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Structural and dynamic evaluation of a test bench for mechanical vibrations using finite elements

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Abstract

This numerical study was conducted to assess the experimental bench structure's adequacy for measuring mechanical vibrations. The bench allows for three types of tests: dynamic imbalance of a single flywheel, dynamic imbalance of dual flywheels, and dynamic imbalance due to defective gears. A CAD (Computer-Aided Design) model of the vibration bench was developed in three distinct configurations. We then conducted structural (static) and dynamic (non-linear) analyses to assess the structural integrity of the bench, its components, and the expected vibration amplitudes during testing. Additionally, the structure's natural frequencies were determined, and their compatibility with the expected excitation frequencies was confirmed. These assessments were executed using finite element models within the Solidworks Simulation™ framework. The findings indicate that, when designing such machinery, it is crucial to ensure that the vibration modes do not coincide with the excitation frequencies, as this can compromise the experimental results. Consequently, we have proposed and accessed two redesigned versions of the bench to ensure structural integrity and prevent resonance during experimental procedures. The redesigns aim to mitigate the risk of resonance by altering the bench's structural parameters to shift the natural frequencies away from the range of excitation frequencies. This proactive approach is based on the idea that a big difference between the natural and excitation frequencies lowers the chance of resonance, which makes the results of the experiment more reliable. The enhanced designs were subjected to a series of rigorous simulations to validate their performance, ensuring that they meet the stringent requirements for educational and research applications.

Keywords: Computational simulation, Frequency response analysis, Machine element analysis, Mechanical vibrations, Modal analysis, Nonlinear dynamic analysis, Structural design, Structural stability analysis, System dynamics.

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Authors' Contributions: Spearheaded the study's conceptualization and led the simulation efforts, which encompassed both structural and dynamic analyses, L.H.M.P.; creating the CAD models of the vibration bench across its various configurations and assisted in interpreting the results of the simulations to assess the bench's performance, J.G.A.M.; played a key role in analyzing the outcomes and formulating proposals for the bench's redesign, D.C.P.M. All authors have read and agreed to the published version of the manuscript.
Transparency: The authors confirm that the manuscript is an honest, accurate, and transparent account of the study; that no vital features of the study have been omitted; and that any discrepancies from the study as planned have been explained. This study followed all ethical practices during writing.
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1. Introduction

The interpretation of vibration signals enables predictive maintenance and failure moments prediction [1, 2], which offers a cost advantage over preventive and corrective maintenance [3, 4]. Test benches for rotating machinery facilitate the study of rotor dynamics via vibrational response spectrum analysis [5-8]. Condition monitoring, a system analysis technique, incorporates this process to predict faults. Structural issues in an experimental bench designed for measuring mechanical vibrations can result in resonance and data variation [9, 10]. To ensure reliability and accuracy, bench verification and structural redesign are essential. This guarantees that the sensor-captured values reflect only the dynamic imbalances generated within the system, unaffected by extraneous factors. Their goal was to induce faults through wear and tooth breakage, as well as capture frequency diagrams of mechanical vibrations for both faulty and non-faulty gears using spectral techniques.

García Jara and Cerón Chalacán [11] designed and constructed an experimental bench to detect faults in straight and helical gears. Their goal was to induce faults through wear and tooth breakage and to capture frequency diagrams of mechanical vibrations for both faulty and non-faulty gears using spectral techniques. The results demonstrated a shift in the spectral band when such faults were present García Jara and Cerón Chalacán [11]. Delgado Barrera [12] also developed an experimental bench for gear fault detection and trained a neural network model with the experimental data to identify the type of damage in gears within rotating machinery. The findings indicated that this model is highly effective in detecting and identifying gear faults Delgado Barrera [12]. Labrador [13] conducted a theoretical-experimental correlation study on gear calculations using various analysis methods, including transmission error, tooth deformation, and lifespan analysis. The results suggested that while computational modeling is robust, it does not always accurately predict the variables in question. Nevertheless, theoretical models remain a valuable predictive tool Labrador [13]. Moreno-Sánchez, et al. [14] created an experimental bench to examine the impact of imbalance and misalignment on mechanical vibration in bearings, with the aim of facilitating early diagnosis. Their research revealed that the acceleration measurements typically used for bearing fault diagnosis are insufficient, leading them to propose an improved method Moreno-Sánchez, et al. [14]. Dávalos Ramírez, et al. [15] designed and constructed a test bench for measuring stresses and vibrations in rotating machinery, specifically in shafts with gears. This test bench is applicable for designing new gears, analyzing gears in operation, or diagnosing faults with fewer components than those reported in previous studies [15].

Moral [16] designed and constructed an experimental bench for measuring vibrations in machinery, featuring seven configurations and a data acquisition system. The bench's structure underwent static analysis using Solidworks™ Moral [16]. Medina and Casanova [17] developed a virtual model to functionally verify a mechanical vibration test bench for rotor imbalance measurement. Utilizing Pychrono™ and finite element formulation, they simulated the bench, which included a shaft with a mass-carrying flywheel and two displacement sensors in orthogonal directions. Jeffcot rotor theory was applied to calibrate the computational model, which accurately replicated the physical phenomena within the system, maintaining error values below 10% in most instances Medina and Casanova [17]. Martínez Palmeth, et al. [18] employed finite element methodology to validate an experimental device for mechanical evaluation, addressing both structural integrity and functional performance. Their methodology entailed conducting static analysis for structural assessment and nonlinear dynamic analysis for functional evaluation Martínez Palmeth, et al. [18]. Bernal Calderón and Cortés Navarrete [19] performed simulations on a didactic mechanical vibration test bench to discern the structural behavior. CAD modeling facilitated a more accurate representation by incorporating materials, operational frequencies, and experimental setup fixtures. They utilized NX™ for modeling and executed static and dynamic analyses to determine stresses and deformations across various experimental configurations [19].

This work aims to conduct modal analysis, static analysis, and nonlinear dynamic analysis on an experimental bench designed for measuring mechanical vibrations. The objective is to ascertain whether the bench's structure satisfies the requisite conditions for conducting experimental procedures or if its configurations possess a natural frequency that closely aligns with the operational frequency of these procedures. If the frequencies match too closely, this study will suggest necessary redesigns to rectify the bench's structure.

2. Materials and Methods

The study was structured into phases. Phase I involved ensuring that the bench's main structure met the necessary structural requirements. Phase II assessed three configurations of the bench: a single flywheel with imbalance, two flywheels with imbalance, and an imbalance in spur gears due to a split tooth. The analysis began with CAD modeling of the experimental bench, including the main structure and experimental configurations. We then developed a finite element model of the bench, which included mesh sensitivity analysis for each configuration, static analysis to assess stresses, deformations, and displacements under static load, modal analysis to estimate the first five natural frequencies and non-linear dynamic analysis to calculate deformations and displacements during bench operation. The final step involved processing the results and comparing them with the minimum acceptable design conditions for an experimental vibration bench. Phase III consisted of executing the final simulations with the newly proposed design to verify compliance with the established requirements.

Static analysis is utilized to ascertain the stresses, strains, and displacements of structural elements under analysis due to applied loads. Displacements for high-precision structures, like an experimental bench for instrument readings, should not surpass0.000001 to 0.00001 inches per inch. Modal analysis aims to gather information on the first five modes and natural frequencies of the structure and each experimental configuration, determining if the structures operate too close to their natural frequencies, which is undesirable as it amplifies vibrations. A system's working frequency should be atleast 50% above or below the closest natural frequency. The criteria for the experimental bench's suitability, which requires a rigid structure, are established as follows: a natural frequency of 120Hz or higher and limiting deformations to a maximum of 0.000001mm/mm, thereby classifying the structure as high precision and ensuring its supports can dampen external vibrations affecting the system under evaluation [20].

We derived the CAD model of the experimental bench from an existing assembly that used1-inch-wide angle structural profiles made from AISI 1020 material, along with 16-gauge sheets (with a thickness of 1/16 inch) composed of A-36 structural steel. The creation of the CAD model was accomplished using Solidworks™ software. The bench structure features a drawer at the bottom, with the experimental configurations and the electric motor positioned on the top of it to the right. Both the described assembly and the CAD model developed in this study are depicted in Figure 1.

Figure 2 illustrates the CAD models of the three experimental setups. Each one has pedestal-type bearings, support shafts, and rotating elements that representspecific failure conditionsthat are being tested, like flywheels or gears.

The Solidworks Toolbox™ component facilitated the creation of the gears, including standardized components like straight-tooth gears. Table 1presents the data used to precisely represent this configuration within the existing experimental bench. In our analyses, we employed second-order tetrahedral (3D) elements with 16 integration points. Meshing was based on curvature, which controls the minimum element size. Mesh-sensitivity analyses were conducted across all studies.

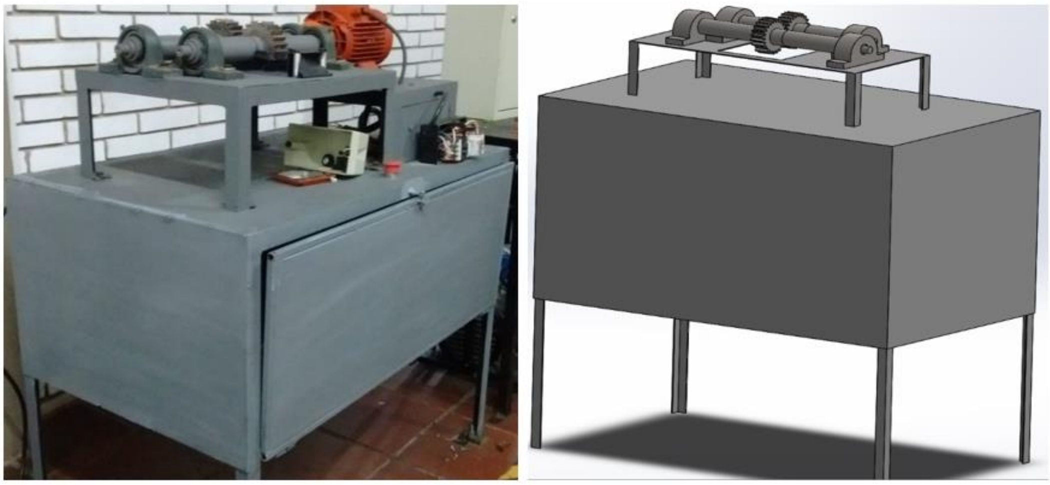


Figure 1.
Existing experimental bench and its CAD model.

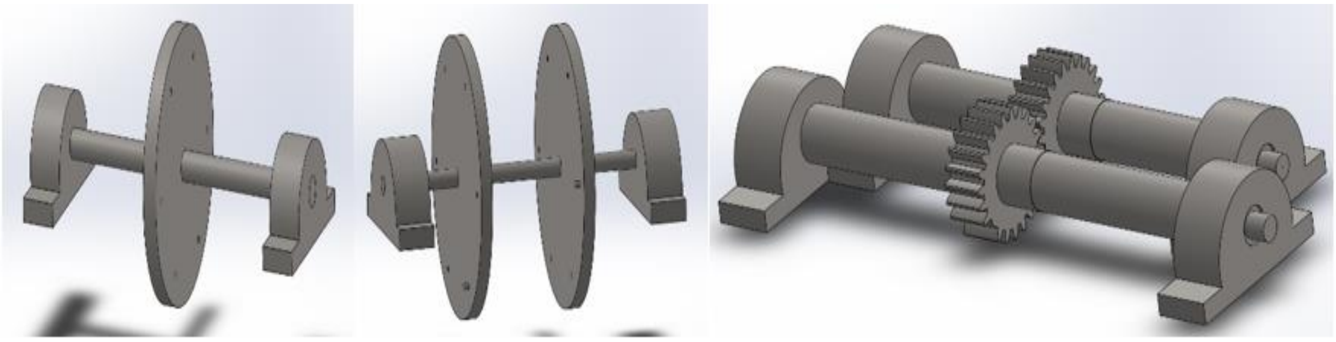


Figure 2.
CAD models of the experimental configurations.

Table 1.
Characteristics of the gears.

Element	Description
Module	4
Numberofteeth	23
Facewidth	35mm
DP - pitch diameter	92mm
CD - outerdiameter	100mm
DR - innerdiameter	82mm

3. Results

3.1. Structural Analysis of the Experimental Bench

Both static and modal analyses were performed to glean insights into the behavior of the experimental bench structure during practical experiments. We applied the following conditions to both types of analyses: Fixed supports on all four faces of the structure's legs; gravity acting in the vertical plane (negative Y direction); rigid global contact; Rayleigh damping of 0.03 in the structure; the material for the entire structure was low-carbon simple alloy steel (AISI 1020) with an elastic, linear, and isotropic behavior model. All numerical models were three-dimensional.

Additionally, for an unbalanced mass (m), the centripetal force (F) generated due to its rotation at an angular velocity (ω), with a radius ®, can be calculated using Equation 1.

$$F = mr\omega^2 \quad (1)$$

Table 2 provides the values used in the computational simulations for the parameters of Equation 1, which were obtained experimentally.

Table 2.
Un-balancing mass according to the configuration of the bench.

	With a flywheel	Withtwoflywheels	Spurgears
r	0.171	0.171	0.100
m	0.014	0.014	0.012
ω	6.280	6.280	6.280
F	0.015	0.030	0.008

Before conducting static, dynamic, and modal analyses, we performed a mesh sensitivity analysis, recognizing that finite element model results depend on the mesh size used.In Figure 3, the calculated value of the structure's first natural frequency, which serves as a monitoring parameter, is plotted against the number of elements used in the mesh of the models. According to the figure, a minimum of approximately 24,000 elements should be employed to model the bench structure, ensuring that the results remain unaffected by the mesh size during analysis.

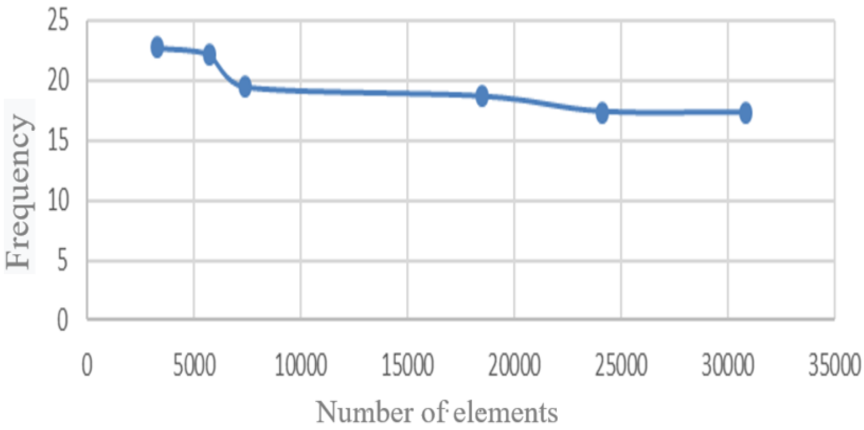


Figure 3.
Mesh sensitivity analysis for the mechanical vibration bench structure.

During the static analysis, we utilize the maximum estimated centrifugal force value obtained from experimental tests: 0.0301 N. This value corresponds to the dual flywheel unbalance configuration, in addition to the weight of the experimental setup. The static analyses were conducted using Solidworks Simulation™ with the FFEPlus solver, which is recommended for problems with numerous degrees of freedom (DOF). We applied a global friction coefficient of 0.05 to the model. Figure 4 shows the stress and deformation fields caused by the maximum unbalance force in the dual flywheel configuration. We take into account the influence of gravity and use full constraint boundary conditions to secure the structure to the floor. Additionally, a load is applied to the top of the experimental bench, representing the weight of the experimental setups and the unbalanced inertial forces. These forces vary for each experimental configuration and exhibit dynamic behavior.

The finite element model assumes that low-carbon steel makes up all parts of the structure and experimental setups. Based on the obtained results, it is evident that the structure does not exceed the yield strength limit of AISI 1020 material (351 MPa), and the maximum stress is 7.4 MPa. Furthermore, we obtain a total deformation value of 0.00001723 mm/mm.

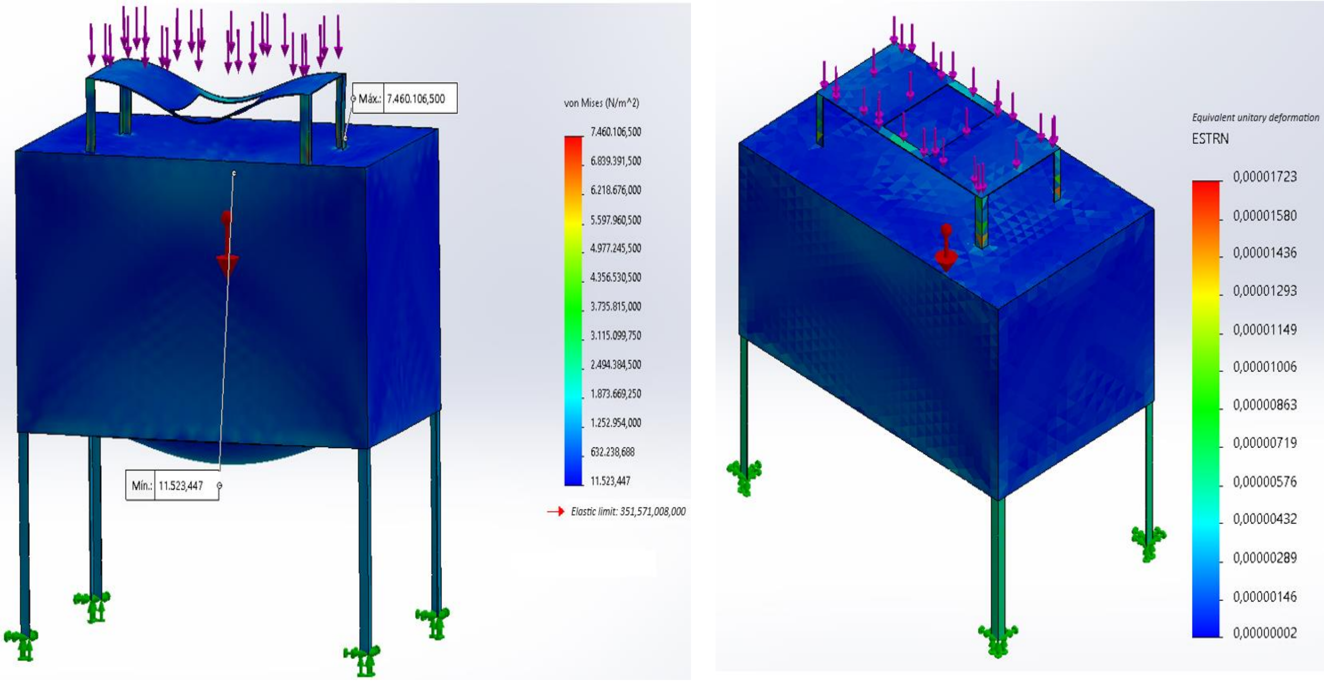


Figure 4.
Von Mises stress and displacement results for static analysis in solid works™.

We conducted modal analyses to verify the first natural frequencies of the structure. These analyses simulated load in the model, considering only the weight—an approach customary for this type of analysis. We employed the DIRECT SPARCE solver from SolidWorks™. We reviewed the first six natural frequencies and applied fixed boundary conditions to the bench's legs. Table 3 presents the values of the first natural frequencies of the structure obtained through simulation.

The mechanical vibration bench structure operates at a frequency that is very similar to its natural frequency. This alignment occurs because the electric motor used in the experimental practices rotates between 30 Hz and 60 Hz. Specifically, Table 3 reveals that the first natural frequencies of the experimental bench range from 17 Hz to 52 Hz in SolidWorks™. This observation indicates that the structure operates within the motor's working frequencies, consequently impacting the results of the experimental practices.

Table 3.
Values of the first natural frequencies [Hz].

Mode	Benchstructure
1	17.31
2	17.52
3	29.48
4	37.25
5	40.21
6	52.00

3.2. Modal Analysis of the Experimental Configurations

For each of the experimental configurations, we conducted modal analysis, nonlinear dynamic analysis, and motion studies. To ensure that the results of these analyses remain unaffected by the element size resulting from discretization, we performed a mesh sensitivity analysis beforehand. Figure 5 illustrates the variation of the first natural frequency against the number of elements in the model. Notably, the single flywheel configuration requires a minimum of 100,000 elements,the dual flywheel configuration require 50,000 elements, and the gear configuration requires approximately 280,000 elements. This mesh density ensures that the results remain uninfluenced by the element size.

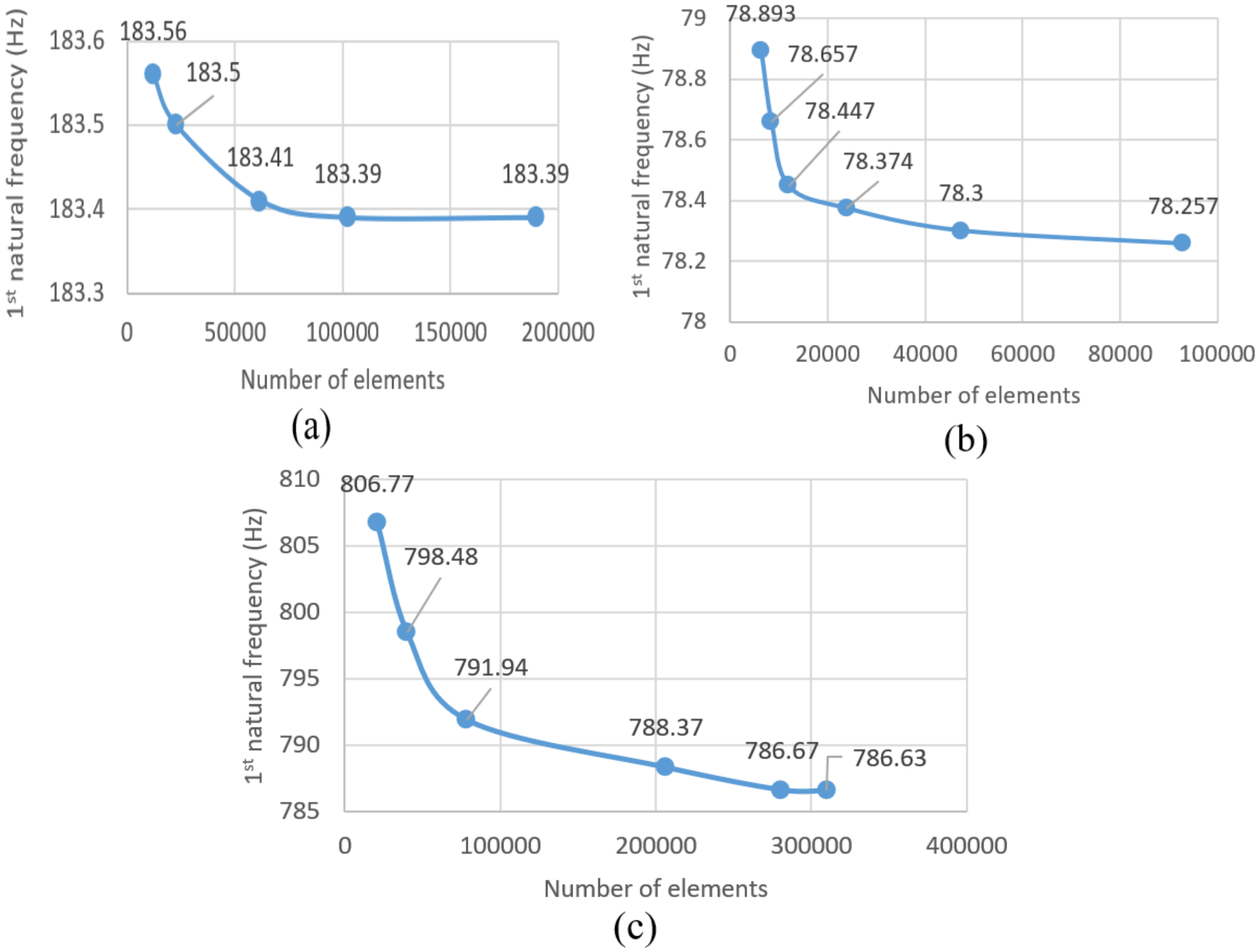


Figure 5.
Mesh sensitivity analysis for the three experimental configurations: Single flywheel balancing (a), dual flywheel balancing (b), and gears (c).

To perform the modal analysis, it is only necessary to define the supports for the experimental configurations. In this case, a fixed face was used at the base of the bearings (refer to Figure 6). The material to be used was also specified, which, in this instance, is all AISI 1020 steel. For this type of analysis, the most critical property is the material density. We conducted modal analysisfor all three experimental configurations. Figure 6 presents the results obtained using Solidworks™ software for the following modes of vibration:Second mode of vibration is for the single flywheel configuration; Third mode of vibrationisfor the dual flywheel configuration; First mode of vibration is for the gear configuration.

Table 4 provides the values of the first five natural frequencies for each experimental configuration. The results demonstrate that the experimental configurations do not exhibit a natural frequency close to the operating frequency range of the electric motor, which spans between 30 Hz and 60 Hz. Therefore, design flaws or resonance phenomena will not affect future experimental measurements conducted with these configurations.

Now, let’s delve into the configurations used in the computational model for nonlinear dynamic analysis. For this type of analysis, we employed a period from 0 to 1 (pseudo-time). We utilized a formulation that accounts for large displacements and deformations. The solution technique involved an iterative approach using the Newton-Raphson method, integrated with the Newmark method.

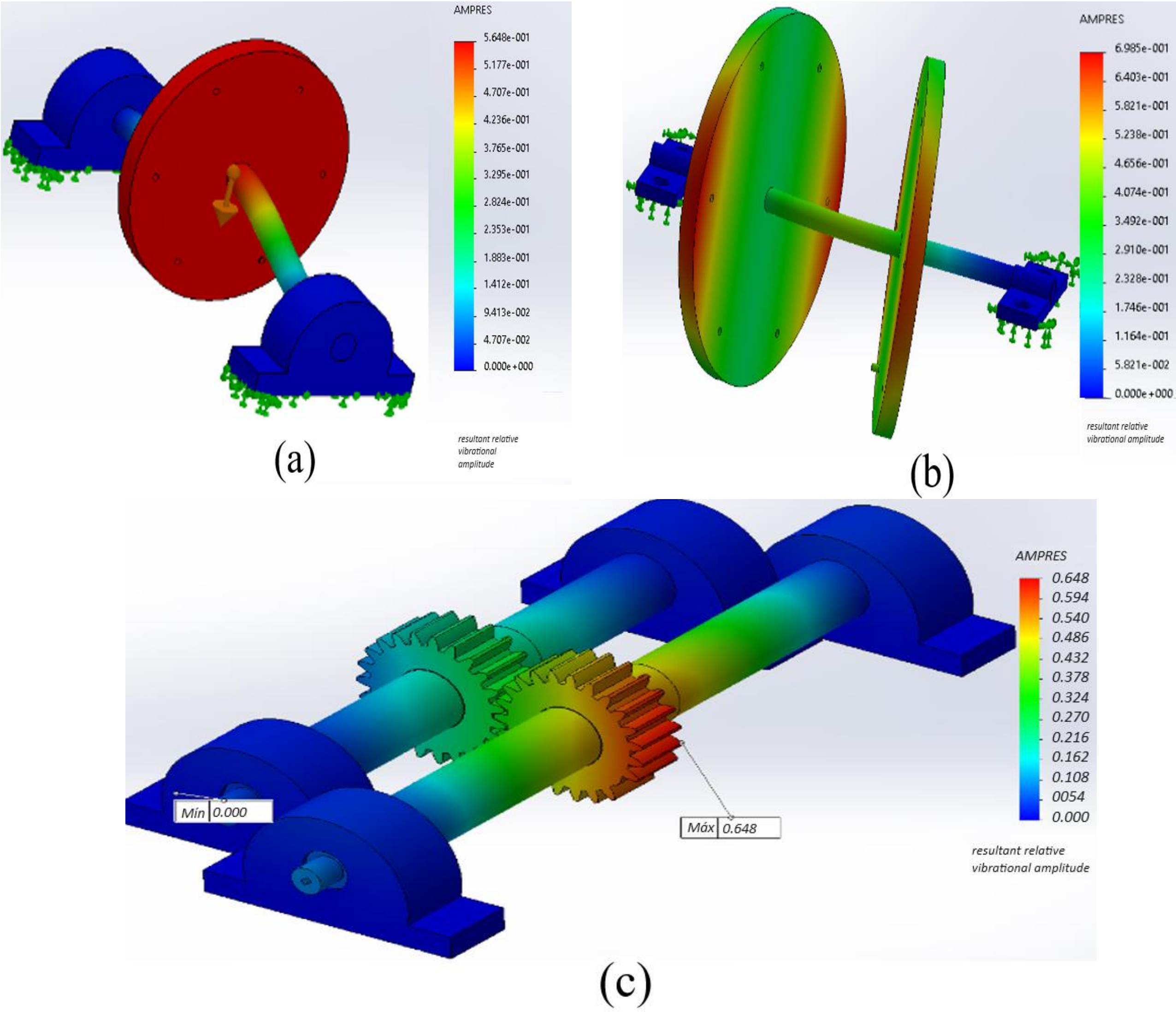


Figure 6.
Modal analysis results. Configurations: (a) Single flywheel, (b) Dual flywheels, and (c) Gears.

Table 4.
Modal analysis results for the configurations. Values of the first natural frequencies [Hz].

Mode	Single flywheel	Dual flywheels	Gears
1	189.69	78.25	899.34
2	308.36	118.17	1018.1
3	309.33	140.14	1343.2
4	568.62	140.25	2082.4
5	569.56	320.16	2313.8

The dynamic analysis incorporated Rayleigh viscous damping, with damping coefficients α and β set to 0.03, as is common when working with steel structures. We conducted all simulations at a rotational speed of 60 Hz, representing the bench’s highest operational speed and the point where the most significant inertial or unbalanced forces occur. The acceleration due to gravity is assumed to act in the vertical direction, which, in the case of the model, aligns with the negative “Y” axis. The bearings are considered fixed or rigidly connected to the main structure, which we model as having fixed lower faces (refer to Figure 7). For nonlinear dynamic analysis, material density remains a crucial factor, and in this instance, we use AISI 1020 steel.

As anticipated during rotation, the experimental configurations exhibit radial displacements (in the case of flywheels and gears). In Figure 7, the deformed experimental configurations are evident due to unbalanced forces. Table 5 provides a summary of the maximum displacement values obtained from simulations in two areas of the model. The experimental test bench should measure these values. Specifically, the measurement areas are on the bearings and on the flywheels or gears (the locations of the rotating elements).

Table 5.
Nonlinear dynamic analysis results. Maximum displacements [μm].

Location	Single flywhee	Dual flywheels	Gears
Bearing	0.30	0.03	9.10
Shaft	1.71	3.03	8.45

3.3. Redesign of the Mechanical Vibration Bench Structure

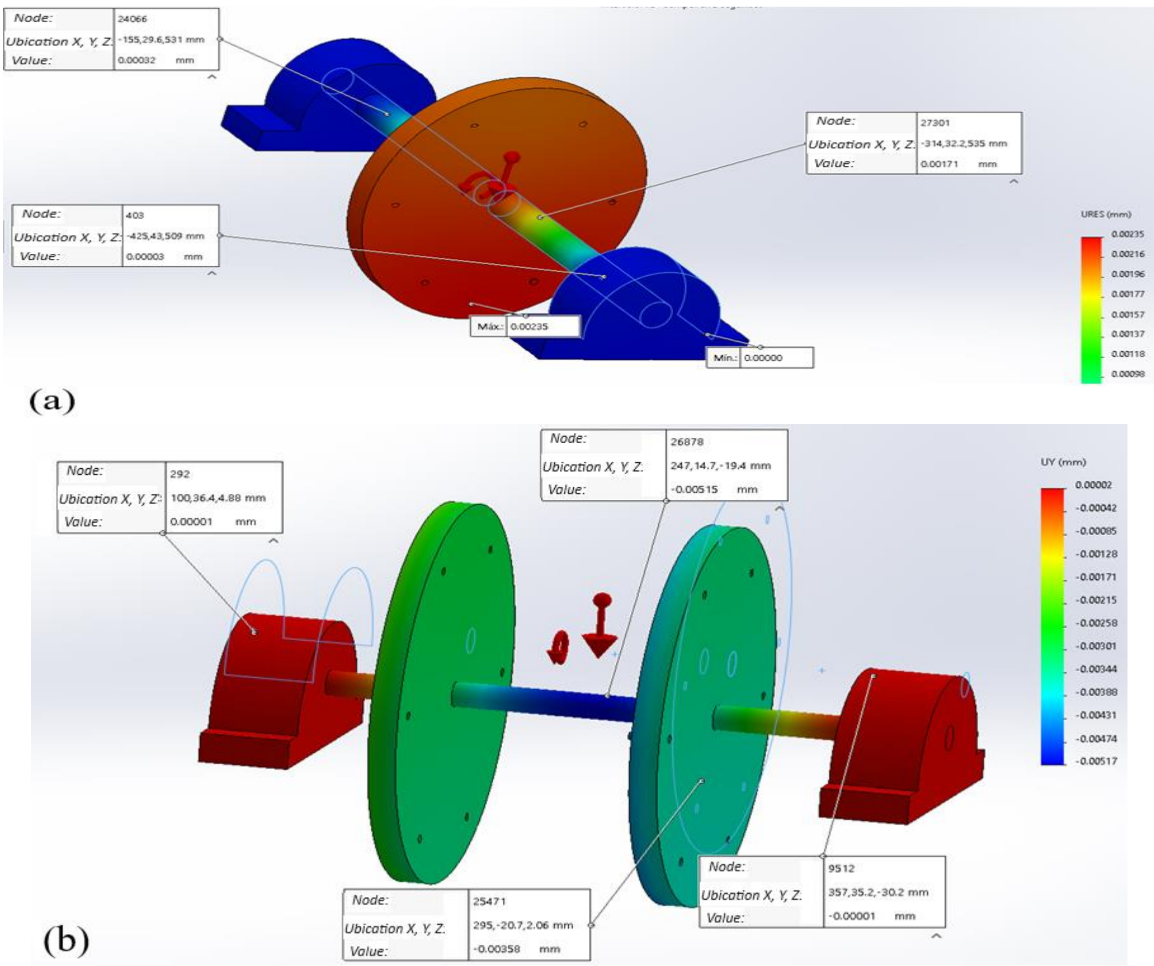
Based on the previously obtained results, there is a need to modify the current design of the experimental bench. Modification to shape, mass, or damping (either active or passive) can control and reduce vibration in structures [21]. In this work, we propose two lines of solution:

- Enhancing the Existing Structure: Our first approach aims to improve the existing structure by enhancing its rigidity. This enhancement will increase the natural frequencies and reduce undesired displacements.
- Proposing a New Structure: Our second approach entails creating an entirely new structure for the experimental bench.

Through a redesign of the structure, our goal is to enhance its performance. Currently, the structure lacks sufficient rigidity in its lower part and in the upper section, where the experimental configurations are supported. To address this, we propose modifications for each part of the structure. Specifically, we recommend reinforcing the structure's legs using one-inch angle profiles arranged in a cross shape at the front, sides, and back of the structure (as depicted in Figure 8 (a)). Additionally, we reinforce the upper part of the base that supports the experimental configurations with angles. These angles are strategically placed on the sheet where the configurations are secured, forming a grid-like pattern to prevent displacements (as depicted in Figure 8 (b)).

We examine the results of the static analysis for the redesigned structure in Figure 9.These results reveal a significant decrease in displacements. Furthermore, we perform both modal analysis and nonlinear dynamic analysis.

Proposing a second solution, Figure 10 (a)showcases a new structure entirely made of duralumin. This slotted-type structure accommodates various experimental configurations. We conduct the three types of analysis: static, modal, and nonlinear dynamic. Specifically, Figure 10(b)presents the results of the static analysis for the newly proposed design.



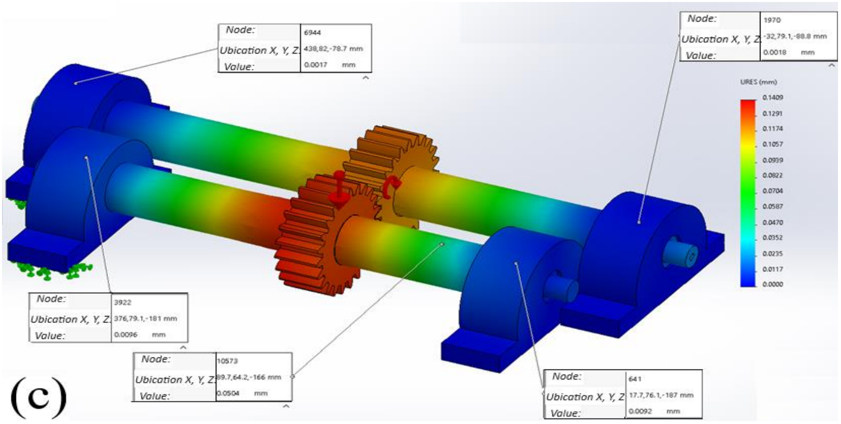


Figure 7.
Nonlinear dynamic analysis results. Configurations: (a) Single flywheel, (b) Dual flywheels, and (c) Gears.

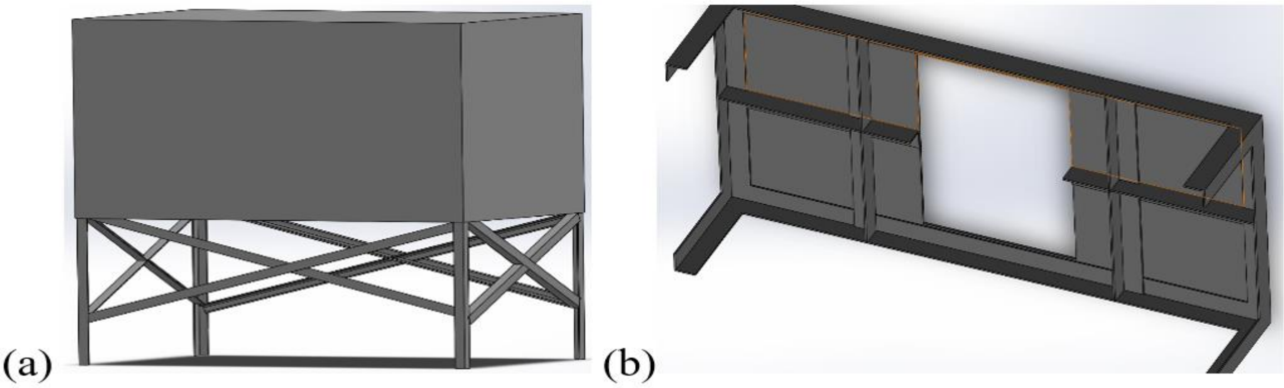


Figure 8.
Redesign of the mechanical vibration bench structure: (a) Stands, (b) Sheet.

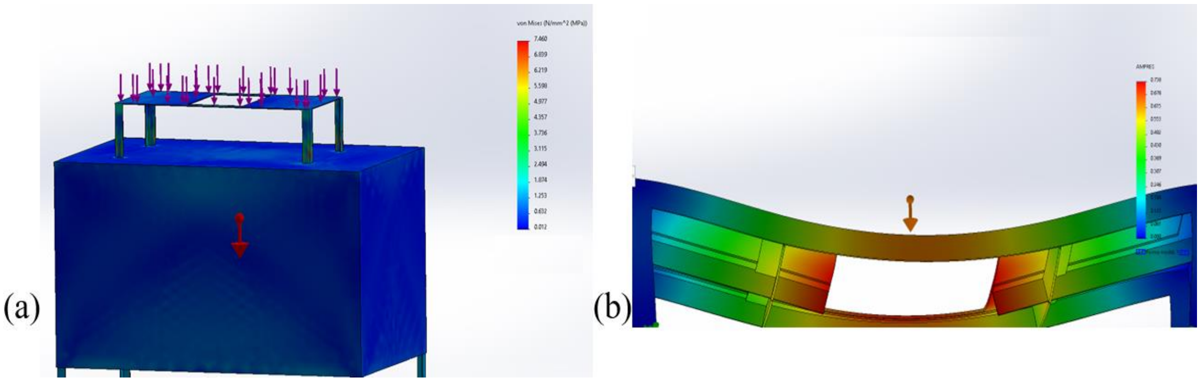


Figure 9.
Results of the static analysis of the redesign.

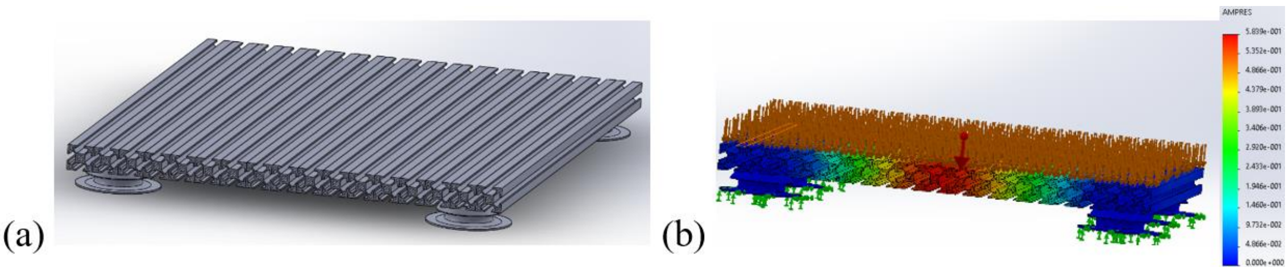


Figure 10.
(a) New design for the mechanical vibration bench. (b) Result of the static analysis.

Table 6 presents a comparison between three cases: the current structure, the redesigned structure, and the new structure.Notably, both new designs significantly reduce the maximum stresses in the structure, with natural frequencies increasing by 400% and 900%, respectively. Addressing one of the most deficient aspects of the current design, the displacements decrease by one to two orders of magnitude, signifying substantial improvement in the proposed designs.

4. Conclusions

This study emphasizes the critical importance of correctly applying all design methodologies when approaching the design of a structure or machine. Neglecting these principles can lead to devices that do not function properly under load conditions. It is essential to recognize that analyzing stresses, deformations, and displacements alone does not guarantee the structural and functional integrity of a machine. For machinery, it is essential to examine the natural frequencies of machines or structures and ensure that these frequencies remain as far as possible from potential operating frequencies. Additionally, it is crucial to recognize how computational methods enable comprehensive and straightforward analysis of all the mentioned aspects, facilitating informed decision-making throughout the design process.

Table 6.
Comparison of stresses, displacements, and 1st natural frequency for the three designs.

	Existing structure	**Redesign**	**New design**
Stress [MPa]	7.46	3.15	3.94
Displacements [mm]	0.140	0.014	0.009
1st Natural Frequency [Hz]	20.00	117.00	192.08

Leveraging Solidworks™, we gained insights into the behavior of the structure and its configurations through static, nonlinear dynamic, modal, and motion analyses. Our findings include the following:

- Structural Displacement: The structure displaces by 0.14 mm under its own weight alone. Consequently, performing experiments on it is inadvisable, as the results would be influenced by the structure itself.
- Natural Frequency Range: The natural frequency range of this structure spans between 10 Hz and 60 Hz across the first six modes. This indicates that the structure is susceptible to resonance phenomena. The experimental practices conducted on it involve the motor rotating at 15 Hz, 30 Hz, and 60 Hz.
- Motion Analysis: By analyzing the motion of the experimental setups, we pinpointed the outcomes that the mechanical vibration sensors should record, independent of the structure's influence.
- In conclusion, we propose a base and coupling structure design for the experimental configurations that meets the minimum requirements. This design ensures that the experimental practices carried out on it remain unaffected. Specifically, the proposed structure must have a natural frequency greater than twice the maximum frequency of rotation of the motor and must not deform under the weight of each configuration.

References

[1] F. Arena, M. Collotta, L. Luca, M. Ruggieri, and F. G. Termine, "Predictive maintenance in the automotive sector: A literature review," *Mathematical and Computational Applications,* vol. 27, no. 1, pp. 1-21, 2021. https://doi.org/10.3390/mca27010002

[2] T. P. Carvalho, F. A. Soares, R. Vita, R. D. P. Francisco, J. P. Basto, and S. G. Alcalá, "A systematic literature review of machine learning methods applied to predictive maintenance," *Computers & Industrial Engineering,* vol. 137, p. 106024, 2019. https://doi.org/10.1016/j.cie.2019.106024

[3] P. Henriquez, J. B. Alonso, M. A. Ferrer, and C. M. Travieso, "Review of automatic fault diagnosis systems using audio and vibration signals," *IEEE Transactions on Systems, Man, and Cybernetics: Systems,* vol. 44, no. 5, pp. 642-652, 2013. https://doi.org/10.1109/TSMCC.2013.2257752

[4] S.-j. Wu, N. Gebraeel, M. A. Lawley, and Y. Yih, "A neural network integrated decision support system for condition-based optimal predictive maintenance policy," *IEEE Transactions on Systems, Man, and Cybernetics-Part A: Systems and Humans,* vol. 37, no. 2, pp. 226-236, 2007. https://doi.org/10.1109/TSMCA.2006.886368

[5] A. Lazarus, B. Prabel, and D. Combescure, "A 3D finite element model for the vibration analysis of asymmetric rotating machines," *Journal of Sound and Vibration,* vol. 329, no. 18, pp. 3780-3797, 2010. https://doi.org/10.1016/j.jsv.2010.03.029

[6] M. S. Lebold *et al.*, "Using torsional vibration analysis as a synergistic method for crack detection in rotating equipment," in *IEEE Aerospace Conference Proceedings. https://doi.org/10.1109/AERO.2004.1368168*, 2004, vol. 6, pp. 3517–3526.

[7] S. Sendhil Kumar and M. Senthil Kumar, "Condition monitoring of rotating machinery through vibration analysis," *Journal of Scientific and Industrial Research,* vol. 73, no. 4, pp. 258–261, 2014.

[8] S. Singh and N. Kumar, "Combined rotor fault diagnosis in rotating machinery using empirical mode decomposition," *Journal of Mechanical Science and Technology,* vol. 28, no. 12, pp. 4869–4876, 2014. https://doi.org/10.1007/s12206-014-1107-1

[9] S. Peter, M. Scheel, M. Krack, and R. I. Leine, "Synthesis of nonlinear frequency responses with experimentally extracted nonlinear modes," *Mechanical Systems and Signal Processing,* vol. 101, pp. 498-515, 2018. https://doi.org/10.1016/j.ymssp.2017.09.014

[10] L. Saidi, J. B. Ali, and F. Fnaiech, "Application of higher order spectral features and support vector machines for bearing faults classification," *ISA Transactions,* vol. 54, pp. 193-206, 2015. https://doi.org/10.1016/j.isatra.2014.08.007

[11] G. García Jara and H. Cerón Chalacán, "Design and construction of a test bench for the diagnosis of failures in straight and helical gear transmission systems through comparative analysis of vibrational spectra," Thesis Presented to Obtain the Title of Mechanical Engineer. Chimborazo Polytechnic Higher School, 2021.

[12] M. Delgado Barrera, "Design and construction of test kits to diagnose faults in rotating machine gears," Thesis Presented to Obtain the Title of Mechatronic Engineer. Autonomous University of Bucaramanga. http://hdl.handle.net/20.500.12749/12162, 2020.

[13] O. S. Labrador, "Theoretical and experimental modeling of a gear transmission," Thesis Presented to Obtain a Master's Degree in Industrial Engineering. University of the Basque Country. http://hdl.handle.net/10810/34838, 2019.

[14] M. E. Moreno-Sánchez, J. A. Villarraga-Ossa, and R. Moreno-Sánchez, "Diagnosis of early bearing failures in mechanisms susceptible to unbalance and misalignment," *UIS Engineering Magazine,* vol. 18, no. 2, pp. 187–198, 2019. https://doi.org/10.18273/revuin.v18n2-2019018

[15] J. O. Dávalos Ramírez, F. Sánchez Palma, A. Villanueva Montellano, and M. A. Lira Martínez, "Assembly of a test bench to measure contact forces and vibration in pairs of gears," *Scientific and Technological Culture,* vol. 16, no. 1, pp. 12–17, 2019. https://doi.org/10.20983/culcyt.2019.1.2.2

[16] V. A. Moral, "Design of a test bench for the basic diagnosis of machines," Thesis Presented to Obtain the Title of Mechanical Engineer. Pompeu Fabra University. http://hdl.handle.net/20.500.12367/491, 2021.

[17] U. L. U. Medina and M. E. L. Casanova, "Development of a virtual test bench for the study of rotor unbalance," *Computational Mechanics Notebooks,* vol. 18, no. 1, pp. 1-11, 2021.

[18] L. H. Martínez Palmeth, M. A. Gonzales Carmona, and J. Miranda Castro, "Design and analysis of a bulge test device," *Engineering and Research,* vol. 41, no. 3, p. e85756, 2021.

[19] F. A. Bernal Calderón and D. F. Cortés Navarrete, "Simulation of a test bench for vibration analysis," Thesis Presented to Obtain the Title of Mechanical Technologist. University Francisco Jose de Calda. https://repository.udistrital.edu.co/bitstream/11349/6143/1/CalderonBernalFabioAnd, 2016.

[20] R. L. Mott, E. M. Vavrek, and J. Wang, *Machine elements in mechanical design*, 6th ed. New York: In Pearson Education, Inc, 2018.

[21] N. Karimi and R. Sarem, "Seismic response of multi-storey building using different vibration technique-A review," *International Journal of Innovative Research and Scientific Studies,* vol. 4, no. 1, pp. 1-13, 2021. https://doi.org/10.53894/ijirss.v4i1.49