ISSN 1330-3651 (Print), ISSN 1848-6339 (Online)

https://doi.org/10.17559/TV-20230129000286

Original scientific paper

Modal Analysis of Damping System of Optical Table Levitating on Magnetic Vibration Isolating Supports

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Abstract: The paper experimentally investigates the modal parameters and damping properties of a damping system of a honeycomb optical table levitating on four magnetic vibration isolating supports. It is shown that self-centring vibration isolating supports based on a permanent magnet system can be used to suppress oscillations of the optical table. After the modal analysis, it is obtained that natural system frequencies are 6.03, 301.8, 438 and 558 Hz.

Keywords: damping; magnetic levitation; modal analysis; optical table; resonant frequency; vibration isolating support

1 INTRODUCTION

Vibration isolators are the most important components used to protect precise technological equipment from external undesirable vibrations in the fields of the micro-manufacturing, lithography, high-resolution metrology, optical, physical, and other high-precision experiments [1]. In practice, these experiments are performed on rigid and isolated bases to eliminate all relative displacements of the arranged equipment. Commonly honeycomb optical tables [2] are widely used because of their excellent rigidity. Modern honeycomb optical tables are made as layered composite structures consisting of the upper cold-rolled steel layer with threaded mounting holes used to attach experimental equipment, the lower steel layer with threaded mounting holes to attach supporting vibration isolators, and the intermediate layer of honeycomb core [3, 4]. Such cellular structures have good vibration damping characteristics, but the main advantage is that they are much lighter than the massive stone tables, e.g., made of granite. The achievement of vibration activity minimization mostly depends on the system ability to work at the resonant (natural) frequencies of the system. Honeycomb optical table vibration frequencies usually lay within the interval from 2 to 7 Hz as the natural frequency of the optical tables [5, 6]. The dynamic properties of the supports also differ and depend on the vibration characteristics and desired damping effect. There are passive and active methods to reduce oscillations, both having advantages and disadvantages [7]. Passive systems are easy to manufacture and set up. They do not require additional energy and have simple structure, but also have a limited performance [8]. Passive systems were adequate until a decade ago, but are now being seriously challenged by more refined requirements in precision for manufacturing and quality control processes. Traditional linear passive isolators are capable to suppress vibrations at frequencies higher than $\sqrt{(2k/m)}$ (k is the stiffness and m is the mass), which are, however, ineffective under low-frequency excitation, since the stiffness cannot be too small to ensure the static load bearing capacity [9]. Active systems are mostly used to isolate vibrations starting from 0.7 Hz frequency. As they run on electricity, they can be negatively influenced by problems of electronic dysfunction and power modulation noise. Very idiosyncratic properties were observed in honeycomb optical tables and other technological equipment, which was supported by quasi-zero (negative) stiffness isolators with the range of resonant frequencies starting from 0.5 Hz [10]. Among other excellent properties of quasi-zero stiffness vibration isolators, they exhibit non-linearity in the system behaviour when stiffness depends on carried load and amplitude of oscillations [10-12]. Many researchers have tried to exploit non-linear vibration isolators that meet the requirements of vibration isolation with low stiffness and high load capacity [13]. For example, the dual-chamber pneumatic vibration isolator is a type of typical non-linear vibration isolator with low cut-off frequency, which can efficiently suppress the external oscillations transmitted to the equipment [13]. But the main disadvantages of the pneumatic vibration isolators are their high mass, price, and the need for some accessories, especially the requirement of the compressed air. This makes them difficult to use in, for example, a vacuum environment. In recent years, the use of rare earth permanent magnets, such as the Nd-Fe-B permanent magnets, which are sintered from neodymium-iron-boron has been greatly developed [13]. The permanent magnet can reach a strong coercive force, hysteresis energy, and the high residual magnetic induction [13, 14]. The characteristics of the permanent magnets were applied in mechanical fields such as magnetic bearings, eddy current brakes, and other magnetic levitation devices [13, 14]. Both the magnetic force and stiffness present a non-linear relationship with a varying gap between the couples of the permanent magnets, since they are working without mechanical friction and energy supply. They exhibit high bandwidth, no contact, and vacuum suitability [15]. All these characteristics of the permanent magnets lead to explore a new kind of magnetic levitation vibration isolation supports. This kind of non-linearity comes up to high-static and low-dynamic stiffness, which attracts the attention of the researchers in recent years [13]. Previous work [11] analysed the load capacity and near zero stiffness zone of the original design passive magnetic support with non-linear stiffness [10-12] that is stable in the vertical and horizontal direction and is intended to attenuate vertical low frequency vibrations. The proposed support is assembled from permanent magnets and additively manufactured components, making use of standard 3D printing techniques based on fused deposition modelling. The distinctive features of the isolators are their ease of manufacture using standard 3D printers, sustainable materials, and low-cost magnets provided by online suppliers available throughout the world. The present paper continues the experimental research with dynamic characteristics. To achieve this aim, a modal analysis of the system consisting of the honeycomb optical table and proposed magnetic vibration isolating supports was performed to allow the establishment of resonant frequencies at which damage or degradation in performance was likely. Modal analysis is a process in which the structure is characterized by natural (resonance) frequency, damping, and modal shapes, i.e. dynamic properties [16, 17]. Since all the bodies are plastic and have a mass, they can vibrate. This is because most engineering equipment is exposed to oscillatory movements. The determination and estimation of resonant frequencies is the most suitable way to understand the problems of any structural vibration. The commonly used method is to determine the modal parameters of the structure. The results of the modal analysis can be used for creating of a mathematical model of the structure dynamics to get calculations results instantly investigating the system behaviour with different physical designs without wasting resources for manufacturing new prototypes. The mathematical model consists of modular forms, each with its eigenfrequency and modal damping. Modal parameters provide a final description of the design dynamics. Experimental modal analysis is based on the determination of modal properties, where data of the object response to excitation is analysed. The modal parameters of analytical modal analysis are computed from finite element models. Two types of experimental modal analysis are distinguished: classical modal analysis and modal analysis in the operational mode. During the classical modal analysis, the frequency response functions (soft punch response functions) are calculated from the measured excitation forces and the excitation response of the analysed system [18]. Operational Modal Analysis (OMA) or Ambient Modal Identification is widely used when the analysed system is in real operating conditions and in situations when it is difficult (or impossible) to control the artificial excitement. During the OMA, excitation forces and ambient noise are considered as an invisible incoming signal, and the output signal gives dynamic properties of the analysed system in real conditions. The OMA methods are classified by two aspects - frequency domain or time domain and Bayesian or non-Bayesian. Bayesian or non-Bayesian methods are based on the statistical estimation with known theoretical properties such as correlation function or spectral density of vibrations. Common non-Bayesian methods additionally include time-dependent stochastic subspace identification (SSI) [19] and frequency dependent - frequency domain decomposition (FDD) [20-22]. This experiment uses the methodology based on experimental data of modal vibration testing. Results are presented by singular values spectral bell in the timeline, normalized sloping curves of vibroactivity and correlation line of extreme values.

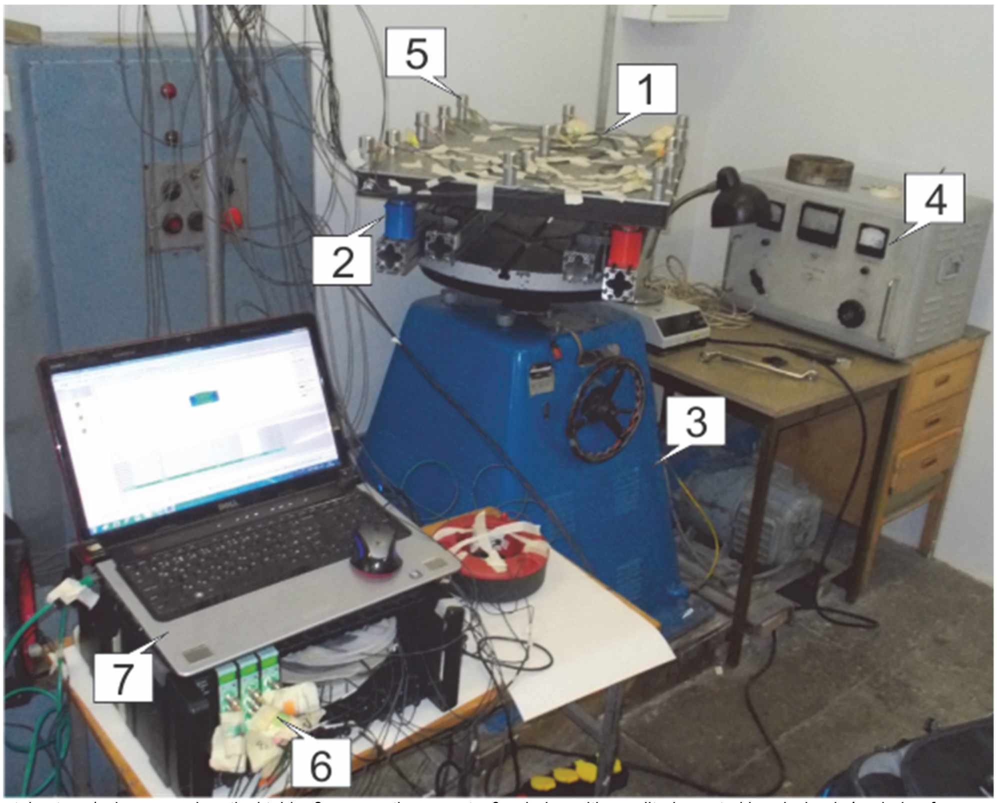


Figure 1 Experimental setup: 1 - honeycomb optical table, 2 - magnetic supports, 3 - shaker with amplitude control hand wheel, 4 - shaker frequency control panel, 5 - piezoelectric accelerometers, 6 - dynamic input data acquisition modules, 7 - computer with installed software for modal measurement and analysis

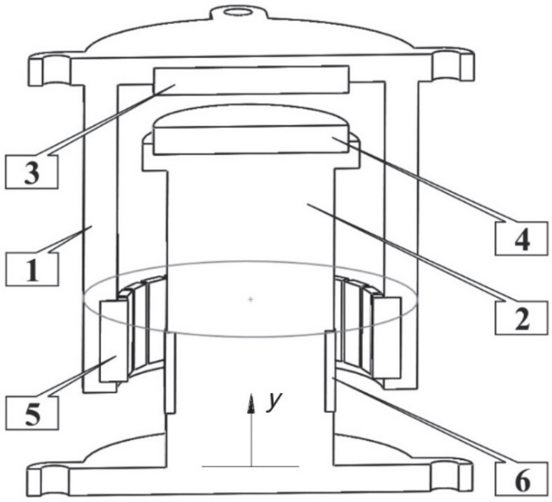
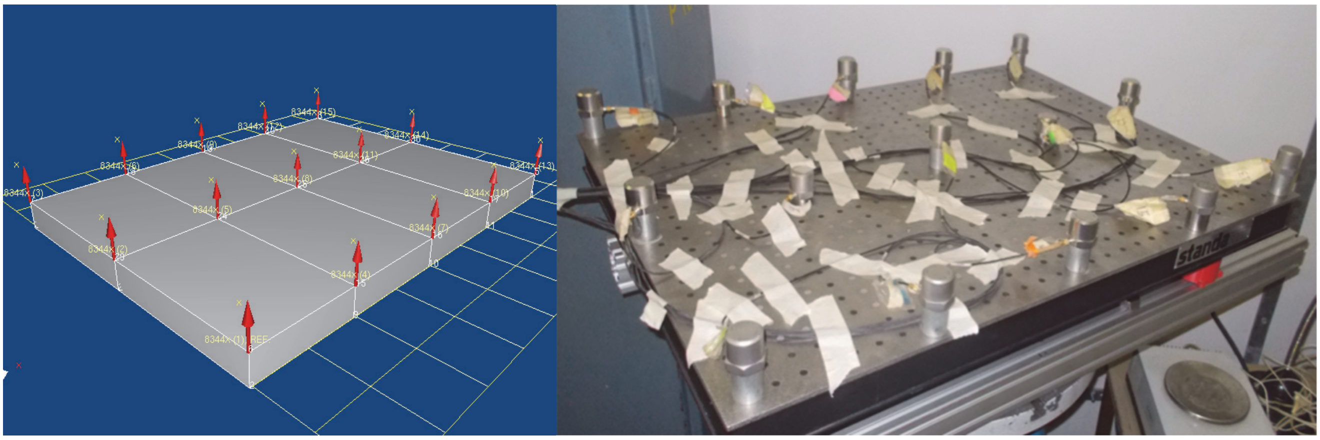


Figure 2 Scheme of the magnetic vibration isolator: 1 - cap, 2 - base, 3, 4 - disc-shaped permanent magnets, 5, 6 - prismatic permanent magnets, *y* - coordinate axis



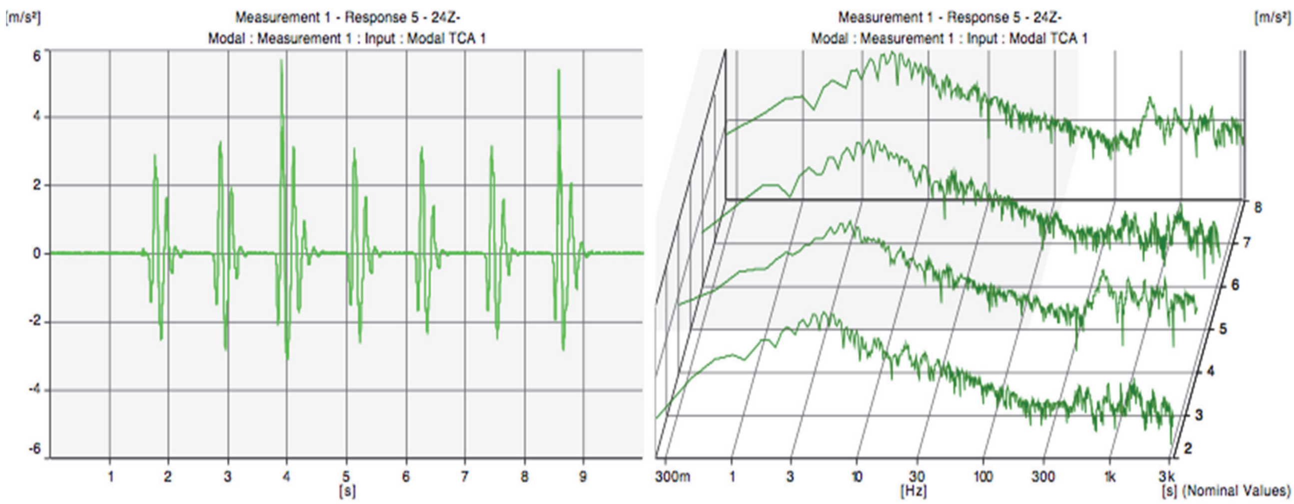
a) b)

Figure 3 Vibration measurement points arrangement scheme (a, shown by red arrows) and actual location (b) of piezoelectric accelerometers on optical table during modal analysis

2 OBJECT OF RESEARCH

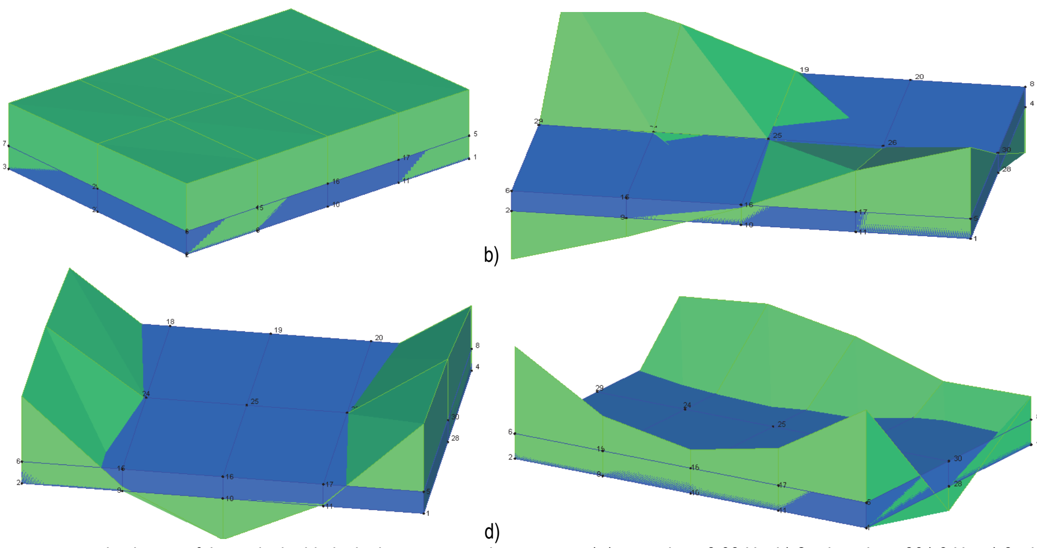
The experimental setup is presented in Fig. 1. The honeycomb optical table 1 is levitated on four magnetic isolators 2, which can be considered as a single-degree-of-freedom magnetic spring damping system. Shaker 3 was used to provide vertical vibrational excitation. Scheme of magnetic vibration isolator is presented in Fig. 2. This self-centring magnetic support consists of two plastic parts manufactured using 3D printing technology: cap 1 and base 2. It utilizes two opposite disc-shaped permanent magnets 3 and 4 and prismatic permanent magnets 5 and 6, which are symmetrically arranged around the circles and are used to centre the cap and to ensure its radial stability. The base 2 with permanent magnet 4 is stationary, while the cap 1 with permanent magnet 3 has a reciprocating motion in direction of the y axis (Fig. 2). The experimentally obtained repulsion force versus cap displacement in the direction of the y axis (Fig. 2) curve, as well as the results of the numeral simulation of the magnetic flux density inside the magnetic support, are presented in previous research [11].

Brüel&Kjær™ mobile frame type 3660-D with LAN-XI type data acquisition modules 6 (Fig. 1) connected with 15 piezoelectric accelerometers of 8344 type, the location of which in the optical table is presented in Fig. 3, and Dell computer 7 (Fig. 1) with PULSE data analysis software were used for the modal analysis. Type 8344 are calibration grade piezoelectric accelerometers designed and optimized for low frequency and low-level measurements. They feature low-noise, built-in constant current line drive preamplifiers and are based on the Brüel&Kjær™ patented DeltaShear design. Their voltage sensitivity is 250 mV/ms^{-2}, frequency range 0.2 - 3000 Hz, mounted resonance frequency > 10 kHz, transverse resonance frequency is 3.5 kHz, residual noise ≤ 40 μV, temperature coefficient of sensitivity is 0.05%/°C, measuring axes - perpendicular to the mounting surface, weight 176 g, magnetic sensitivity is 0.5 ms^{-2}/T, operating temperature range from –50 to 100 °C. The piezoelectric accelerometers were attached to the table with M5 screws, the mounting torque was controlled and did not exceed the recommended 3.5 Nm torque.



a) b)

Figure 4 Vibration acceleration time-domain signal recorded by accelerometer (a) and its modal spectrum (b)



a) b) c) d)

Figure 5 First four resonant mode shapes of the optical table levitating on magnetic supports: a) 1-st mode at 6.03 Hz, b) 2-nd mode at 301.8 Hz, c) 3-rd mode at 438 Hz, d) 4-th mode at 558 Hz

3 RESULTS AND DISCUSSION

The typical excitation response vibration acceleration time-domain signal recorded by one of the fifteen accelerometers (Fig. 1, Fig. 3) used is presented in Fig. 4a. Modal spectra were generated after excitation using software. A typical modal vibration acceleration spectrum obtained in frequency and time coordinates is shown in Fig. 4b. After the excitation, the first four resonant mode shapes were calculated and visualized. The vibration mode shapes of the optical table are shown in green colour in Fig. 5. The first mode shape (Fig. 5a) represents the movement of the optical table in the vertical direction, i.e. it represents the movement of the magnetic supports. The rest of the mode shapes shown in Fig. 5b to Fig. 5d, represent bending and twisting deformations of the optical table. Using the singular value spectral bell function, an inverse Fourier transform was performed for the determination of damping and natural frequencies. The obtained normalized correlation functions are shown in Fig. 6a, and Fig. 6b (top curves). The top curves presented in Fig. 6a and Fig. 6b exhibit a typical response of a resonating system that decays exponentially. The scattered region indicates the part of the correlation function that was used for the estimation algorithm.The first two modes (Fig. 5a, and Fig. 5b) are presented in Fig. 6 as an example. The normalized correlation functions for specific modes (Fig. 6a, and Fig. 6b, top curves) were obtained from the data recorded from all 15 measurement points (Fig. 3). The damping ratio was obtained using the expression of a logarithmic decrement and the correlation function calculated from the maximum values of the sloping sine curve. Estimation is performed with a linear regression technique (red part of the bottom curves presented in Fig. 6a, and Fig. 6b). The results obtained after experimental modal analysis are summarized in Tab. 1. It is evident that sloping forms of the system behave differently at all four modes during excitation. A damping ratio of the 1-st mode (Tab. 1) shows that the amplitude decays by about 17% over one period of oscillation.

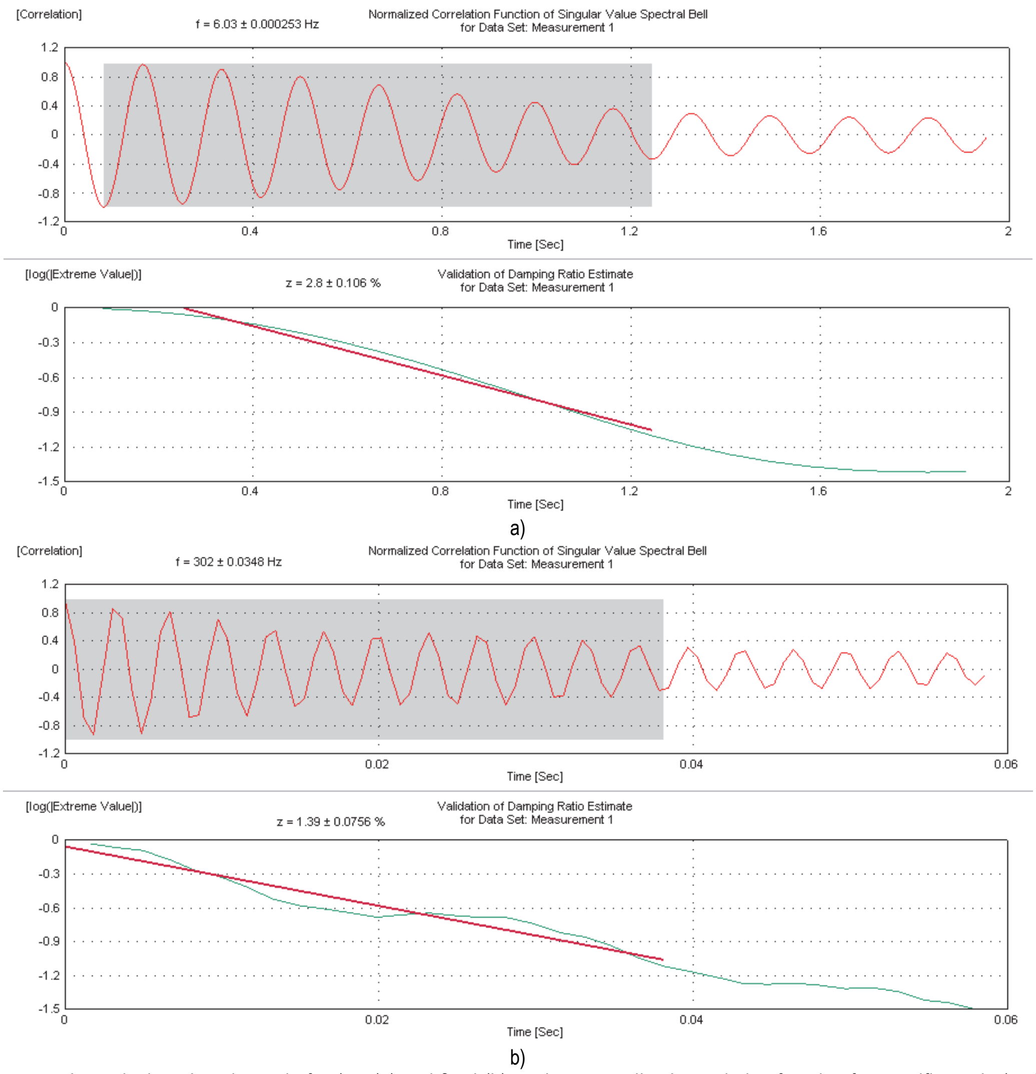


Figure 6 Curves used to calculate damping ratio for 1-st (a) and 2-nd (b) modes: normalized correlation function for specific mode (top) and extreme correlation function values used to calculate damping ratio (bottom)

Table 1 Results of experimental modal analysis

Mode Nr.	Resonant frequency / Hz	Damping ratio / %
1	6.03	2.8
2	301.8	1.389
3	438	1.271
4	558	0.2027

It should be mentioned that the first mode shape represents the table movement in the direction of the axis of magnetic supports (Fig. 5a). This mode should be avoided when precision experiments with optics are performed. Other three modes represent the movement shape of optical table at higher resonance frequencies (Fig. 5b to Fig. 5d, Tab. 1). Some electric devices and power supplies can generate frequencies that are close to frequencies of higher modes thus becoming a critical component for the dynamic properties of the optical table.

4 CONCLUSIONS

The results of the experimental modal analysis represent the dynamical parameters of the damping system: natural frequencies, modal damping ratio, and modal shapes. Using the modal parameters, it is possible to evaluate, analyse and understand the influence of resonant frequencies (shapes) on the investigated damping system of the honeycomb optical table levitating on magnetic supports. In addition, the results of the modal analysis can be used to solve problems related to a particular mode and as basic calculating parameters for analytical investigations. The first four modes were obtained during a modal analysis. The 1-st mode at 6.03 Hz represents the resonant frequency of magnetic supports. The 2 - 4-th modes represent resonant frequencies and modal shapes of the optical table as a deformed object. Under operating conditions, those modal parameters should be assessed and avoided due to the resonance effects that can occur at particular frequencies. Natural frequency analysis proved the ability of the proposed supports to realize the low-frequency vibration isolation. The supports can be applied in the field of precision measurement, advanced manufacturing, and physical, optical, and chemical experimental instruments to suppress outer environmental oscillations. The performance of a magnetic vibration isolator can be further improved by using higher quality magnets. Subsequently, a simulation of the vibrational response should be carried out to determine the behaviour of the isolator over a wider range of frequencies.

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