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DEVELOPMENT OF A DIDACTIC BENCH FOR SHAFT MISALIGNMENT AND ROTOR UNBALANCE EVALUATION OF VIBRATION LEVELS

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Abstract. *The present study aims to evaluate the features present in a didactic equipment built for the study material enhancement of an academic laboratory. The main purpose of this equipment is to simulate vibration defects and its correction through the use of accelerometers and its signal analysis interpretation. This would adequately demonstrate the effects of vibration defects in a rotating machine, exploring its structure stiffness and level of defects. An enhancement is expected in the quality of practical studies of the UTFPR Mechanical Engineering community. The assembling process is described concerning the material used which directly impacts in the structural stiffness, thus, level of observed vibration. The evaluation methodology is based on the signal analysis of the observed vibrations. In order to reach the correct readings, the equipment had to undergo alignment and balancing procedures. The results show that the identification of both defects, misalignment and unbalance were possible while comparing the literature with the resulting signal analysis charts. It was also possible to analyze how does the structural stiffness and damping influence the vibrations amplitudes. The validity of the proposed didactic equipment was verified through the measured vibration spectrum, which results shown to be in accordance with literatures.*

Keywords: *Vibration, Misalignment, Unbalance, Didactic Bench, Stiffness, Signal Analysis, Cross-Spectrum.*

1. INTRODUCTION

A didactic equipment was built for the UTFPR vibration laboratory. The purpose of this equipment was to allow the academic community to explore the effects caused by several vibration defects in rotating machines. These defects are: unbalance, parallel and offset misalignment.

The problem presented in this study is directly related to the level of vibration produced in laboratory experiments. Due to an elevated stiffness in the base of the equipment, the vibration propagation of defects were relatively low. This would impact in incorrect readings in the resulting signal analysis from the accelerometers readings.

The stiffness of a system is directly linked to its geometry and mechanical properties. According to INMAN, 2007 a material can perform as a spring depending on its configurations and movements longitudinal, transversal and torsional. The structure which composes the base of the equipment can be modeled as a slender beam. Thus, the height of this system impacts directly in the observed stiffness. Because of this, assemblage of the system were developed intending to reduce its base stiffness via height increase.

Signal analysis is the main method for identification of the vibration defects discussed in this paper. The rotation frequency of the induction motor is the base for this analysis. In this case 30 Hz was the rotating frequency. The present literature states that for an accurate defect identification it is necessary to observe the frequency readings at two and three times the rotation frequency NEPOMUCENO, 1989.

Firstly, the equipment went through a process of balancing and alignment. This intended to ensure the quality of the results. Accelerometers were then attached to the operating system while simulating both vibration defects, misalignment and unbalance. For unbalance and angular misalignment defects the rotation frequency dominates the spectrum with the highest vibration peak. For the parallel misalignment the dominating frequency is two times the rotation frequency, 60 Hz in this case SCHEFFER & GIRDHAR, 2004.

In order to allow significant vibration levels on the system, due to defects, two support configurations were developed and analysed. Each configuration had different stiffness and damping, the analysis was done by impact test. From this analysis, it was chosen the configuration that presented highest vibration levels in order to facilitate the diagnoses during the classes.

By measuring the vibration spectrum, this didactic bench allows the community to easily understand how to diagnostic the misalignment and unbalance defects in rotating machinery. Therefore, in the industry, the vibration analysis should identify developing problems before they become serious and cause a failure.

2. METHODOLOGY

2.1 Structural Assemble

To develop the didactic equipment proposed in this paper, two configurations were projected, built and analysed. The main purpose of this was to choose which configuration would present higher levels of vibration while experiencing induced defects of misalignment and unbalance. And most importantly, the easiness of test execution and defect correction. In both configurations AISI 1020 steel sheets were used to sustain the induction motor and shaft, the dimensions of this are presented in Fig. 1.

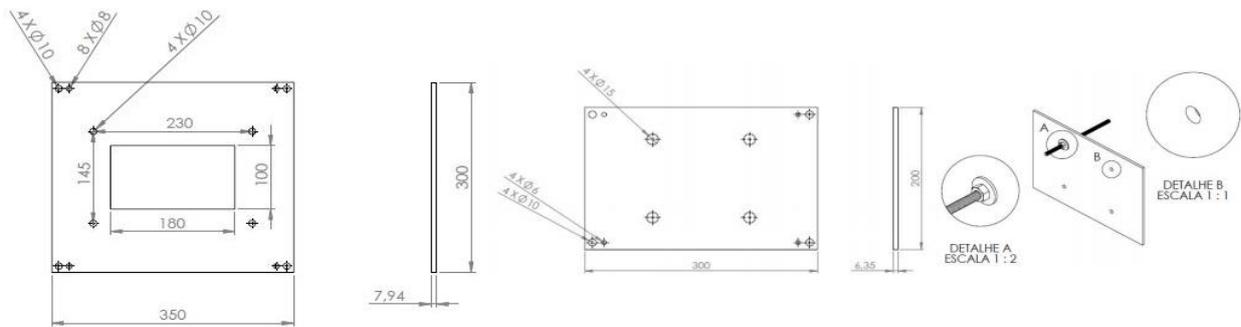


Figure 1 – Material used for the assembly

The main difference within the two configurations was its fixation base. For the first one, it was chosen a threaded bars system with a stainless steel M08 bolts, the main characteristic for this configuration was low levels of damping and higher levels of stiffness. Furthermore, this configuration allows height adjustment of the steel bases from the nuts as demonstrated in Fig. 2.



Figure 2 – First Configuration

In the second configuration the system base is composed by cushions (stud vibration isolators). Therefore, higher levels of damping and lower levels of stiffness as presented in Fig. 3.



Figure 3 - Second Configuration

The induction motor used (fixated in one base) was a WEG, 60Hz and 1700 rpm. The rotor (fixated in the other base) was composed by a 35-millimetre diameter AISI 1020 shaft with a 500-millimetre length. This shaft was sustained by two UCP 207 rolling bearings. Two AISI 1020 discs were fixated in the rotor aiming for the performance of unbalancing experiments. The two shafts (induction motor and shaft) were coupled by an elastic coupling.

2.2 Flexibility Evaluation

In order to analyse the vibration levels of the two configurations, in a wide frequency range, it was performed a flexibility analysis. Initially, the rotor was balanced by the two-plane method. This process was performed with an accelerometer and an oscilloscope. Secondly, the shaft was aligned by the laser aligner method. The equipment used was of Easy-Laser brand, model D450. This method presents a high degree of accuracy. It is presented in Tab. 1 the tolerances accepted for the rotation evaluated (1795 rpm).

Table 1 - Laser Alignment Tolerances (Easy-Laser Manual 05-0100)

Alignment Tolerances		
	Offset	Angular
Excellent	0.05 mm	0.04 mm
Acceptable	0.1 mm	0.07 mm

After the systems were balanced and aligned, vibration defects were introduced to the system for vibration level analysis. Firstly, a known mass of 16.5 grams were added to the rotor to create an unbalance.

According to SCHEFFER & GIRDHAR, p. 90, 2004, it is possible to identify static and couple unbalance by the spectrum analysis for the amplitude peak. According to the authors, the vibration cross-spectrum for a rotor machine with an unbalance defect must demonstrate a peak at the rotor frequency, or 1x rpm for radial vibration. For this reason, and also to perform a phase analysis, two accelerometers were attached to the shaft bearings in the radial direction as shown in Fig. 4.



Figure 4 - Accelerometers in the radial direction

With the motor running, the signal was measured and stored in a computer for further mathematical processing. So, it was measured the vibration cross-spectrum between the signals in the radial direction, as shown in the Fig. 4. After this, the unbalance mass was removed to evaluate the vibrations due to the misalignment defect.

To impose a misalignment defect it was used the laser aligner to change the horizontal position of the motor, keeping the vertical plane aligned. A total of 0,3 mm of horizontal misalignment was introduced to the system. According to HINES, JESSE AND EDMONFSON, 1970 it is possible to keep the alignment of a plane and still observe the effects of misalignment by changing the other plane. The calculated values to correct the angular and parallel misalignment were obtained by the following Equations (1) and (2)

$$Parallel\ Misalignment = \sqrt{(Parallel\ Misalignment_{Horizontal})^2 + (Parallel\ Misalignment_{Vertical})^2} \quad (1)$$

$$Angular\ Misalignment = \sqrt{(Angular\ Misalignment_{Horizontal})^2 + (Angular\ Misalignment_{Vertical})^2} \quad (2)$$

According to SCHEFFER & GIRDHAR, p.94 2004, to identify an angular misalignment it is necessary to analyze the vibration cross-spectrum in the axial vibration. The spectrum must indicate peaks in both 1x and 2x rpm, Furthermore, a 180° phase difference (in the axial direction) must be observed on the bearings of the two machines across the coupling.

To identify a parallel misalignment, it is necessary to analyze the vibration in the radial direction, searching for peaks at 2x rpm. Furthermore, a 180° phase difference (in the radial direction) must be observed on the bearings of the two machines across the coupling.

The positioning chosen to measure these vibrations, axial and radial, are shown in the Fig. 5.

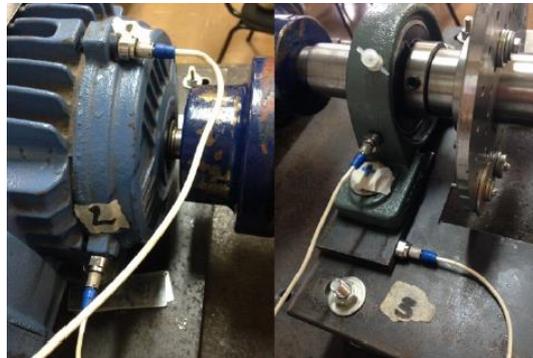


Figure 5 - Accelerometers for misalignment detection

To complement the flexibility analysis, to know the natural frequencies of the systems (the two configurations), it was performed an impact test in the didactic bench. In this test, it was measured the frequency response of two settings at specific points.

This test is important because it can foresee which system will have higher levels of vibration in a vibration range. Both systems were tested with an ENDEVCO impact hammer and two 256-100 ENDEVCO accelerometers, these were positioned in the axial and radial directions as shown in Fig. 6.



Figure 6 - Impact test, axial and radial directions

3. RESULTS AND DISCUSSION

The vibration levels were evaluated in both assembly configurations according to the methodology previously described. The time course of the experiments was measured. The resulting time course needed to finish each experiment was about 20 minutes. This result is important due to the time limit of an experimental class which is 90 minutes at UTFPR.

Before performing vibration measurements for simulated defect diagnostics, a rotor balancing and shaft alignment procedure were performed. The residual values obtained after alignment with the laser aligner are shown in Tab. 2. A comparison between these values with the laser aligner tolerances in Tab. 1 shows that the values for the system with a threaded bar are excellent and for the system with the cushions are acceptable.

Table 2 - Resulting readings for the alignment process

	Readings - Threaded Bars		Readings - Cushions	
	Offset	Angular	Offset	Angular
Vertical	-0,031 mm	0,031 mm	-0,004 mm	0,059 mm
Horizontal	0,019 mm	0,018 mm	-0,005 mm	0,059 mm

The balancing procedure results demonstrated in Tab. 3 indicate that the two-plane balancing procedure significantly reduced the final vibration levels observed. This procedure was conducted only once, this is because the shaft remained the same for the rest of the experiments. To conclude, the corrective masses placed along the rotor were used to the other experiments.

Table 3 shows the rotor bearing vibration amplitudes before and after the balancing procedure by the two-plane method. The values of the correction masses added to the rotor discs are also shown. The plane A being the left side disc and B the right side. It is noticed that in the plane A the vibration decreased more than twice and in B more than the three times. These masses were kept attached to the system at all experiments.

Table 3 - Resulting readings for the balancing process

	Plan A	Plan B
Original Vibration (m/s ²)	0,78∠279°	2,7∠120°
Vibration Ma = 7g	10,4∠50°	7,15∠65°
Vibration Mb = 7g	6,43∠45°	12,35∠50°
Correction Mass		
Plan A	5,2∠110°	
Plan B	8,81∠-86°	
Final Vibration Levels		
Plan A	0,34 m/s ²	
Plan B	0,75 m/s ²	

3.1 Misalignment Defect Identification

3.1.1 Angular Misalignment

For this type of defect, it was not possible to effectively identify the angular misalignment for the equipment assembled with threaded bars. However, for the cushion assembly, according to SCHEFFER AND GIRDHAR, P.94, the results indicate an angular misalignment defect with the increase of the vibration level at 1x rpm (30 Hz in this case) in one of the bearings, as demonstrated in Fig. 7 below.

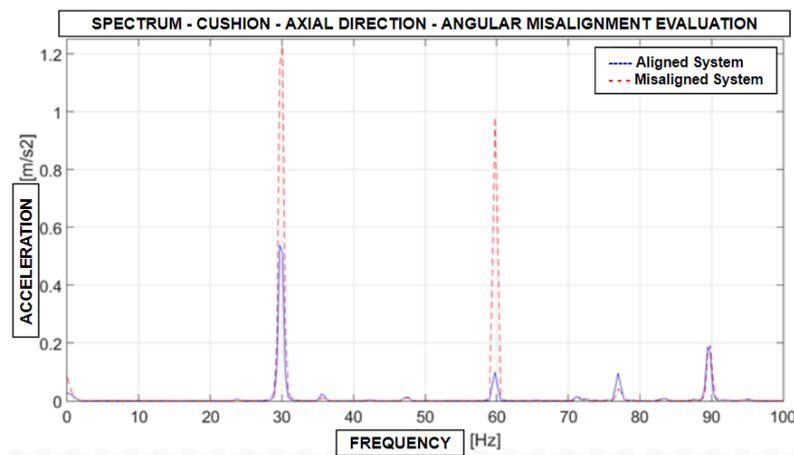


Figure 7 – Results for angular misalignment identification

3.1.2 Parallel Misalignment

Different from the previous defect, it was possible to identify the parallel misalignment for both systems, assembled with threaded bars and cushions. According to SCHEFFER AND GIRDHAR, P. 96, the identification of this type of defect relies on the rise of the radial vibration level at 2x rpm or 60 Hz. The results for the threaded bars assembly are presented in Fig. 8, a sharp rise in the vibration level is indicated at 60 Hz.

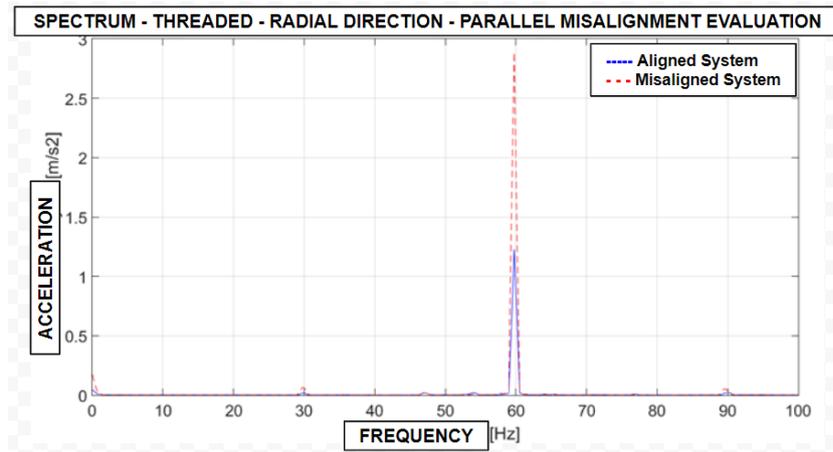


Figure 8 – Results for parallel misalignment identification - Threaded Bars

The results for the cushion assembly are presented in Fig. 9, an increase in the vibration level is indicated at 60 Hz, this represents a parallel misalignment defect. It is important to note that the net frequency is equivalent to 2x rpm or 60 Hz. This might have influenced in the results by increasing the readings at 60 Hz.

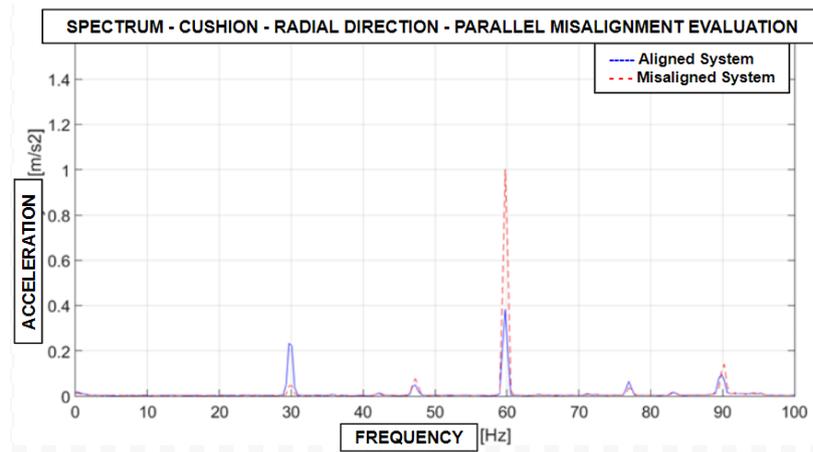


Figure 9 – Results for parallel misalignment identification - Cushions

3.2 Unbalance Defect Identification

The results show that it was possible to detect unbalance defects in both systems. According to SCHEFFER AND GIRDHAR, p.90, the vibration level at 1x rpm or 30 Hz should sharply increase for an unbalance defect. It is presented in Fig. 10 the result of the experiment that simulated an unbalance situation.

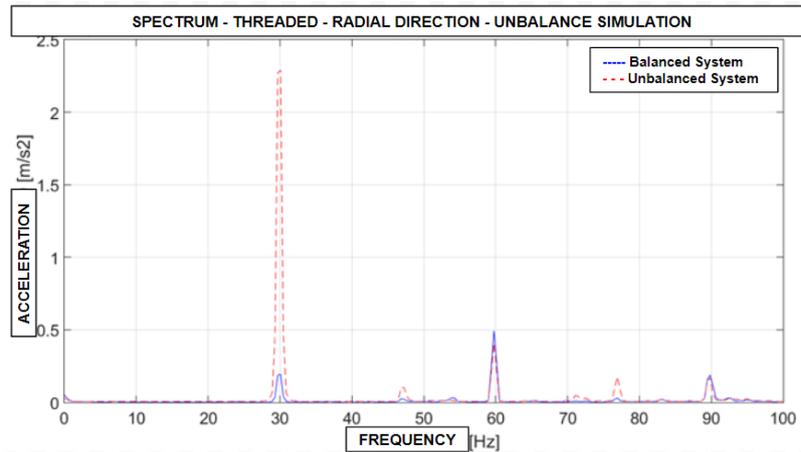


Figure 10 – Results for unbalance identification – Threaded Bars

It is presented in Fig. 11 the result of the experiment for the system assembled with cushions. The sharp rise at 1x rpm indicates the unbalance defect according to the authors cited above.

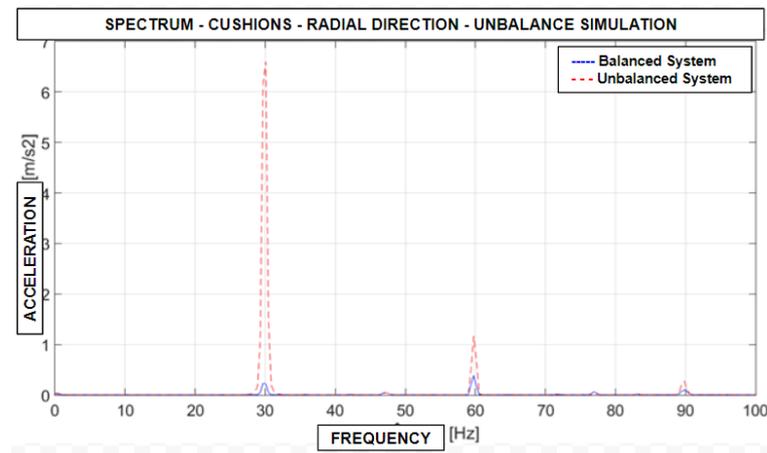


Figure 11 – Results for unbalance identification – Cushions

3.6 Frequency Response Function

In figures 12 and 13 it is shown, respectively, the measured frequency response function for the axial and radial directions, according to the Fig. 5. The peaks represent the critical frequencies of the system, i.e., the natural frequencies.

It can be seen that, for the cushion system, the response in the frequencies 1x and 2x (30 Hz and 60 Hz) is greater than that for the threaded system. This agrees with the data previously shown where the actual level of vibration is higher for the cushions assembled system than for the threaded bars.

The shape of the graphs also represents the difference between the damping capacity of the studied materials. Comparing the frequency response functions, it is possible to see that the cushion system response shows a higher damping level, that is, smoother peaks. This tends to give higher amplitudes at the anti-resonances and lower at resonant frequencies.

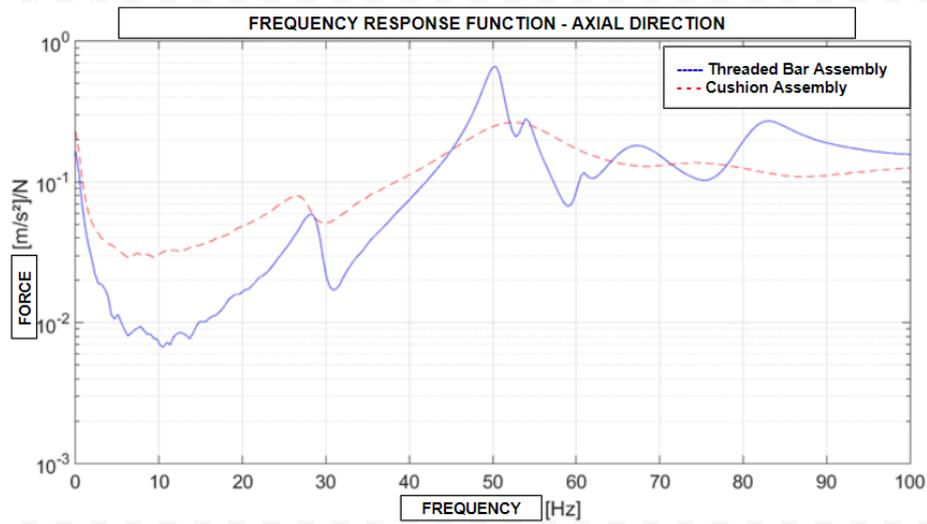


Figure 12 - Frequency response function - Axial comparison

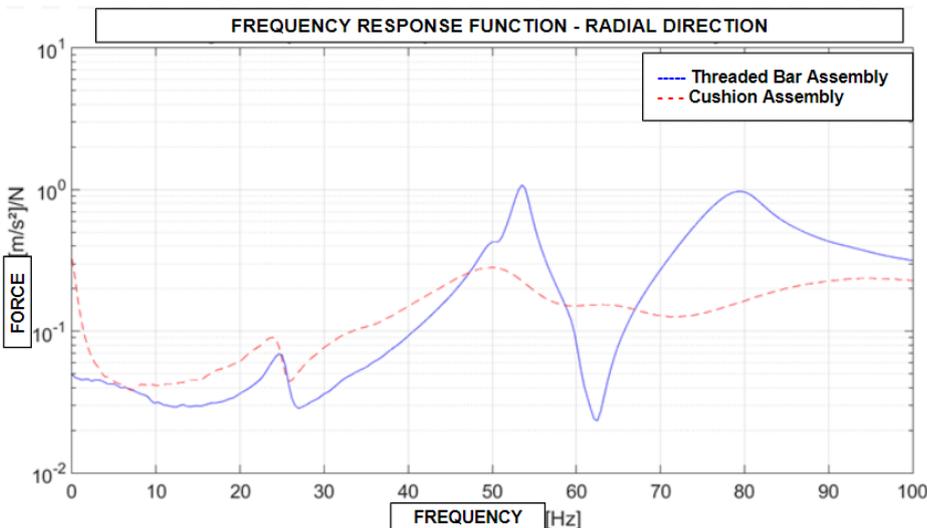


Figure 13 - Frequency response function - Radial comparison

4. CONCLUSION

The use of the didactic device has proven to be valuable for the academic community. This is because the results from the experiments made it possible to practically identify vibration defects and vibration properties. This will certainly enhance the laboratory equipment which should prepare the academic community. The possibility of change in the configuration of the equipment allows a large range of possibilities for the academic community to explore in the vibration studies along with signal analysis.

The strong features of the didactic equipment are: quick alignment process, easy alignment in both planes, vertical and horizontal, ordinary tools use, easy misalignment and unbalance defects identification.

It is suggested for further studies the effects of a frequency inverter acting in the system, the Figures 11 and 12 demonstrate the critic frequencies. Once the frequency inverter is set to these frequency it is possible for the academic community to study the effects of resonance in the system.

5. REFERENCES

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