Numerical and experimental analysis of the mechanical response of a rotatory balancing system for industrial in situ calibration

Otávio T. Krey[1], Leonel Echer[2], Boris N. Rojo Tanzi[2], Andre S. Casagrande[3], Matheus S. G. Gomes[3], Ignacio Iturrioz[2]

[1]*Simetriza Balanceamento Industrial*
Rua Padre Felipe 1777, 93280-063, Esteio/RS, Brazil
otavio@simetriza.com.br

[2]*Dept. of Mechanical Engineering, Federal University of Rio Grande do Sul*
Rua Sarmento Leite 425, 90050-170, Porto Alegre/RS, Brazil
boris.rojotanzi@ufrgs.br, leonel.echer@ufrgs.br, ignacio.iturrioz@ufrgs.br

[3]*DUO Engenharia Criativa*
Rua Vicente da Fontoura 2352, 90640-002, Porto Alegre/RS, Brazil
andre.casagrande@duoengenharia.com, matheus.gomes@duoengenharia.com

Abstract. The present work introduces the validation of an unorthodox solution for the balancing of rigid rotors: *SimMov*, a piece of customized equipment for the transport of specific machinery to carry out the balancing process in situ. For such purpose, the FE models used to assess the mechanical response of the structure are exposed. The numerical results were compared, in terms of acceleration, with experimental measurements obtained with the *SimMov* equipment. The acceleration response was also tested through standard balancers with a permanent and rigid base, which is the usual practice for similar machinery. Moreover, a simple rotor dynamics model was solved to verify the structure's critical operating behavior. These solutions were used as input data for the FE models employed to predict the structure's response. In the FE models, high-order shell elements were used to solve modal problems using the Lanczos block algorithm. The experimental results were probed and compared at critical points, predefined by the numerical models. Data acquisition was performed with six MEMS sensors (designed for industrial applications). A sampling rate of 10.00 kHz was employed. Data processing was performed using power spectral density (Welch's method). The comparison of results demonstrated the correct functioning of *SimMov* for the unbalance level considered as allowable by the machine manufacturer.

Keywords: Rotors balancing, Modal response, Signal analysis, *SimMov*.

1 Introduction

Machinery rotating parts like rotors are the means of transmitting energy and effectively carrying out work. In general, the greater the angular speed, the more critical is the mechanical balancing of such a rotating component. However, due to imperfections from the manufacturing process as well as the result of wear and corrosion during operation, it is common for rotors to be unbalanced [1].

Unbalanced components originate a series of problems, varying from light vibrations and discomfort (vibration hazards in the scenario of human-operated machinery) to excessive vibrations and critical failure of components and structures. Therefore, the balancing of rotatory components is a fundamental service to the maintenance sector of industries. Correcting unbalanced rotors is a must to guarantee safety, operation comfort, and productive indicators.

The balancing of rotatory components is a long-established process in numerous applications, ranging from small parts (e.g., high-speed automotive turbochargers) to large machine rotors, such as oil pumps, harvesters, and hydroelectric turbines. Whenever a rotor is unbalancing it takes several steps to be corrected, involving costly maintenance stops, due to dis-assembly, logistics, and re-assembly operations. In terms of medium and large industrial applications, the costs of these procedures become critical, mainly in terms of non-operational time and logistics.

Established in this context, the Brazilian company SIMETRIZA Balanceamento Industrial has developed a system (*SimMov*) for the transport of specific balancing equipment to carry out such a process in situ. Thus, all costs and risks related to the transport of the rotors to perform the balancing service are eliminated, also being possible to align the service with planned maintenance shutdowns, reducing the non-operational time due to machine downtime.

1.1 Scope and motivation

The *SimMov* system must be able to offer a balancing service with the same quality as that provided at SIMETRIZA facilities. Thus, it was necessary to develop a piece of customized equipment that facilitates the transport of specific machinery and ensures accurate measurement using balancing sensors. These requirements resulted in the design of equipment: (i) as light as possible; (ii) that enables road transport; (iii) adaptable to different types of floor, with different misalignment; and (iv) with a sufficient amount of mass to suppress vibrations. According to manufacturer' guidelines, the balancing equipment must be installed on a specific concrete floor, so the development of orthodox movable equipment that promotes similar conditions of operation was a great challenge.

SimMov was designed aiming to ensure its reliability in measuring unbalanced rotors. For such purpose, the sensors' response could not be biased due to vibrations originated or propagated by *SimMov*'s structure. Thus, it was fundamental to ensure the equipment's efficiency by validating it against conventional balancing machinery, which is the scope of the present work.

1.2 Work-flow

The present works aim to assess *SimMov*'s capability to perform the balancing process without biasing the balancing sensors' response. In other words, the *SimMov*'s structure must not propagate nor amplify the vibrations generated when dealing with unbalanced rotors. To achieve this objective, two validation steps were posed. First, numerical models of the *SimMov*'s structure were developed and compared with experimental data (obtained during the *SimMov*'s operation with a calibration rotor of known unbalance). Second, the data acquired with *SimMov* was compared with equivalent data acquired in the conventional balancing equipment installed at SIMETRIZA. During the second step, all the balancing operations were conducted with commonly unbalanced rotors. It is important to stress that the conventional balancing machinery was installed and operated following all the manufacturers' guidelines [2].

Using these two, aforementioned, validation steps, the *SimMov*'s capability to perform in situ balancing service was put to test. Such investigations are fundamental to make the *SimMov* service commercially viable. In the first validation step, the *SimMov*'s modal response was evaluated using six MEMS sensors [3] placed in key positions. The experimental results were confronted with numerical models and these results gave us the confidence to proceed to validation step two, i.e., conduct real balancing tests with unbalanced rotors at high angular speed.

The second validation step consisted on the balancing of two distinct rotors, comparing results obtained through standard balancing machinery and *SimMov*. In this step, three MEMS sensors were placed at each machinery (two pedestals and the base). A set of three stages was developed to gather enough data to validate the *SimMov*'s operation. First, the two unbalanced rotors were submitted to balancing using SIMETRIZA's machinery and *SimMov* simultaneously. Then, based on the comparison data (SIMETRIZA's machinery), the rotors went through the balancing process until reaching the balancing tolerance. Finally, the first measure phase was repeated, allowing a direct unbalancing comparison between conventional and unconventional machinery. All the data obtained over the second validation step was processed using power spectral density (Welch's method) to access *SimMov*'s efficiency.

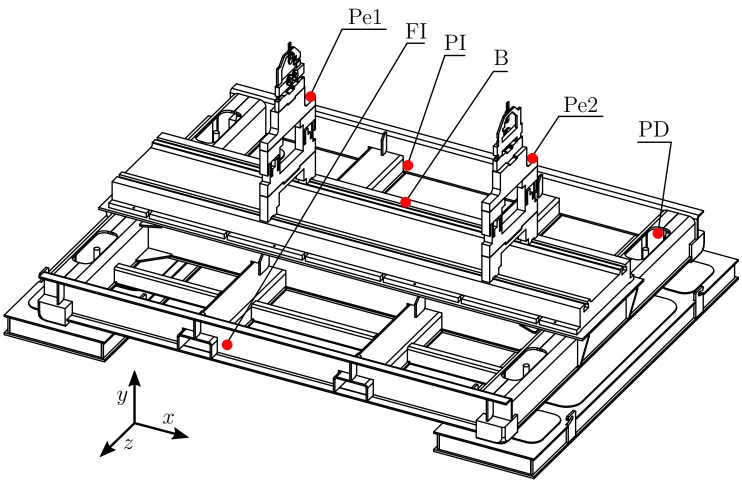
2 *SimMov* balancing system

SimMov's geometrical configuration was designed to efficiently distribute its mass and suppress the modal displacement amplitude of its first eigenmodes. The balancing machinery was positioned along with the structures central region, which is formed by a stiff longitudinal component to resist bending. This central element is connected to opposed custom structural elements on its two endings, which are also connected with each-other, such configuration results in high torsional strength [4]. The structure endings are supported by two massive blocks, which act anchoring the whole structure [5]. The connection between the anchoring blocks and the main element endings happens through four leveling devices, which secure *SimMov*'s alignment and correct attachment to the ground. The *SimMov*'s geometrical configuration is presented in Figure 1. *SimMov* was designed to support rotors with maximum length equal to $2.90\,\mathrm{m}$, max diameter of $1.60\,\mathrm{m}$, and max initial unbalance equal to 1% of the rotor's total mass (3000.00 g mm in the worst allowed scenario, i.e., total mass equal to 300.00 kg).

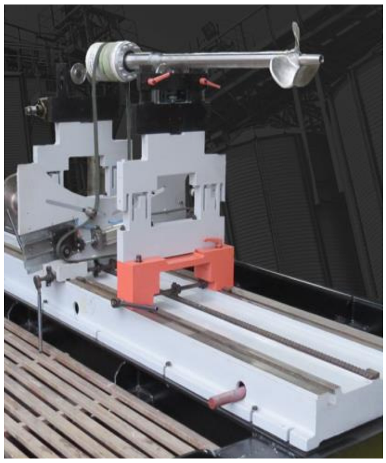
The complete structure was built with ASTM-A36 structural steel. The mechanical properties, as presented by ASM [6], were considered as follows: elasticity modulus $E = 210.00\,\mathrm{GPa}$; Poisson ratio $\nu = 0.30$, density $\rho = 7850.00\,\mathrm{kg\,m^{-3}}$, ultimate yield stress $\sigma_y = 250.00\,\mathrm{MPa}$, and ultimate rupture stress $\sigma_r = 400.00\,\mathrm{MPa}$. Plates with different thicknesses were used, the main components were built with 12.70 mm thick metal sheets, the secondary components used plates of 9.53 mm, and key components employed 25.40 mm thick metal sheets.

2.1 Structure and instrumentation

The *SimMov*'s structure is presented in Figure 1. It is important to note the presence of the complete balancing machinery [2] installed on *SimMov*. Moreover, the positioning of the sensors is illustrated as well.



(a) *SimMov*'s schematics



(b) Balancing machinery coupled to *SimMov*

Figure 1. *SimMov*

As presented in Figure 1, six different MEMS sensors were installed on *SimMov* in order to obtain the structures acceleration response when operating with unbalanced rotors. The sensors' positioning can be easily identified in the structure's isometric view, on the right of Figure 1. The nomenclature used and channels configuration, [3, 7], for each sensor is presented in Table 1.

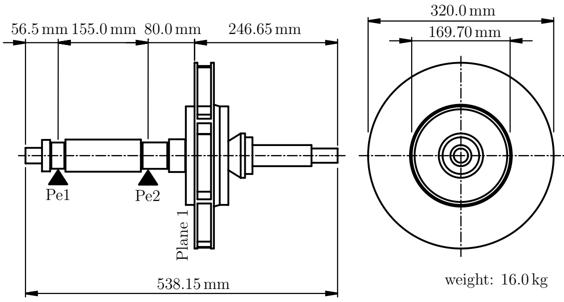
Table 1. Sensors description.

	Property								
Sensor ID	SimMov-Pe1	SimMov-Pe2	SimMov-B	SimMov-FI	SimMov-PI	SimMov-PD	Angular Velocity	Electrical current	Out of Range
Channel	15	7	14	9	2	10	0	8	1
Axis	z	z	y	z	z	x	–	–	–

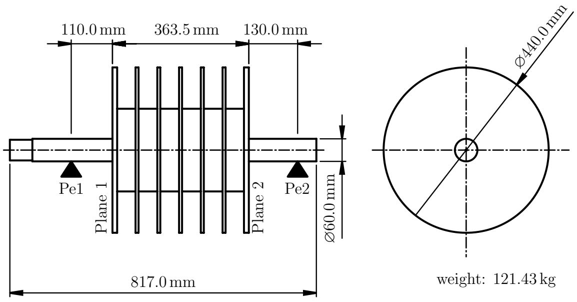
The sensors [3] counted with a resonant frequency equal to 28.00 kHz, and linear response range equal to 15.00 kHz. These sensors are designed for industrial applications and present low noise with a wide bandwidth. Its capacity is limited to one-axis acceleration measurements, operating with a sensitivity of $1.00\,\mathrm{mV\,m^{-1}\,s^{-2}}$

2.2 Testing and Operation

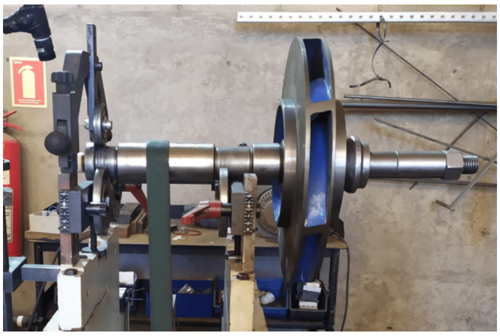
The validation tests to ensure *SimMov*'s efficiency encompassed the balancing of two distinct rotors, namely Rotor A and Rotor B. These two different unbalanced rotors are illustrated in Figure 2. For both components it is presented their schematics with all relevant geometric characteristics, Figures 2a and 2b, and the rotors as installed in the balancing machines, Figures 2c and 2c. It is important to note that Rotor A is unsymmetric and presents only one plane of balancing, plane 1, while Rotor B is symmetric and presents two planes of balancing. Moreover, the support positions for each rotor are indicated, Figure 2c, as Pe1 and Pe2.



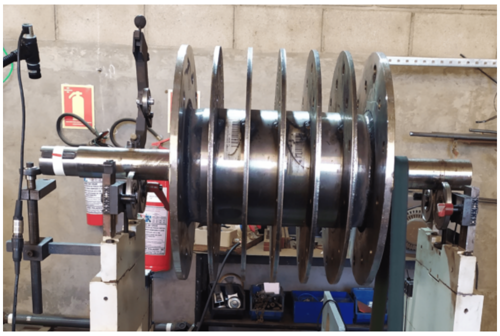
(a) Rotor A - schematics



(b) Rotor B - schematics



(c) Rotor A - balancing setup at *SimMov*



(d) Rotor B - balancing setup at *SimMov*

Figure 2. Rotors balancing being conducted using unconventional machinery: *SimMov*

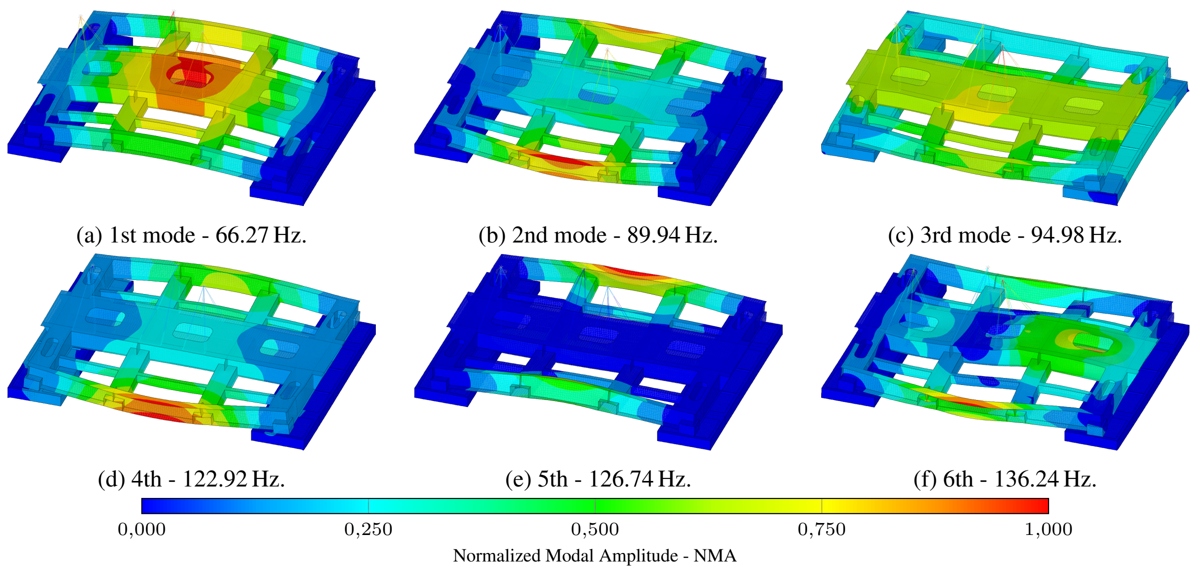
The logical sequence employed in the balancing process followed the stages presented in Section 1.2. The balancing operation was conducted with an angular velocity of $\approx 600\,\mathrm{min}^{-1}$ for Rotor A and $\approx 470\,\mathrm{min}^{-1}$ for Rotor B. Rotor A counted with an initial unbalance of $\approx 55\,\mathrm{g}$ and Rotor B $\approx 99\,\mathrm{g}$. The balancing measurements/corrections continued until a minimal residual unbalance was reached. For Rotor A, this parameter was equal to $2\,\mathrm{g}$ and for Rotor B it was $5\,\mathrm{g}$.

3 Results

The structure's modal response was obtained through numerical simulations using a commercial FE software [8]. Bi-quadratic shell FE were used to model the whole structure. The balancing machinery (motor included) was modeled with rigid links and concentrated masses, which were included in the center of gravity of such components. Block Lanczos algorithm was employed to solve the resulting eigenvalues problem. The solver was set to extract 270 modes, from which 90 were expanded. Boundary conditions consisted of restricting the vertical displacement, degree of freedom in the direction perpendicular to the ground plane (simple-supported). The free-body modal response was evaluated as well, these results are not shown, however, these eigenvalues are included in the spectrum curves, Figures 4 to 7.

The modal response, for the first six eigenmodes, considering simple-supported BCs, is exposed in Figure 3. The region of maximum modal amplitude occurred near the structure center of gravity for mode one, and in the lateral components for all the subsequent five modes. Modes one and three were dominated by bending, while the other four modes presented torsion or the coupling between bending and torsion. Even the lower eigenvalues occurred past the operation limits for the balancing machines ($1500\,\mathrm{min}^{-1}$ or $25\,\mathrm{Hz}$).

The numerical results presented by Figure 3 were compared with the natural frequencies response obtained



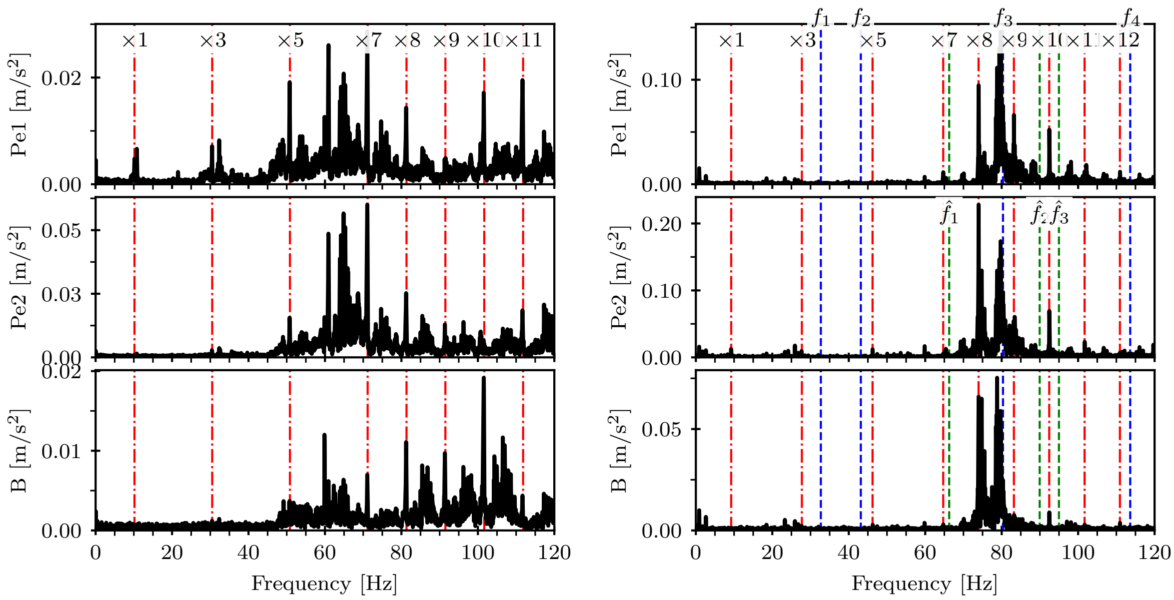
(a) 1st mode - 66.27 Hz. (b) 2nd mode - 89.94 Hz. (c) 3rd mode - 94.98 Hz.

(d) 4th - 122.92 Hz. (e) 5th - 126.74 Hz. (f) 6th - 136.24 Hz.

Figure 3. Modal response for the complete structure of *SimMov* coupled with dynamic balancing machinery.

experimentally, by means of impact test. The first three modes were clearly present in the spectral response for the impact tests, while modes four to six were present but not plainly discernible.

The balancing tests with Rotors A and B were conducted after gaining confidence regarding the *SimMov*'s adequate modal response, past (at least 2.50 times greater) the machinery operation range. The signal response in the frequency domain for the balancing of Rotor A is presented in Figure 4. These curves correspond to the sensors' signal coupled to the pedestals of the balancing machinery and the base of the structure. Data were processed using power spectral density (Welch's method) and only the signal fraction corresponding to a constant angular velocity region was used, this velocity and its harmonics are illustrated with red dashed vertical lines. The following results correspond to the structure's response prior to the rotor balancing, i.e., rotor's initial state.

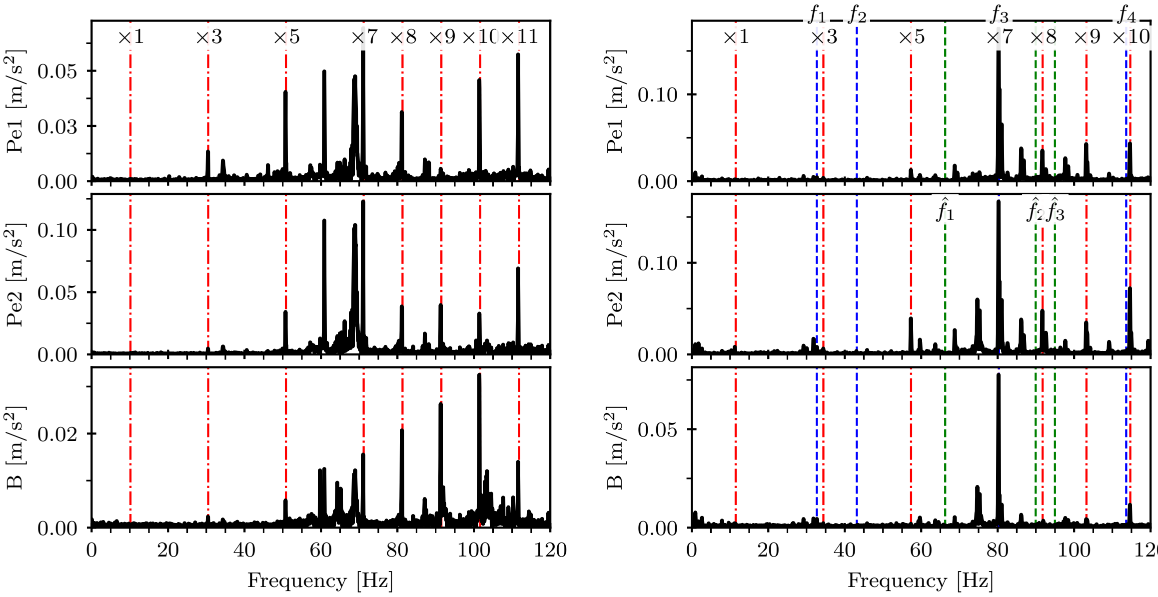


(a) Initial SIMETRIZA. (b) Initial *SimMov*.

Figure 4. Balancing testing - Rotor A, initial.

Results indicated a considerable vibration, due to rotors unbalance, with the whole structure responding at lower frequencies. Moreover, *SimMov*'s response is similar, even better, to the one presented by conventional balancing machinery. The frequency response for *SimMov* is also overlapped with its natural frequencies (for the scenario of free body BC) dashed green lines.

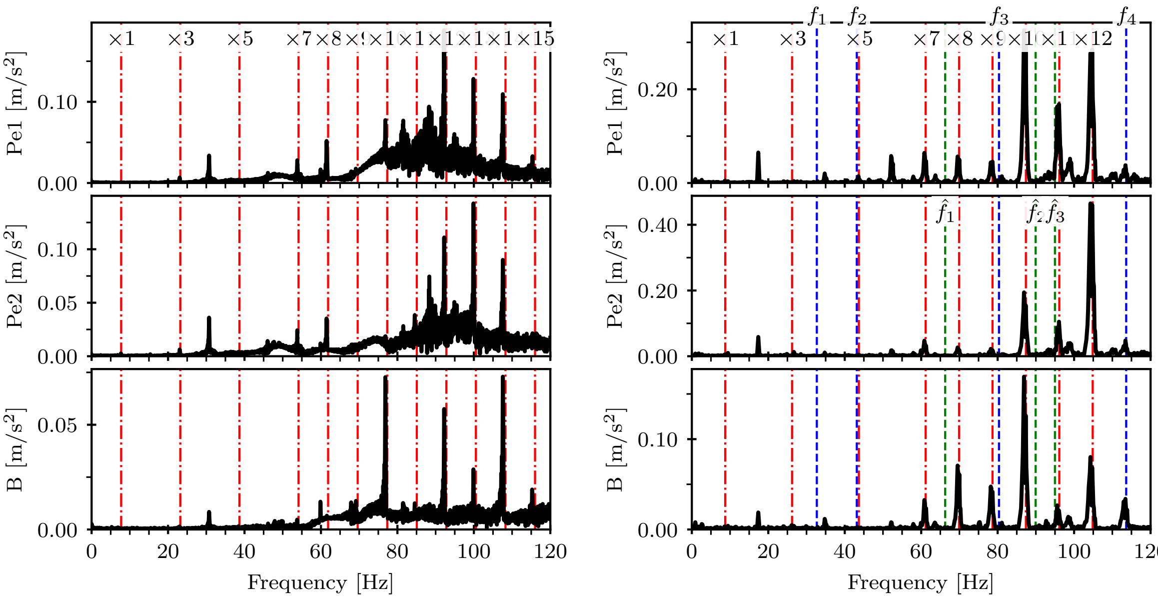
The balancing corrections were applied to Rotor A and the balancing test was redone. The results regarding the tests with Rotor A after balancing, final state, are presented in Figure 5.



(a) Final SIMETRIZA. (b) Final *SimMov*

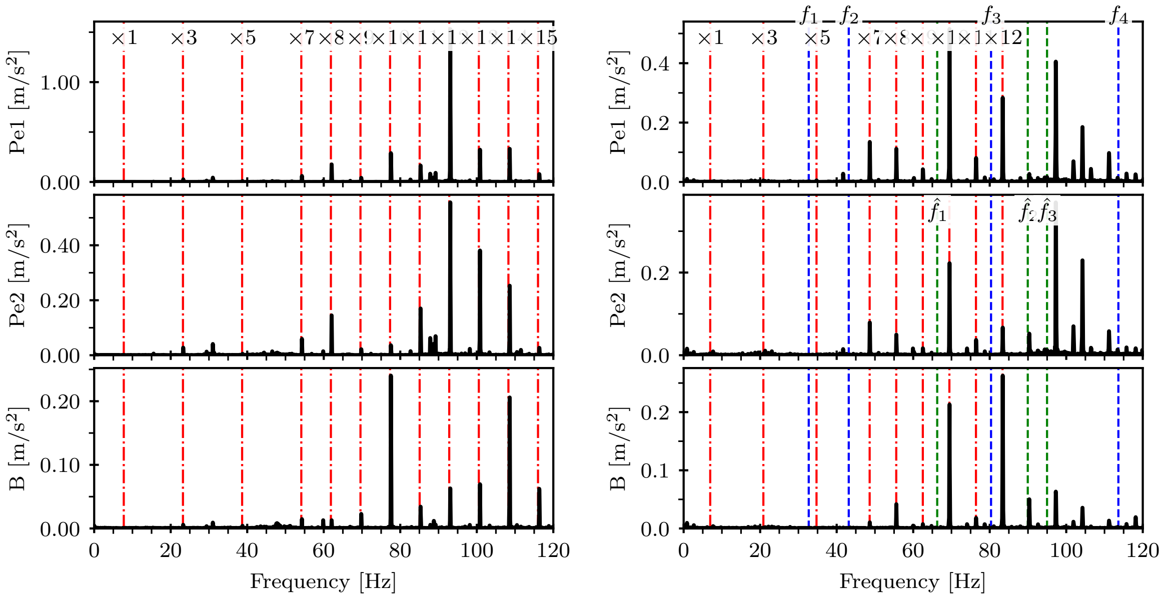
Figure 5. Balancing testing - Rotor A, final.

The improvement in the structure's response is clearly visible with considerable suppression of the frequencies amplitude. Moreover, *SimMov*'s response is at least as good as the one obtained with conventional machinery. Analogously to the aforementioned results, Figures 6 and 7 presents the data post-processed after the balancing tests for Rotor B. Similar discussions to the ones previously presented, for Rotor A, could be draw for Rotor B.



(a) Initial SIMETRIZA. (b) Initial *SimMov*

Figure 6. Balancing testing - Rotor B, initial.



(a) Final SIMETRIZA.

(b) Final *SimMov*.

Figure 7. Balancing testing - Rotor B, final.

4 Conclusions

The *SimMov*'s structure presented the expected behavior with modal amplitude being suppressed by the combination of high rigidity connection components and overweight applied directly on the two opposite pedestals, acting as anchors. The experimental results proved the reliability of the numerical ones. Furthermore, *SimMov*'s response proved its efficiency to be used as an alternative to conventional balancing machinery.

Acknowledgments. The authors are thankful to SIMETRIZA - Balanceamento Industrial for financially supporting the present work. We also express our gratitude to Prof. Jorge D. Rieira for the valuable discussions regarding the evaluation of the stiffness and mechanical strength of granular soil.

Authorship statement. The authors hereby confirm that they are the sole liable persons responsible for the authorship of this work, and that all material that has been herein included as part of the present paper is either the property (and authorship) of the authors, or has the permission of the owners to be included here.

References

[1] S. Tresser, A. Dolev, and I. Bucher. Dynamic balancing of super-critical rotating structures using low speed data via parametric excitation. *Journal of sound and Vibration*, vol. 415, pp. 59–77, 2018.
[2] Schenck. *Horizontal Balancing Machines*. Schenck RoTec GmbH, Darmstadt, DE, 2013.
[3] Analog Devices. *ADXL1003: Low Noise, Wide Bandwidth, MEMS Accelerometer*. Analog Devices. Rev. 0, 2018.
[4] J. D. Riera. Avaliação do projeto: Desenho provisório de base para fabricação unidade móvel simetriza. Technical Report PRT1, Laboratório de Dinâmica Estrutural e Confiabilidade / UFRGS, 2020.
[5] G. Bycroft. Forced vibrations of a rigid circular plate on a semi-infinite elastic space and on an elastic stratum. *Philosophical Transactions of the Royal Society of London. Series A, Mathematical and Physical Sciences*, vol. 248, n. 948, pp. 327–368, 1956.
[6] I. H. C. ASM. *Properties and Selection- Irons, Steels, and High-performance Alloys*, volume 1 of *ASM handbook*. ASM International, Russell Township, 10th edition, 1990.
[7] Omega. *OM-USB-1608FS-PLUS: 8-channel 16-bit multifunction USB data acquisition module*. Omega Engineering, Inc., Stamford. Users Manual, 2016.
[8] ANSYS®. Academic research mechanical, release 17.0. Ansys Inc., Canonsburg, Pennsylvania, 2017.