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Influence of the suspension shape on the dynamic characteristics of the electrodynamic shakers

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Abstract Electrodynamic shakers are being used for years in dynamic tests of machines and devices. One of the main components of electrodynamic shakers is the armature suspension. Suspension is one of the factors responsible for the dynamic characteristics of the shaker. Various types of suspensions are used in shakers [1, 2]. Their role is to ensure the correct stroke of the armature and to keep the coil concentrically in the magnetic gap. As part of the research, the influence of the shape of the suspension springs of the electrodynamic shaker on its dynamic characteristics was evaluated. During the research, self-designed electrodynamic shaker was used, in which a suspension consisting of two-disc springs made of glass fiber with a thickness of 0.5 mm was used. Eight different spring types were prepared for the study. The tests were carried out in laboratory conditions on a previously prepared test bench. For each of the springs tested, dynamic frequency characteristics were determined for three frequency ranges: 10 – 100 Hz with a step of 10 Hz, 100 Hz – 1000 Hz and 1000 Hz – 10000 Hz with a step of 100 Hz. Studies show a significant influence of the shape of the springs used on the dynamic characteristics of the modal shaker. For the preselected springs, tests were carried out without and with a load. The characteristics determined during the research were analyzed, which allowed to indicate the optimal shape of the spring, due to the values of the generated force and the linearity of the dynamic characteristics [3, 4].

Keywords: electrodynamic shaker, spring, dynamic characteristics, construction, experimental research.

1. Introduction

Electrodynamic shakers have been used for years in modal analysis to stimulate to vibrations the structure under the test. Their main application is structural analysis, but they are also used for calibration of vibration sensors, vibration resistance tests and fatigue tests [5-7].

Typical small and medium-sized electrodynamic shakers consist of a permanent magnet or an electromagnet to create a magnetic field in which the coil moves. The coil moving in the magnetic gap is responsible for the implementation of the given forcing through the shaker armature (see Fig. 2).

Another element of the construction of modal shakers is the suspension of the armature. The springs are characterized by low axial stiffness and allow to ensure the appropriate stroke of the shaker armature. An additional role of the suspension is to hold the coil concentrically in the magnetic gap. According to the dynamic characteristics of the shakers included in the manufacturers technical documentation [8], the suspension of the shakers allows one to obtain a useful excitation frequency range of up to 6.5 kHz for shakers with a force of 100N and up to 13 kHz for shakers with a force of up to 18N. An example of a dynamic characteristic of an shaker with force 20 N is shown in Fig. 1 [8].

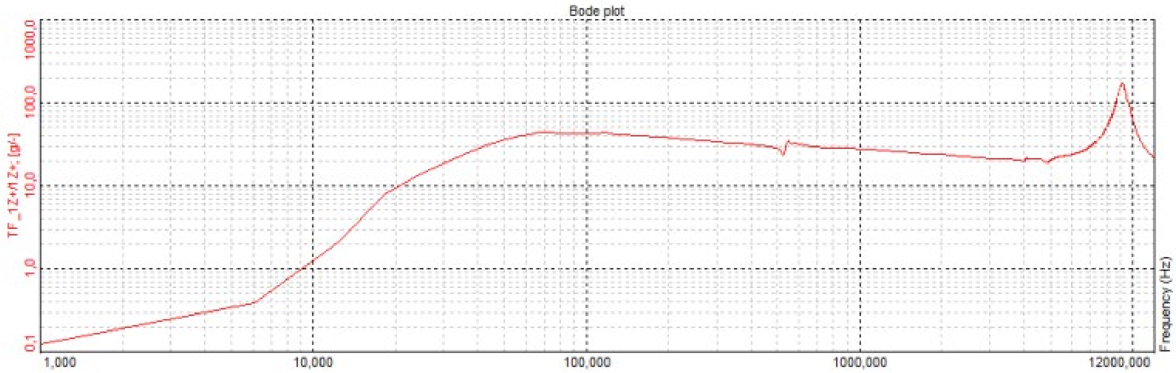

Figure 1. Example of dynamic characteristic of an shaker with a force 20 N [8].

The suspensions differ in the design of the spring, the material used, as well as in the number and placement of the springs. There are different types of suspension solutions [2] the main ones are:

- strip springs made of metal or composites (see Fig. 3),
- pneumatic (air cushion) (see Fig. 3).

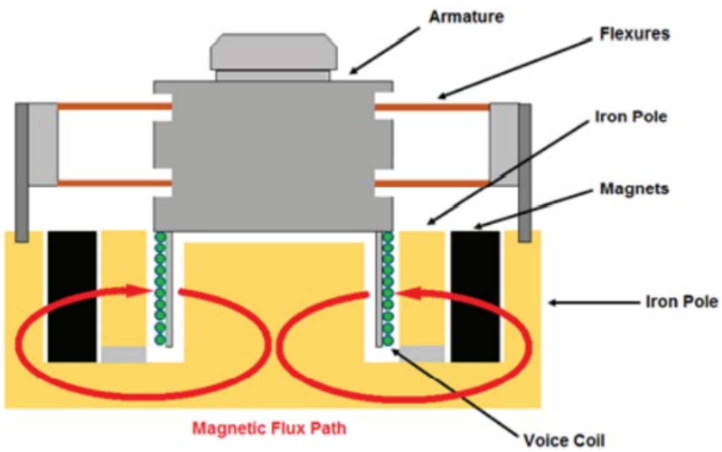

Figure 2. General structure of the shaker [2].

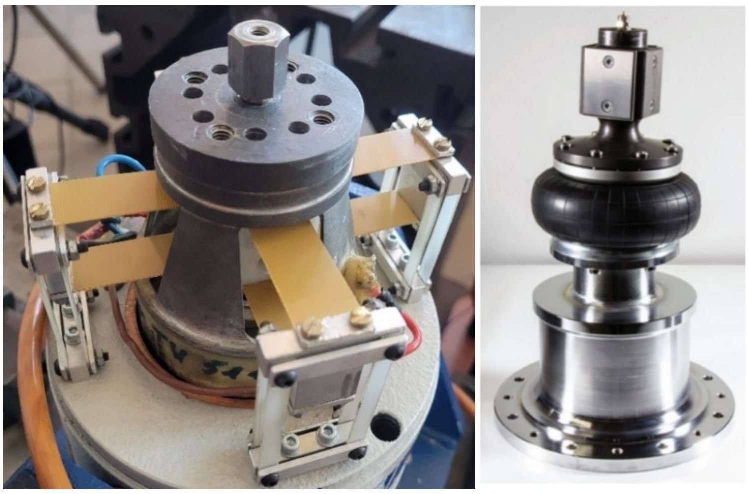

Figure 3. Examples of shaker armature suspensions [2].

During the authors research an electrodynamic shaker with disk springs (Fig. 8) suspension was designed. Optimisation of disk springs was main aim of presented in the article research.

2. Design and optimization of spring shapes

The designed electrodynamic shaker has a coil suspension and armature consisting of two-disc springs made of glass epoxy composite FR-4 [9] with a thickness of 0.5 mm. The selected material is characterized by very high mechanical strength with a relatively low specific weight and at the same time has high fatigue strength. Composite materials are characterized by mechanical non-linearity due to their complex structure and material properties [10-11]. The non-linear stiffness characteristics can affect the amplitude-frequency characteristics of the modal inductor resonance. Both springs have the same geometric dimensions, however for the purposes of research, structures have been developed that differ in the topology of geometric shapes inside the disk (see Fig. 4). The topology of the tested springs was arbitrarily selected based on experience and results of topological optimization carried out during simulation studies using CAD software.

Topological optimization was performed for static analysis. During topological optimization, the assumed value of the shaker suspension stroke and the value of the force resulting from the parameters of the shaker's electromagnetic system were determined. The main criterion for optimization was to achieve low spring stress values. The model obtained with the help of topology optimization was subjected to editing and adapted to the manufacturing process. The results of the optimization shape and the target spring are presented in Fig. 4.

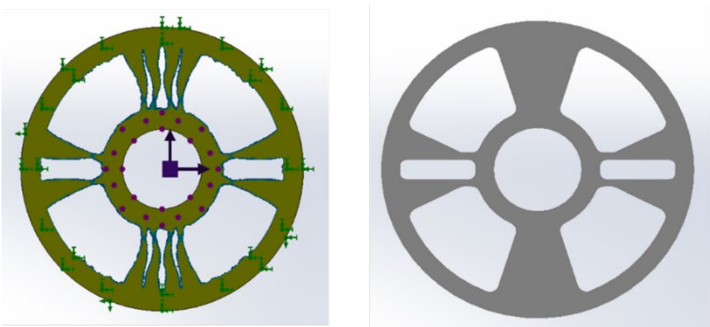

Figure 4. The process of processing the model obtained because of topology optimization.

Both springs designed based on topological analysis as well as structures developed on the basis of own experience were subjected to endurance simulation in FEM software. Each of the spring designs was subjected to a static axial force of 15N directed downward. The purpose of the simulation, apart from assessing the state of stress and indicating the weakest places of the structure, was to determine the deformation value in the context of the assumed maximum stroke of the shaker armature of 4mm. Based on the strength analysis, corrections were made to the spring geometries to ensure the required range of armature displacements and the best strength properties. By introducing modifications, attempts were made to maintain similar stiffness of the springs. Examples of the results of the strength analysis of selected springs are presented in Fig. 5. As a result of design and optimization work, eight types of springs were developed, the topology of which is presented in Fig. 6. The springs were subjected to a comparative assessment based on the dynamic characteristics of the shaker in which they were mounted. Characteristics were determined during experimental research.

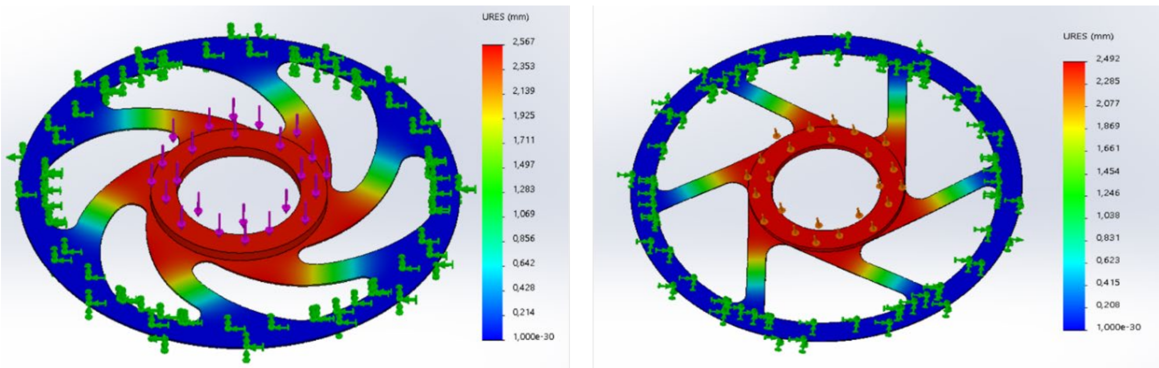

Figure 5. Displacement value results for FEM analysis.

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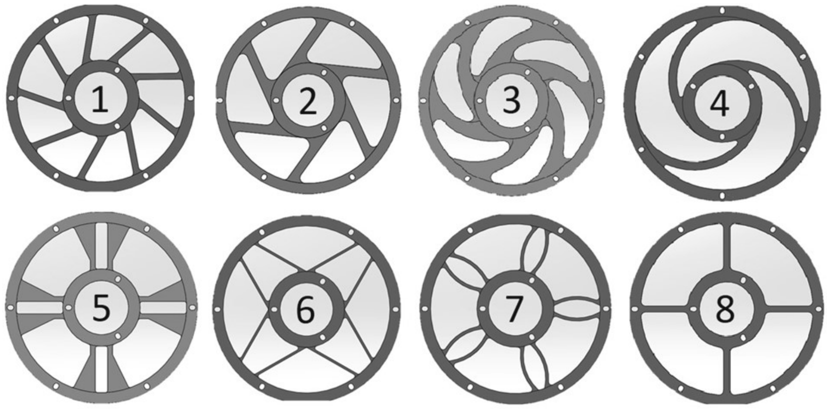


Figure 6. Developed suspension shapes.

3. Test bench

In order to conduct experimental research aimed at determining the dynamic characteristics of the shaker with designed springs, a stand consisting of:

- test shaker adapted to the installation of designed springs,
- power amplifier,
- excitation signal generation system,
- vibration measurement system of the shaker table,
- computer with software.

The measurement and excitation signal generation system was developed on the basis of the National Instruments CompactDAQ platform consisting of compactDAQ chassis cDAQ-9174, C Series Voltage Output Module NI-9263 output signal card for excitation signal generation, analog input card C Series Sound and Vibration Input Module NI-9234, to operate with the T352C34 piezoelectric accelerometer from PCB with a sensitivity of 100 mV/g . The accelerometer was used to measure the vibrations of the shaker table in the frequency range 0.5 Hz - 10,000 Hz. The CompactDAQ platform was operated under control of software running in the Matlab environment. The developed software allowed to automatic control of vibration tests as well as collection of measurement data and determination of dynamic characteristics. The block diagram of the test bench setup is presented in Fig. 7.

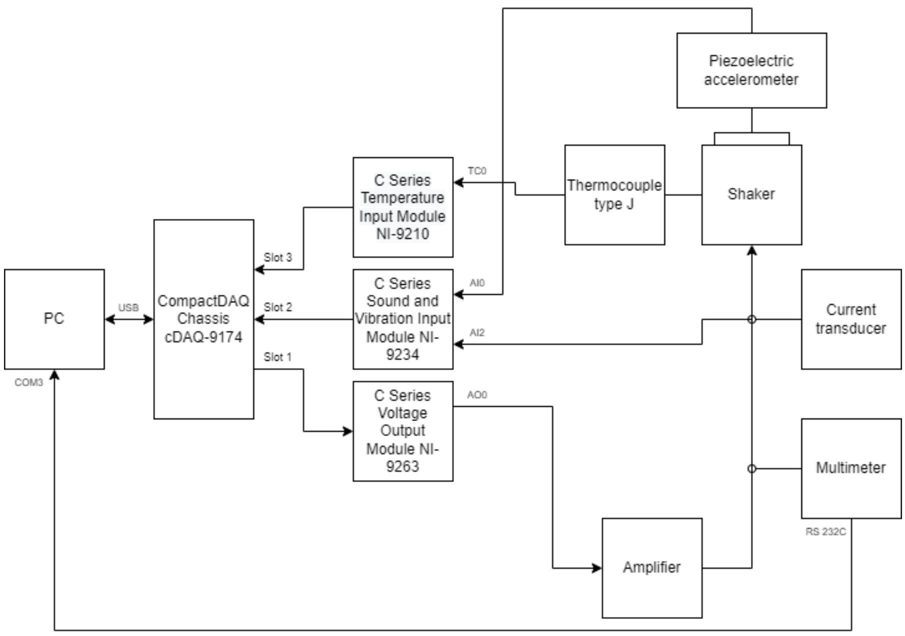


Figure 7. Diagram of the research station.

3.1. Test Plan

Research consisted on supplying the test shaker with a harmonic signal of variable frequency and simultaneous measurement and analysis of the shaker table vibration in order to identify dynamic characteristics. During each test, the shaker had different suspensions depending on the shape of the springs used. The research was carried out for two configurations of the working axis of the shaker armature: vertical and horizontal. The frequency of the harmonic signal supplying the shaker was changed incrementally according to test plan (see Tab 1). Tests were carried out for different signal amplitudes at the input of the shaker amplifier. For selected suspensions, the tests were also carried out with an additional load on the armature of 0.8 kg. In the case of some suspensions, the springs were mounted in an opposed configuration, shown in Fig. 8. The full matrix of the experiment is presented in Tab. 1.

Table 1. Experiment matrix.

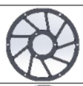
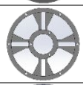
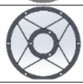
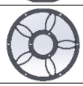
Suspension spring type		Spring configuration	Frequency ranges	Frequency jump	Input signal amplitude	Extra weight 0,8kg	Orientation of shaker axis
Shape 1		Opposed	10–100 Hz 100–1000 Hz 1000-10000Hz	10 Hz 1 Hz 1 Hz	5 V		Vertical Horizontal
Shape 2						X	
Shape 3						X	
Shape 4							
Shape 5		Compatible					
Shape 6							
Shape 7							
Shape 8							

Figure 8. Presentation of the opposing installation of springs.

4. Analysis of the collected data

Acceleration signals recorded during the experiments were processed and analyzed, which allowed one to determine the vibration acceleration peak amplitude as a function of the frequency of the excitation signal. An example of the identified characteristics (Bode plots) in the 10-10000Hz band for each type of suspension is shown in Fig. 9. The identified frequency response of the shaker was analyzed, which consisted of indicating the modes with the highest amplitude. In addition, several modes with amplitudes of 50% and 20% of the maximum amplitude value, respectively, were established. For the frequency range 1000-10000 Hz, the first threshold value was 45%. The analysis was carried out in the frequency bands 10-100 Hz,100-1000 Hz,1000-10000 Hz and the results are presented in Tab. 2.

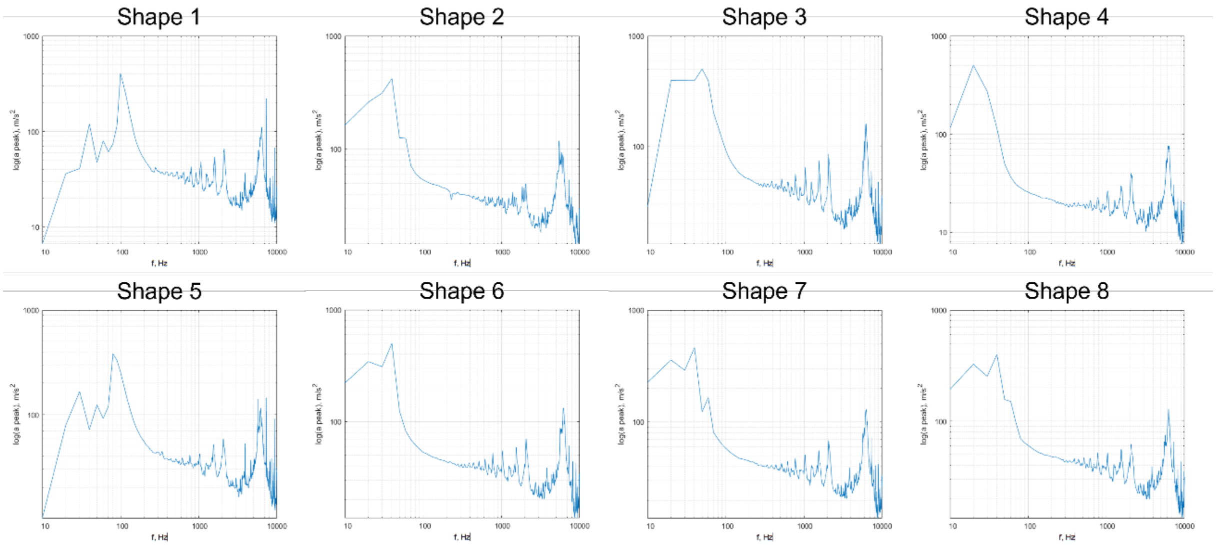
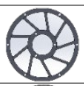
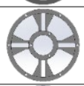
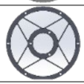
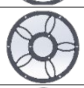


Figure 9. Designated characteristics Suspension.

Table 2. Frequency analysis of springs.

Shapes		Mode 1		Frequency band					
				10 – 100 Hz		100-1000 Hz		1000-10000 Hz	
		f, Hz	A (peak), m/s^2	50%	20%	50%	20%	45%	20%
Shape 1		93	471.67	2	2	0	0	1	3
Shape 2		45-46	502.52	3	3	0	0	0	1
Shape 3		48	502.52	11	3	0	0	0	6
Shape 4		20	502.50	Armature collision					
Shape 5		78	408.06	1	4	0	0	0	11
Shape 6		40	500.55	3	2	0	0	0	7
Shape 7		41	495.94	4	3	0	0	0	3
Shape 8		43	471.58	3	5	0	0	0	4

As can be seen, the shape of the shaker suspension springs affects the value of the natural frequency and the maximum amplitude of the shaker response at this frequency. The frequencies of the first mode changed in the range of 20 Hz, -93 Hz while the corresponding amplitudes changed in the range of 408-502.5 m / s2. The maximum values of acceleration amplitudes for all suspended shapes have an average of 480 m/s2 and their dispersion is 30.5. The shape of the suspension No. 4 was not subjected to shape optimization, therefore during the tests it caused a collision of the coil with the magnet. For this reason the test for this type of suspension was abandoned. The frequency values with the highest acceleration values are in the range of 10-100 Hz, for five springs this values is in the range of 40-50 Hz. However, for the other three springs they are respectively 93, 78 and 20 Hz. From the point of view of the functionality of the shaker in its applications, it is desirable to obtain a low amplitude value during the occurrence of resonant phenomena and their occurrence outside the useful range of frequency of the shaker. In the dynamic characteristics of the commercial shaker presented in Fig. 1, we can see a rapid increase in amplitude at a frequency close to 12 kHz. In the frequency range of 100-10000 Hz, the characteristics are almost linear with minor fluctuations occurring. Below the frequency of 50 Hz, a sudden drop in amplitude is visible.

4.1. Assessment of the quality of the excitation signal

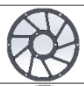
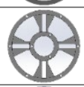
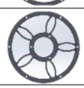
Based on the vibration acceleration signal measured on the shaker table, it was decided to assess the impact of the type of spring on the quality of the excitation signal. For this purpose, such parameters as:

- mean total harmonic distortion,
- mean value of peak accelerations,
- mean force value.

The parameter values are summarized in Tab. 3. The Table 3 also contains the standard deviations of the parameters. The most favorable values of the parameters are marked in green, while the least favorable ones in red.

After analyzing the frequency characteristics of the shaker for various springs using graphs and frequency analysis, it was decided to analyze the calculated parameters, such as: The THD (Total Harmonic Distortion) coefficient, the peak acceleration value, and the force value.

Table 3. Comparison of spring parameters.

Shapes		$\overline{\mathrm{THD}}$, dB	σ (THD)	$\overline{\mathrm{a\_peak}}$, m/s^2	σ (a_peak)	$\bar{F}$, N	σ (F)
Shape 1		-18.47	10.43	31.65	26.08	6.71	5.53
Shape 2		-15.47	8.61	36.33	25.61	7.7	5.43
Shape 3		-17.58	9.57	39.39	37.76	8.35	8
Shape 4		-14.7	8.39	21.66	10.57	4.59	4.7
Shape 5		-16.15	8.58	35.37	27.42	7.5	5.81
Shape 6		-15.71	9.11	38.54	30.63	8.17	6.49
Shape 7		-15.44	8.81	38.26	29.84	8.11	6.33
Shape 8		-16.15	8.58	35.37	27.42	7.5	5.81

The table shows a comparison of the mean values of the spring parameters. The highest mean amplitude of peak accelerations was recorded for springs No. 3, 6, 7 and No. 2. Spring No. 4 was the worst, where at low frequency values there was a collision of armature. When analyzing standard deviations, it should be noted that spring No. 2, not counting spring No. 4, has the smallest standard deviation of peak acceleration values. Summing up the comparison of the analyzes of the conducted tests of the characteristics of the springs, it can be concluded that the optimization of the springs was carried out correctly. Each of the created springs significantly exceeds the parameters of spring No. 4. Based on the research conducted, it can be concluded that two shapes of springs stand out from the others, and springs No. 2 and 3 were selected for further analysis. The comparison for the shaker characteristics of the selected springs is presented in Fig. 10.

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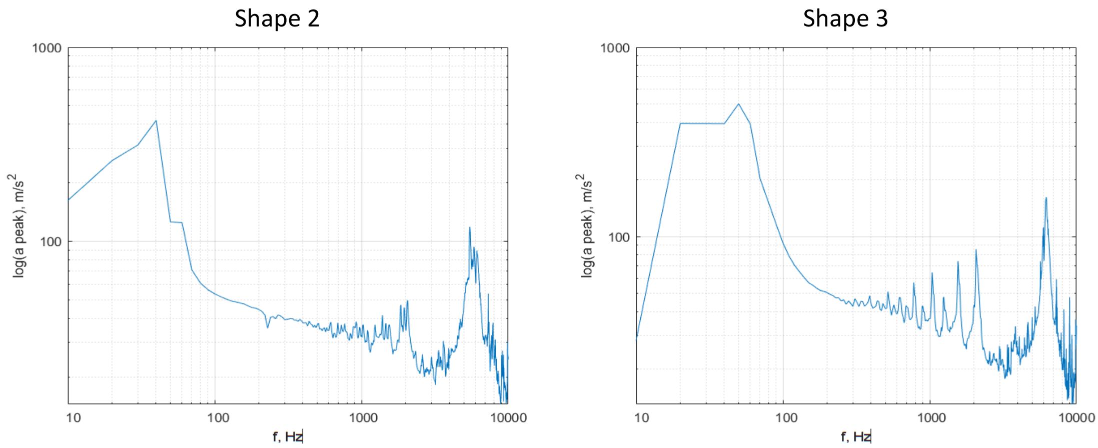

Figure 10. Comparison of shaker characteristics for springs with the best parameters.

4.2. Tests of selected types of suspension under load

Suspension types using springs No. 2 and No. 3 were tested at a weight of 0.8 kg attached to the shaker table. During the tests, the shaker was driven by a harmonic signal of different amplitudes. The values of the qualitative parameters of the shaker are presented in Tab. 4. The values of the THD coefficient decrease with increasing voltage. That is, during tests with load, as the voltage of the input signal increases, distortion decreases. It can also be seen a slight influence of the voltage values of the input signal on the designated parameters.

Table 4. Results of additional tests with an 800-gram weight.

Shapes		Voltage	$\overline{THD}$, dB	σ (THD)	$\overline{a\_peak}$, m/s^2	σ (a_peak)	$\bar{F}$, N	σ (F)	$\overline{I\_RMS}$, A
Shape 2		1 V	-20.15	10.58	11.17	8.36	11.24	8.41	0.189
		10 V	-22.38	11.08	12.167	8.22	12.24	8.27	0.201
Shape 3		1 V	-20.97	10.9	11.62	8.68	11.69	8.73	0.189
		5 V	-22.07	11.18	12.08	8.57	12.15	8.62	0.187
		10 V	-22.77	11.44	12.6	8.44	12.68	8.49	0.186

4. Conclusions

Based on the analysis of the test results, it can be concluded that the influence of the suspension shape on the dynamic characteristics of the modal shaker is significant. The shape of the spring is largely responsible for the obtained characteristics and for the emerging non-linearities. The use of an appropriately shaped springs assure linear characteristics and allows to eliminate locally occurring resonances, which have a negative impact on the characteristics of the modal shaker. The shape of the spring is also reflected in the accuracy of the input signal reproduction by the shaker armature. It is confirmed by the average mean values of the THD coefficient, which allow to determine the harmonic deformation. High level of THD (Total Harmonic Distortion) can be the result of several factors occurring simultaneously, such as: composite material non-linearity and stress non-uniformity, improper spring attachment, amplifier distortion, unstable power source or power line interference. Investigating the source of the high THD and its reduction will be the subject of further research. Studies of the fatigue properties of the developed suspensions are also planned to assess the suspensions in terms of wear rates. The value of the average value of peak accelerations, and thus the value of the force generated, also depends on the shape of the suspension. With an inadequate suspension shape, the value of the generated forces may be much lower than the design of the modal shaker would allow.

Summarizing the results of the conducted research, it can be concluded that the optimal shape of the spring is proposal No. 3. The experimental research carried out confirmed the results of FEM analyses, where the shape of spring No. 3 obtained the lowest value of maximum stresses among the proposed shapes.

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Additional information

The authors declare no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

References

1. G. F. Lang, D. Snyder; Understanding the physics of electrodynamic shaker performance; Sound and Vibration, 2001, 35, 24–33
2. Sentek Dynamics; An introduction to Sentek Dynamics vibration testing systems; https://www.sentekdynamics.com/introduction-to-sentek-dynamics-vibration-testing-systems (accessed on 2023.12.06)
3. D. Augustyn; Wyznaczanie wybranych charakterystyk wzbudnika modalnego; Master's Thesis, Silesian University of Technology, Gliwice, 2021
4. D. Augustyn; Optymalizacja konstrukcji wzbudnika elektrodynamicznego; Transitional Work, Silesian University of Technology,Gliwice, 2021
5. G. D'Emilia, A. Gaspari, F. Mazzoleni, E. Natale, A. Schiavi; Calibration of tri-axial MEMS accelerometers in the low-frequency range - Part 1: Comparison among methods; Journal of Sensors and Sensor Systems, 2018, 7, 245–257; DOI: 10.5194/jsss-7-245-2018
6. M. Česnik, J. Slavič; Vibrational Fatigue and Structural Dynamics for Harmonic and Random Loads; Strojniški vestnik – Journal of Mechanical Engineering, 2015, 60, 339–348; DOI: 10.5545/sv-jme.2014.1831
7. G. R. Henderson, A. G. Piersol; Fatigue damage related descriptor for random vibration test environments; Sound and Vibration, 1995, 29, 20–2.
8. Dewesoft documentation; https://www.manualslib.com/manual/2054662/Dewesoft-Ds-Pm-20.html (accessed on 2023.12.06)
9. Material documentation; G10/FR-4 Glass Epoxy; https://www.curbellplastics.com/Research-Solutions/Materials/G10-FR-4-Glass-Epoxy (accessed on 2023.12.06)
10. P. Blaschke, S. Schneider, R. Kamenzky, D. J. Alarcón; Non-linearity identification of composite materials by scalable impact modal testing; In: Conference Proceedings of the Society for Experimental Mechanics Series, 2017, 5B, 7-14; DOI: 10.1007/978-3-319-54987-3_2
11. P. Czop, D. Slawik, D, Validation of Fatigue Model of a Hydraulic Shock Absorber Equipped with Shim Stack Valves. In Journal of Physics: Conference Series, 2022, 2184; DOI: 10.1088/1742-6596/2184/1/012057