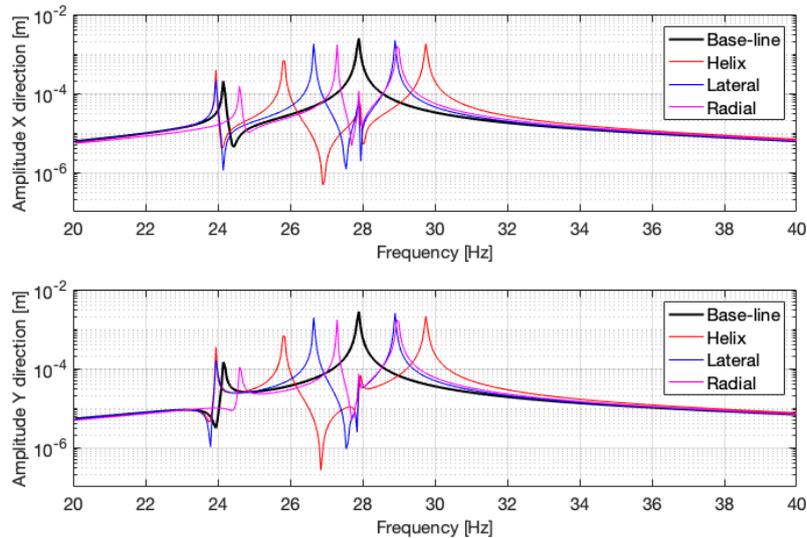


Figure 13: omparison between different concepts for the case with single absorber at free-end

Based on the rotating system response with helix absorber at different locations, Fig. 15 and Fig. 16, it is possible to state that the use of two absorbers creates a wider band gap than a single absorber, previously observed. Thus, the results of the cases 1-FE 1-C and 2-FE present larger band-gap than the cases 1-C and 1-FE respectively. Moreover, it is possible to verify that the absorber positioning between the rotor disks is significantly better than the assembly at the free-end. However, the assembly of the absorber in the central region of the rotor often presents limitations due to the spatial constraints of the machinery in real applications.

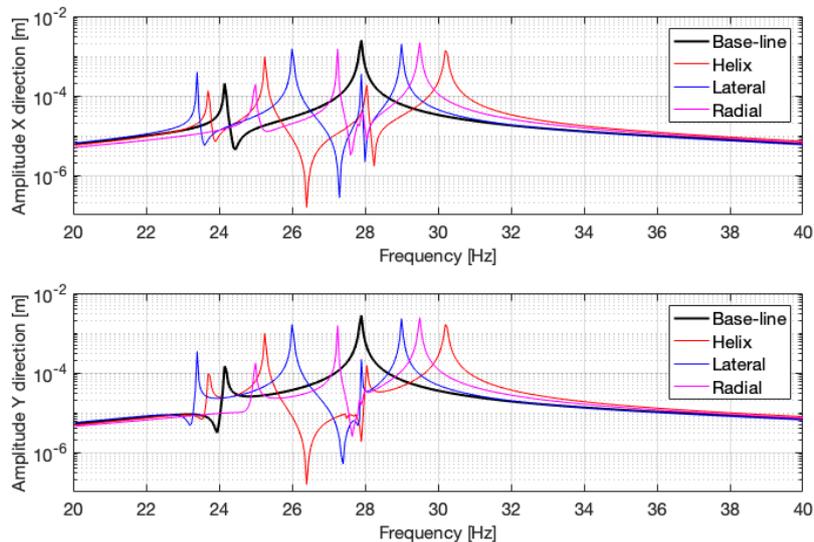


Figure 14: Comparison between different concepts for the case with dual absorber at free-end

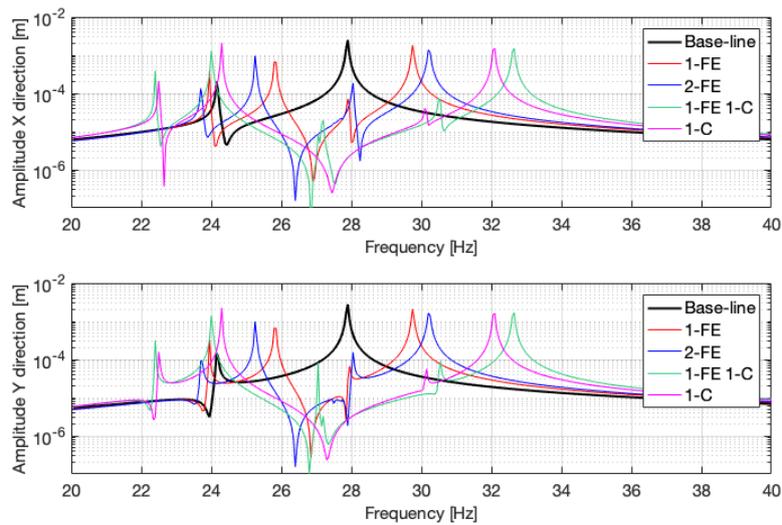


Figure 15: Comparison between different locations for the helix absorber assembly shown at disk 1

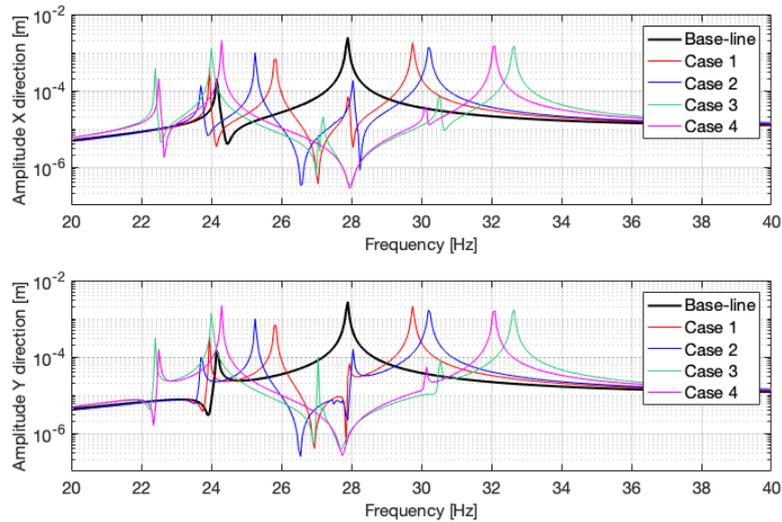


Figure 16: Comparison between different locations for the helix absorber assembly shown at disk 2

For summarizing results, numeric values for band-gap is defined as the length of frequency range between the two amplitude peaks created by the absorbers, and amplitude reductions is measured between the original system peak and the greatest amplitude peak inside the band-gap. All results are shown at table 3 and 4.

Geometry	Band Gap (x)	Amplitude Reduction (x)	Band Gap (y)	Amplitude Reduction (y)
Lateral Pendulum	2.3Hz	2.44mm	2.3Hz	2.72mm
Radial Pendulum	1.7Hz	2.38mm	1.7Hz	2.71mm
Helix	3.9Hz	2.43mm	3.9Hz	2.72mm

Table 3: Results summary for case with single absorber at free-end

Geometry	Band Gap (x)	Amplitude Reduction (x)	Band Gap (y)	Amplitude Reduction (y)
Lateral Pendulum	3.0Hz	2.14mm	3.0Hz	2.57mm
Radial Pendulum	2.3Hz	2.46mm	2.2Hz	2.75mm
Helix	5.0Hz	2.40mm	5.0Hz	2.63mm

Table 4: Results summary for case with dual absorber at free-end

4. CONCLUSION

In this paper three concepts of dynamic vibration absorbers were proposed in order to reduce vibration amplitude of a rotating system operating close to its second critical speed, at approximately 28Hz, under unbalance conditions. For this, three absorbers with simple geometries, easy manufacturing and assembly on the shaft are designed and tuned to have a resonance frequency as close to 28Hz as possible, and then they are assembled directly on the rotating system.

The computational simulations performed in this work show that all three absorbers were capable of reducing amplitude levels on the desired frequency, creating a band-gap around the second critical speed of the rotating system. For the radial pendulum absorber, the band-gap is quite narrow, being lower than 2Hz (5.9% of the second critical speed) for the case with 1 absorber at free-end. On the other hand, the helix absorber has the best result overall, being the band-gap about 4Hz (14.1% of the second critical speed) for the case with 1 absorber at free-end.

Although band-gap created by pendulum absorbers were not as large as desired, it still provides a significant reduction of the amplitude levels. The problem associated with narrow band-gap is related to the precision of the machine operational speed. If band-gap is not large enough, a small change of the operation speed may cause high amplitude levels, thus resulting danger conditions. It is also possible to conclude that the increase mass on the absorbers is capable of widening the band-gap created, however it must be considered carefully due to spacial constraints and also changes in the dynamic characteristics of the rotating system.

When varying the assembly position it is clear that positioning the absorber centralized between the disks is the best theoretical option. However it might not be possible to arrange due to spacial constraints of the machinery in real applications.

Finally, the tuning of the absorber was made considering a modal analysis of only the absorber itself, being its inner bore fixed as boundary condition. This process showed to be efficient for a first evaluation, but the band-gap created may not be centralized around the critical speed used as reference for tuning. Thus, it is recommended to perform fine adjustments in order to guarantee both efficiency and robustness. Based on the promising results already achieved, a future optimization can be performed to improve the absorber effect on the rotating system, obtaining the geometric properties and dimensions that ensure greater vibration reduction and wider band-gap.

5. ACKNOWLEDGEMENTS

The authors thank the National Council for Scientific and Technological Development (CNPq) and the University of Campinas (UNICAMP) for infrastructural and financial support.

6. REFERENCES

- Bavastri, C.A., da S. Ferreira, E.M., de Espíndola, J.J. and de O. Lopes, E.M., 2008. "Modeling of dynamic rotors with flexible bearings due to the use of viscoelastic materials". *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, Vol. 30, No. 1, pp. 22–29.
- Campos, R.O. and Nicoletti, R., 2014. "Vibration reduction in vertical washing machine using a rotating dynamic absorber". *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, Vol. 37, No. 1, pp. 339–348.
- Fontes, Y.C. and Nicoletti, R., 2015. "Rotating dynamic absorber with viscoelastic element". *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, Vol. 38, No. 2, pp. 377–383.
- Hamzehlouia, S. and Behdinan, K., 2020. "Squeeze film dampers supporting high-speed rotors: Rotordynamics". *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, Vol. 235, No. 3, pp. 495–508.
- Ishida, Y. and Inoue, T., 2007. "Vibration suppression of nonlinear rotor systems using a dynamic damper". *Journal of Vibration and Control*, Vol. 13, No. 8, pp. 1127–1143.
- Liebich, R., Scholz, A. and Wieschalla, M., 2012. "Rotors supported by elastomer-ring-dampers – experimental and numerical investigations".
- Ormondroyd, J., 1928. "The theory of the dynamic vibration absorber". *Trans., ASME, Applied Mechanics*, Vol. 50.
- Piccirillo, V., Tusset, A.M. and Balthazar, J.M., 2019. "OPTIMIZATION OF DYNAMIC VIBRATION ABSORBERS BASED ON EQUAL-PEAK THEORY". *Latin American Journal of Solids and Structures*, Vol. 16, No. 4.
- Prado, L. and Ritto, T., 2020. "Vibration reduction of a rotating machine using resonator rings". *Mechanics Research Communications*, Vol. 107, p. 103533.
- Shen, G., Xiao, Z., Zhang, W. and Zheng, T., 2005. "Nonlinear behavior analysis of a rotor supported on fluid-film bearings". *Journal of Vibration and Acoustics*, Vol. 128, No. 1, pp. 35–40.
- Watts, P., 1883. "On a method of reducing the rolling of ships at sea".

7. RESPONSIBILITY NOTICE

The authors are solely responsible for the printed material included in this paper.