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Research Paper

Laboratory Tests and Numerical Simulations of Two Anti-Vibration Structures Made by 3D Printing – Comparative Research

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This article presents a comparison of test results from two models of anti-vibration systems (I and II) made employing MJF 3D printing technology and two different materials. The research included laboratory tests and numerical simulations, assuming a linear nature of the mechanical properties for the materials and models of structures. The aim of this research was to assess the consistency between laboratory test and numerical simulation results. In addition, evaluation of the suitability of using MJF technology to produce anti-vibration systems was conducted. During the laboratory tests, the response of the two models of structures to vibrations generated by an exciter was recorded using a high-speed camera. Subsequent image analysis was performed using the MOVIAS Neo software. The obtained values of vibration displacements and resonant frequencies were used to validate the numerical model created in the Simcenter Femap software. Relative differences between the values of resonant frequencies obtained experimentally and through simulations were determined. In the case of the structural model I, creating its numerical model without considering the nonlinearity of mechanical parameters was found to be unjustified. The comparison of the displacements determined during numerical simulations showed relative differences of less than 16% for both models in relation to the laboratory test results. This comparison result indicates a satisfactory accuracy in simulating this parameter. An assessment of the quality and accuracy of MJF technology-produced prints, led to the conclusion that due to the formation of internal stresses during the print creation, the use of "soft" materials in this technology is problematic.

Keywords: finite element method; 3D printing; mechanical vibrations.

1. Introduction

In numerous research centers around the world, developmental research on new materials and systems designed to limit mechanical vibrations generated by machinery and devices is currently underway. Vibrations constitute a potentially harmful factor both for mechanical devices, causing their faster damage or failure, as well as for people, by reducing work comfort or leading to adverse health effects.

An innovative approach to advancing the concept of anti-vibration systems involves the use of 3D printing technology as a quick, precise and easily accessible way to create structures with consistent mechanical properties. The advancement of various 3D printing technologies opens up the possibility of using a wide range of materials with different physical properties that can be selected based on the expected properties of the prints. For instance, some studies explored the possibility of using a material with shape memory (shape memory polymer, SMP) in 3D printing (Yang *et al.*, 2016). Many studies also attempted to use 3D printing to test new solutions in the field of body protection (Park, Lee, 2019), metamaterials (Zolfagharian *et al.*, 2022) or shock absorbers (Sathyapriya *et al.*, 2022). Confirming the usefulness of 3D printing for the production of anti-vibration systems could reduce both the time and cost of developmental work in this field.

The rising popularity of developing numerical methods together with the increasing computing capabilities of computer systems creates more and more opportunities for testing anti-vibration systems by means of computational simulations. Such an approach is a significantly faster and lower in cost than carrying out traditional laboratory tests of subsequent physical models subjected to modifications. The potential of using the finite element method (FEM) to study anti-vibration systems is presented in many scientific papers (Burlayenko *et al.*, 2019; Sari *et al.*, 2022; David Müzel *et al.*, 2020; Kamel *et al.*, 2019; Shi *et al.*, 1997). Various researches employ the FEM in studies related to limiting the effects of mechanical vibrations, predicting their impact on the environment, as well as analyzing the properties of 3D prints (Abbot *et al.*, 2019; Jindal *et al.*, 2020; Żur *et al.*, 2019). Verifying compliance between numerical models created using the FEM and laboratory results obtained for real models could accelerate the development of new solutions aimed to reduce mechanical vibrations.

2. Anti-vibration systems – the object of research

In order to assess the possibility of using FEM simulations for the to design of 3D anti-vibration structures, comparative tests of two anti-vibration structures produced using the multi jet fusion (MJF) method involving the thermal, selective sintering of powders were carried out. The MJF technology was chosen because of its capacity to employ large printing areas, enabling the printing of over 30 elements in one printing process while maintaining a short printing time. Two materials were used for the prints: nylon PA12 and Ultrasint TPU01, and their parameters are presented in Table 1.

Table 1. Basic mechanical parameters of materials used for 3D printing.

Material	PA 12	TPU01
Young's modulus [MPa]	1700	56
Density [kg/m^3]	907.2	1206.6
Tensile strength [MPa]	48	8

Comparative studies were carried out on structurally distinct models of structures: I and II, as shown in Fig. 1.

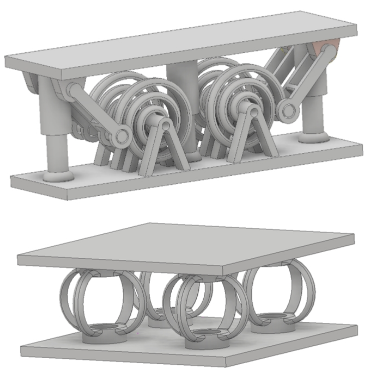

Fig. 1. Developed models of anti-vibration systems with external dimensions: model I – $24 \times 77 \times 29.6$ mm (top); model II – $60 \times 60 \times 25.3$ mm (bottom).

3. Research method

3.1. Laboratory tests

The results of numerical simulations were verified through comparison with measurement data obtained during the tests of real anti-vibration structures subjected to mechanical vibrations. This validation was based on the registered vibration displacements, e.g., the values of the resonance frequencies of both models and the values of displacements of the upper planes of the structures during resonance. The models of anti-vibration systems were tested on the test stand apparatus shown in Fig. 2. The models were excited to oscillate with a tunable sinusoidal signal with a frequency range of 5–100 Hz. During the tests, the models were loaded with masses whose values were selected experimentally. Displacements of elements in the system models were recorded using a high-speed camera. Subsequent image analysis was performed using the MOVIAS Neo software.

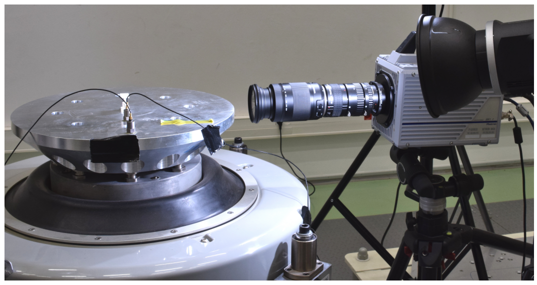

Fig. 2. Laboratory measurement stand consisting of a mechanical vibration exciter and a high-speed camera.

In this way, vibration displacement profiles over time at specific points across the tested system models were obtained. An example of the markers' arrangement at measurement points on the tested models is shown in Fig. 3.

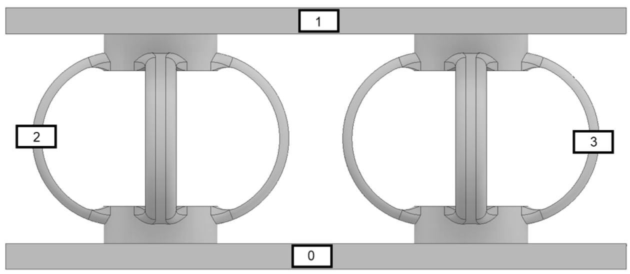

Fig. 3. Arrangement of measurement markers on model II.

The following loading masses were used: 9, 18, and 27 g for structure I, and 60, 120, and 180 g for structure II. The selected loading masses enabled the observation of system resonances without causing any damage to the elements of the tested models. For both models and each applied load, resonant frequencies and their corresponding displacement values were deter-

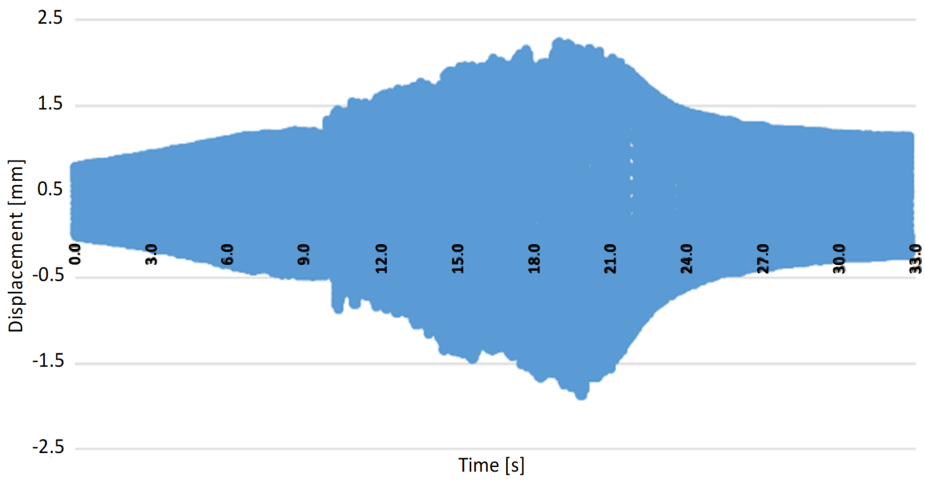

Fig. 4. Diagram of the displacement profile of measurement point 1 for model II loaded with a 60 g mass along the y-direction (vertical to the exciter table).

mined. For further analyses, the displacement values of the measuring point 1 along the y-axis were used. A representative graph of the displacement variation of point 1 over time is shown in Fig. 4.

The data obtained in this process, including resonance frequencies and displacements of individual points in the system models, were used to validate the numerical simulations.

3.2. Numerical simulations

The numerical simulations were performed using the Siemens Simcenter Femap software, a platform designed to conduct engineering analyzes using the FEM. The system models presented in Fig. 1 were subjected to discretization, leading to the generation of a finite number of elements. Next, meshes were generated from the obtained elements for both models. They are presented in Fig. 5.

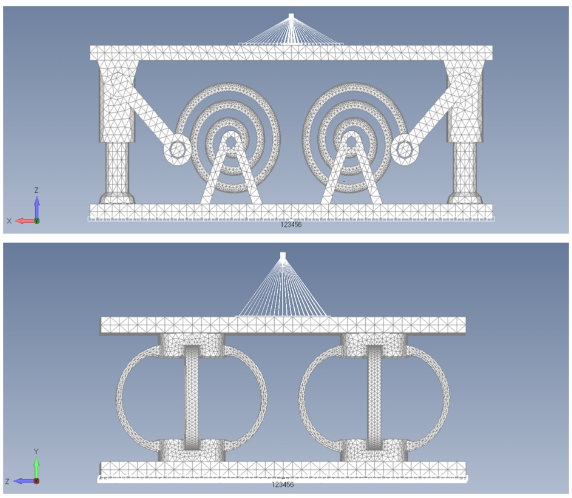

Fig. 5. Grids of the anti-vibration structures generated after the discretization of the continuous 3D models.

In the case of the system model I, a structural element-the central guide, was removed. This removal did not affect the simulation results and increased the number of mesh elements, which ultimately influenced the speed of calculations.

The main assumption of the simulations was the premise of linear characteristics of the material used. The loading of the structures was carried out with the use of RBE2 elements, connecting the upper surface of the models with a designated point that was assigned a mass condition. The same elements were used in the bases of the models to constrain their mobility, by effectively connecting them to the vibration exciter table during laboratory tests. An additional boundary condition used in the frequency analysis involved considering acceleration corresponding to the acceleration of gravity.

In order to determine the resonance frequencies of the system models and their displacements, two numerical analyzes were carried out: modal and frequency.

Modal analysis yields a set of vibration modes of the tested system model together with resonant frequencies. The shape of the modes makes it possible to evaluate the behavior of the model during vibrations of a specific frequency.

On the other hand, frequency analysis makes it possible to determine the actual displacements of individual nodes of the mesh of elements in addition to showing the prevalent stresses and forces. In order to determine the appropriate damping coefficient during the simulation for a given model structure, a number of test simulations were carried out. They involved selecting the damping value for the lowest applied load in such a way that the displacements obtained for the simulation closely matched the values obtained during the experimental tests.

Then, the selected damping coefficient value was used in simulations featuring a different load.

4. Comparison results

The values of resonant frequencies obtained during the laboratory tests and through numerical simulations are presented in Table 2. Comparison of displacements obtained using a high-speed camera and numerical simulations is shown in Table 3. Symbols used in the tables:

- F_{lab} – resonant frequency obtained from laboratory tests [Hz];
- F_{sym} – resonant frequency obtained from numerical simulations [Hz];
- ΔF – relative difference between resonant frequencies from laboratory tests and numerical simulations [%];
- D_{lab} – displacement of measuring point 1 obtained from laboratory tests [mm];
- D_{sym} – displacement of the node coincident with the location of measurement point 1 obtained from numerical simulations [mm];
- ΔD – relative difference between the displacements obtained in the laboratory and from numerical simulations [%].

Table 2. List of resonance frequencies obtained on a laboratory stand and with the use of numerical simulations along with the relative difference of the obtained values.

	Model I			Model II		
Load	9 g	18 g	27 g	60 g	120 g	180 g
F_{lab} [Hz]	25.6	20.8	18.9	22.2	16.4	11.4
F_{sym} [Hz]	45.3	36.1	30	23.4	17.2	14.3
ΔF	77.0%	73.6%	58.7%	5.4%	4.9%	25.4%

Table 3. Summary of the maximum displacements obtained for measuring point 1 using a laboratory stand and numerical simulations along with the relative difference of the obtained values.

	Model I			Model II		
load	9 g	18 g	27 g	60 g	120 g	180 g
D_{lab} [mm]	5.4	7.2	9.9	4.0	8.5	11.7
D_{sym} [mm]	5.3	8.35	11.3	4.02	7.25	10.5
ΔD	2.1%	16.0%	13.7%	0.2%	15.0%	10.3%

By analyzing the obtained relative differences of resonance frequencies ΔF for system I, one can notice large differences between the results of laboratory tests and the ones from simulations. The differences exceed 58% for all loads used. In the case of structure II, the differences for loads of 60 g and 120 g do not exceed 5.5%, while for the highest load of 180 g the difference increases significantly and exceeds 25%. However, in the case of system II, despite a large relative difference expressed in percentage, the difference between the resonance frequency values is only 3 Hz.

For both models of anti-vibration systems, the relative differences in the obtained displacements did not exceed 16%, which can be considered a satisfactory result. The obtained relative differences correspond to the real displacement differences of about 1.4 mm.

5. Conclusions

The presented research showed the potential of using numerical methods to predict resonance frequencies and displacements of 3D-printed anti-vibration structures made. The tests of the two presented models showed that obtaining reliable results through numerical simulations strongly depends on the construction of the anti-vibration system model.

The uncomplicated construction of models, such as tested model II, in which the vibration energy is dissipated only in the structure of the material, allows the use of a linear model to a limited extent. However, this approach is associated with low accuracy of the obtained simulation results. This is confirmed by the obtained increased differences in the determined values of the resonance frequency (laboratory and FEM simulation) for model II under a load of 180 g, at which the model of the structure probably behaves non-linearly.

In the case of models with more complex structures, such as model I, the correct determination of resonant frequencies may require taking into account not only the material property non-linearities, but also non-linearities resulting from the structure of the system.

The relative differences in displacements obtained experimentally and through simulation can be considered acceptable. However, due to large differences in the obtained resonant frequencies, the reliability of displacement values determined for model I might be questionable.

The obtained differences in results could be influenced not only by non-linearities in the mechanical parameters of the structures and the materials used, but also by inaccuracies in workmanship and internal stresses generated during the printing of the model structures.

When analyzing the applicability of MJF technology based on powder sintering at high temperatures for producing 3D anti-vibration structures (model made of TPU01 material), an uneven distribution of the generated thermal energy was noticed during the printing process. This indicates that different elements of the same structure may have significantly different temperatures during printing. A high temperature gradient in the structure itself can cause high internal stresses that affect both the accuracy and quality of prints (deformations) and their mechanical properties.

Numerical modeling of models produced in this way may pose significant challenges and lead to unreliable simulation results.

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