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MODELLING STEEL BEAMS FILLED WITH A COMPOSITE MATERIAL

Key words: FEM modelling, modal analysis, model updating, composite beams, damping.

Abstract: Modelling of structural elements using materials with significantly different parameters is a demanding issue, mainly due to the possibilities of commercial computing software. The paper presents a method of modelling the damping properties of a component consisting of a steel beam filled with a composite material. The compatibility of the modelling using the finite element method with the results of experimental studies have been proved. These results provide the first step for innovative machine tool body construction that consist these types of elements.

Modelowanie belek stalowych wypełnionych materiałem kompozytowym

Słowa kluczowe: modelowanie MES, analiza modalna, identyfikacja modelu, belki kompozytowe, tłumienie.

Streszczenie: Modelowanie elementów konstrukcyjnych, wykorzystujących materiały o znacząco różniących się parametrach, jest wymagającym zagadnieniem głównie ze względu na możliwości komercyjnych pakietów obliczeniowych. W artykule przedstawiono autorską metodę modelowania właściwości dyssypacyjnych komponentu składającego się ze stalowej belki, wypełnionej materiałem kompozytowym. Wykazano zgodność wyników modelowania metodą elementów skończonych z rezultatami badań doświadczalnych. Wyniki te stanowią punkt wyjścia do budowy nowatorskiej konstrukcji obrabiarki, której elementy nośne wykonane będą z tego typu elementów.

Introduction

The functional properties of machine tools depend mainly on the properties of the body elements and the drive systems. The vast majority of currently manufactured machine tool bodies are made in the form of grey iron castings [1, 2]. This material is characterized by good casting properties, good machinability, high vibration damping ability (resulting from high internal friction of the material), and relatively low production cost in large scale. The disadvantages of cast bodies are their relatively high mass, high cost for unit production, and significant limitations in terms of the possibility of modifying the existing structure [3–5].

The aforementioned disadvantages do not apply to welded steel bodies that are alternatively used in the construction of machine tools. They have a lower mass (cast iron bodies are about twice as heavy while maintaining the same stiffness), much better strength parameters, and modifying their structure is easier and less expensive. However, steel welded bodies have to be annealed, ensuring the stress stabilization after the welding process Additionally, steel bodies have a low damping capacity, which virtually eliminates them for applications in machine tool bodies [6].

To overcome this drawback, hybrid bodies combining the advantages of cast iron and steel have been developed [7–14]. In [15], the authors designed the slides of high speed milling machines by joining high-modulus, carbon-fibre, epoxy composite sandwiches to welded steel structures using adhesives and bolts, which provided a significant increase in damping properties and a reduction in mass.

In [8], the authors inserted a friction layer between the aluminium-composite interface to increase the structural damping of machine tool structures, such as columns and spindle holders. It has resulted in weight and static deflection reduction and much higher damping capacities than those of aluminium structures.

In presented paper an alternative solution is proposed. It consists in filling the hollow square steel box beam interior with a composite material to improve the damping properties of steel bodies and to reduce mass while ensuring the required stiffness. The material consists of a filling with a different size of the filling fraction and an adhesive resin that fulfils the task of the binder. The proposed structure differs from the abovementioned solutions, both in terms of filling type and its placement in the structure. Therefore, modelling methods cannot be transferred directly. Thus, a new modelling method was developed to predict static and dynamic behaviour of proposed structures.

1. Identification procedure

The research objects were two steel beams with a square cross-section of 70x70 mm, a wall thickness of 3 mm, and a length of 1000 mm, filled with a composite material consisting of a filling with a different size of the filling fraction and a bonding resin fulfilling the task of the binder. The described beams are the basic elements of a hybrid machine tool body presented in Figure 1. The mass percentage of subsequent components of composite filling is presented in Table 1.

The growing popularity of hybrid material bodies forces the need to develop reliable methods for modelling its static and dynamic behaviour. One of the most popular methods for predicting these properties is the finite element method. However, at present, this method

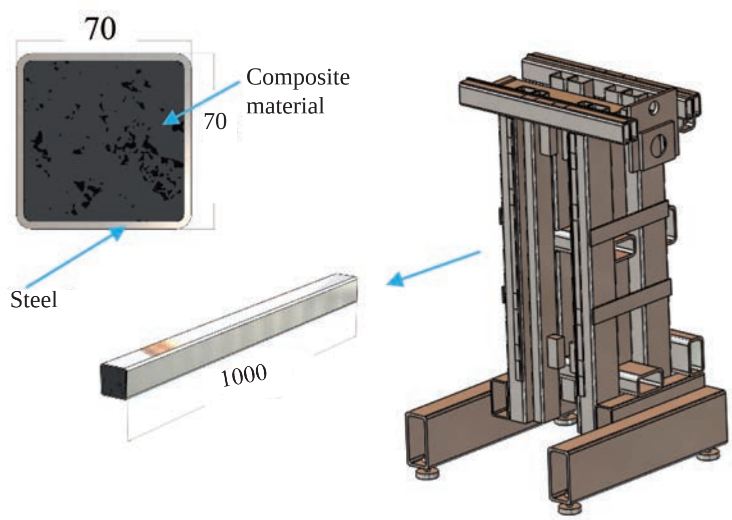

Fig. 1. Research object

does not provide sufficiently accurate models consistent with experiment results, in particular, a representation of the damping properties of the actual structure. In order to improve the FE models, a model update procedure was developed.

The proposed procedure shown in Fig. 2 was introduced in order to determinate the model structure based on the experimental results and then update the uncertainty parameters in the initial finite element model to obtain high model compatibility. Consequently, the proposed procedure is a process of improving a mathematical model of a physical structure based on experimental results.

Table 1. Composition of the composite material

Specimen	Resin	Ash	Small fraction (0,25-2 mm)	Medium fraction (2-10 mm)	Coarse fraction (8-16 mm)	Steel fiber
M05	26.1 %	1.7 %	15.7 %	13 %	30.5 %	13 %
M11	13.2 %	1.3 %	20 %	15.9 %	47.5 %	2.1 %

The algorithm is based on the iterative change of model parameters (within the limits of measurement uncertainty) in order to obtain the best possible compatibility with the real object. It is based on two adjustment criteria: the mode shapes compatibility criterion (expressed in the form of a MAC indicator), and the amplitudes of the FRFs comparison. The test data should be provided as the initial input parameters.

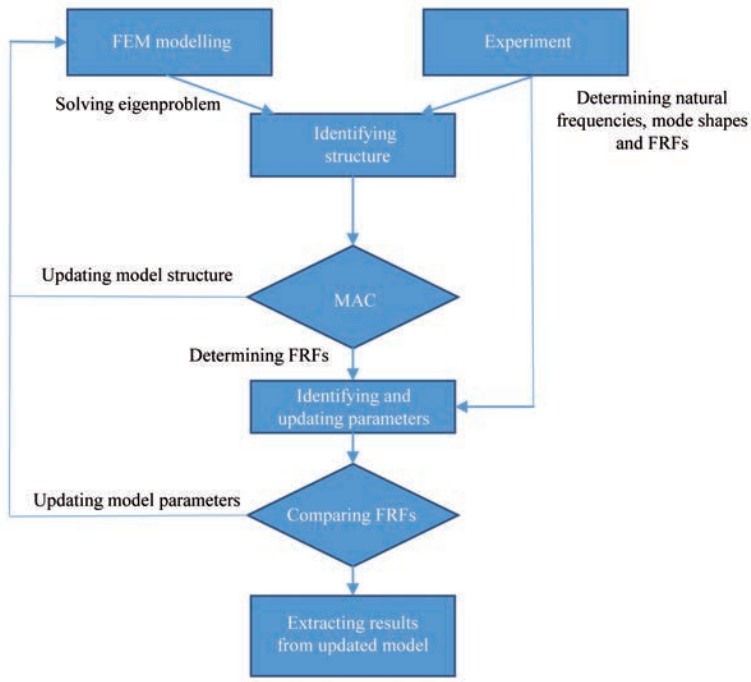

Fig. 2. Model identification procedure

2. Experimental investigations

Despite the fact that research objects have relatively simple geometry, due to composite filling presence, predicting the model structure becomes problematic. There is also lack of literature sources treating about dynamic behaviour of such structures. Therefore, in order to identify model structure, some preliminary tests were conducted. The preliminary tests included checking if the object meets the basic assumptions of modal analysis, i.e. linearity and time invariance, Maxwell's reciprocity theorem, and relatively low damping. After ensuring that research objects fulfil the abovementioned assumptions, it was decided to use the impulse test to determine natural frequencies and mode shapes of the analysed objects.

An important issue for modal data acquisition is the number of measurement points needed to properly identify object mode shapes. An excessive number of measuring points leads to an unnecessarily large data set and a long time needed to carry out the experiment. An insufficient number will result in a poor representation of the structure and may not be adequate to determine mode shapes. To establish an optimal number and arrangement of measure points, a series of tests with different variants were conducted.

After the preliminary tests, the actual experiment was carried out. It consisted in exciting the tested objects, suspended on steel cables, by a modal hammer. Excitation was carried out simultaneously with the verification of signal levels in individual measurement channels and checking values of the coherence function and the spectral density of the excitation signal.

The excitation point did not change. The measurement of the response was carried out at 56 points in the X and Z directions by three-axis ICP accelerometers. The measuring system allowed the observation of the frequency response function, which was determined after each impact and after their averaging. The double impacts or impacts causing overloading on any of the measurement channels were automatically rejected. The layout of the test stand is shown in Fig. 3.

On the basis of determined FFR's, the Polymax algorithm was used to estimate the poles of the modal model. The estimation process was carried out through the interpretation of the stabilization diagram. As a result of the selection of stabilizing poles, a preliminary version of the modal model was build, which was subjected to validation with the use of the AutoMAC indicator before the final interpretation.

Natural frequencies and mode shapes sets (Table 2 and Fig. 4, respectively) for examined beams were obtained as modal tests results. It should be noted that, regardless of the composition of the composite filling, the mode shapes are the same, as is clearly shown in the MAC matrix in Fig. 4. Analysing the mode shapes, it can be stated that the examined objects behave like a classical beam object, despite the presence of a filling.

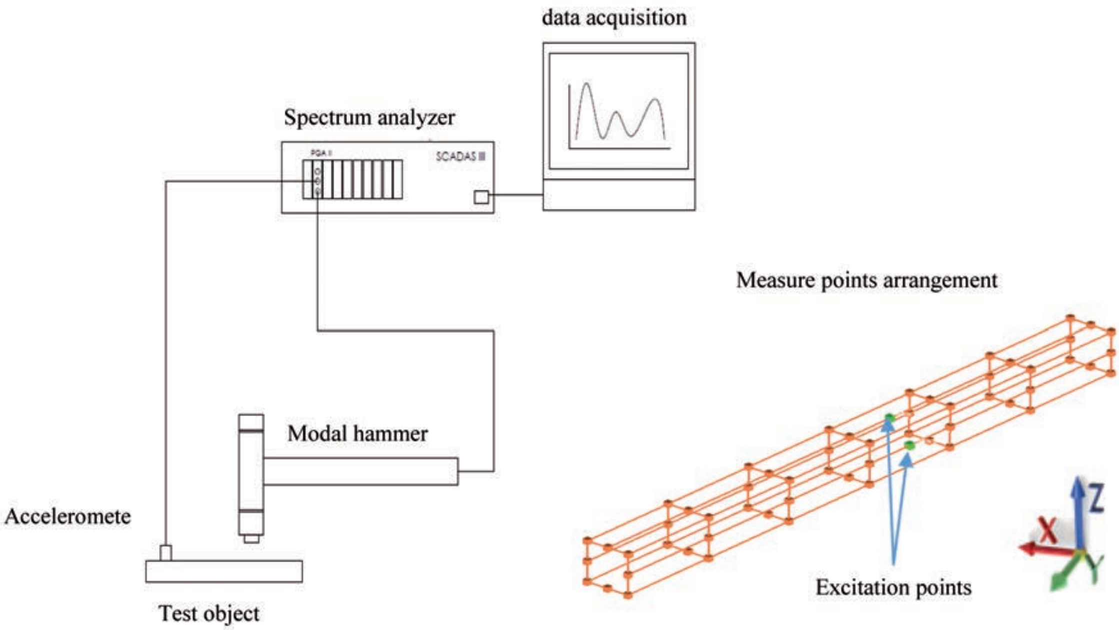

Fig. 3. Test stand

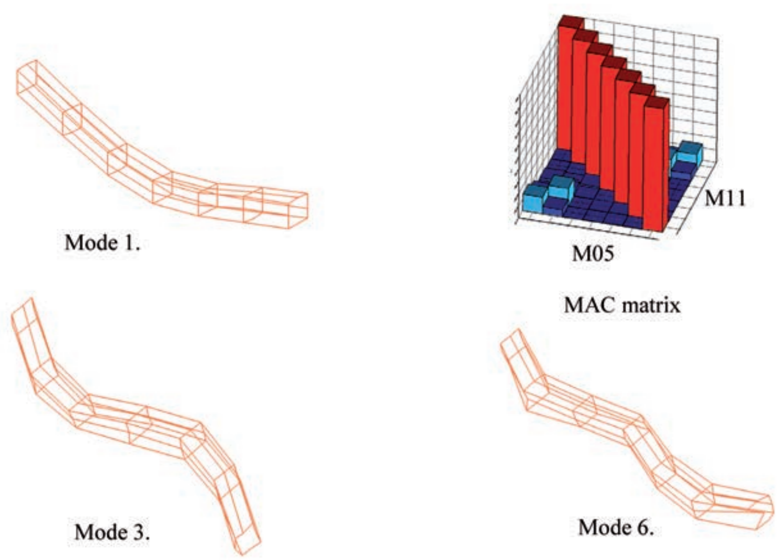

Fig. 4. Mode shapes visualization of 2 types of examined beams

Table 2. Natural frequencies of examined beams

Mode number	M05	M011
1	350.40 Hz	339.17 Hz
2	350.92 Hz	339.25 Hz
3	922.55 Hz	896.65 Hz
4	923.69 Hz	896.77 Hz
5	1290.08 Hz	1263.83 Hz
6	1693.96 Hz	1660.74 Hz
7	1697.04 Hz	1666.14 Hz

Additionally, complementary static experiments were performed to determine Young’s modulus and Poisson ratio required for FEM modeming. The tests were carried out on an Instron 8850 machine. The machine worked in an air-conditioned laboratory with a temperature of 23°C and a relative humidity of 50%. The samples spend 72 hours in the air-conditioned laboratory before testing. Table 3 contains material data, Young's modulus, and Poisson's ratio, which was determined on the basis of static tests, and the damping ratio was determined using frequency response functions (FRFs) and a half power method. This data was used as the model’s initial parameters.

Table 3. Material properties

Parameter	Steel	Composite material M05	Composite material M11
Young's modulus	212 ±5 GPa	16.8 ±0.2 GPa	16.0 ±0.2 GPa
Poisson's ratio	0.28 ±0.03	0.20 ±0.05	0.14 ±0.05
Density	7487 ±35 kg/m³	2118 ±6 kg/m³	2099 ±6 kg/m³
Damping ratio ζ	0.0011 ±0.00005	0.0024 ±0.00012	0.0025 ±0.00012

3. Finite element model

In the next stage of the work, a finite element model of the presented beams was made using Midas NFX software. Modelling began with the construction of a geometrical model, i.e. giving the structure the appropriate dimensions and shapes, and defining some material constants describing the actual mechanical system.

Subsequently, the geometrical model was discretized using CHEXA type elements (six-sided isoperimetric solid element with 8 nodes). The composite filling was modelled as linear isotropic due to preliminary tests results. To improve the efficiency of the FEM, a structured meshing technique was applied. The model subjected to discretization is shown in Fig. 5.

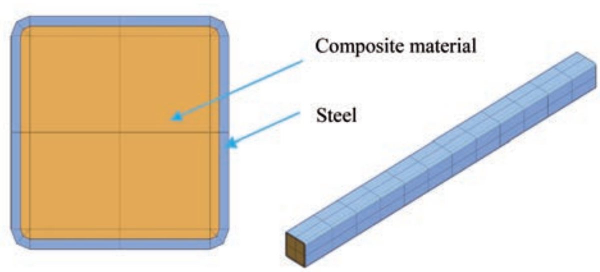

Fig. 5. Discretized model

The contact between steel and composite material was mode led as nodes coincidence, which is equivalent to welded contact model. The use of such a model was dictated by a clear analogy of the subjected objects to concrete filled steel tubes (CFST). Based on the research conducted on CFST, it can be concluded that both the steel coat and the composite filling transfer the load, behaving like beams made of one material [16–19].

The next step of the analysis was to determine a damping model describing the structure. Two damping models were selected for the analysis: Rayleigh damping and structural damping expressed using the complex stiffness.

The damping matrix in the Rayleigh model is composed of the linear combination of the mass matrix and the stiffness matrix of a structure [20, 21]:

$$\boldsymbol{C} = \alpha \boldsymbol{M} + \beta \boldsymbol{K} \tag{1}$$

where α, β – matrix multipliers expressed respectively in [s^{-1}] and [s], $\boldsymbol{K}$ – stiffness matrix, $\boldsymbol{M}$ – mass matrix.

For a mechanical system with multi degrees of freedom, the motion equation can be written in the following form:

$$\boldsymbol{M}\ddot{\boldsymbol{x}} + \boldsymbol{C}\dot{\boldsymbol{x}} + \boldsymbol{K}\boldsymbol{x} = \boldsymbol{P}(t) \tag{2}$$

where $\boldsymbol{P}(t)$ – force vector, $\ddot{\boldsymbol{x}}, \boldsymbol{x}, \boldsymbol{x}$ – vectors of acceleration, velocity and displacement, respectively.

Transforming the system into the modal coordinates yields the following:

$$\boldsymbol{\phi}^T\boldsymbol{M}\boldsymbol{\phi}\ddot{\boldsymbol{\xi}} + \boldsymbol{\phi}^T\boldsymbol{C}\boldsymbol{\phi}\dot{\boldsymbol{\xi}} + \boldsymbol{\phi}^T\boldsymbol{K}\boldsymbol{\phi}\boldsymbol{\xi} = \boldsymbol{P}(t) \tag{3}$$

where $\ddot{\boldsymbol{\xi}}, \dot{\boldsymbol{\xi}}, \boldsymbol{\xi}$ – generalized acceleration, velocity and displacement vectors, $\boldsymbol{\phi}$ – eigenvector matrix. Eq. (3) may be written in the form of n-uncoupled equations:

$$\ddot{\xi}_j + 2\zeta_j\omega_j\dot{\xi}_j + \omega_j^2\xi_j = \boldsymbol{P}_j(t) \tag{4}$$

where ω_j – j-th natural frequency of the system. Hence, the damping matrix can be written in the following form:

$$\boldsymbol{\phi}^T\boldsymbol{C}\boldsymbol{\phi} = \begin{bmatrix} \alpha + \beta{\omega_1}^2 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & \alpha + \beta{\omega_n}^2 \end{bmatrix} \tag{5}$$

According of the fact that the presented matrix is a symmetric matrix, it can be written as follows:

$$\begin{aligned} 2\zeta_1\omega_1 &= \alpha + \beta{\omega_1}^2 \\ 2\zeta_2\omega_2 &= \alpha + \beta{\omega_2}^2 \\ &\cdots \\ 2\zeta_n\omega_n &= \alpha + \beta{\omega_n}^2 \end{aligned} \tag{6}$$

where ζ – modal damping factor.

Based on the experimental data and algorithm proposed in [22], the following coefficients values were obtained: $\alpha = 6.1930\ s^{-1}$ oraz $\beta = 5.6301 \cdot 10^{-8}\ s$.

Alternatively, to describe the damping of the structure, a complex stiffness model was used. In this case, the model stiffness matrix $\boldsymbol{K}$ takes the following form:

$$\boldsymbol{K} = (1 + i\eta)\boldsymbol{K} \tag{9}$$

where i – imaginary unit, η – loss factor defined as

$$\eta = 2\zeta \tag{10}$$

Figure 6 presents the relation between the damping ratio and frequency for Rayleigh and structural damping.

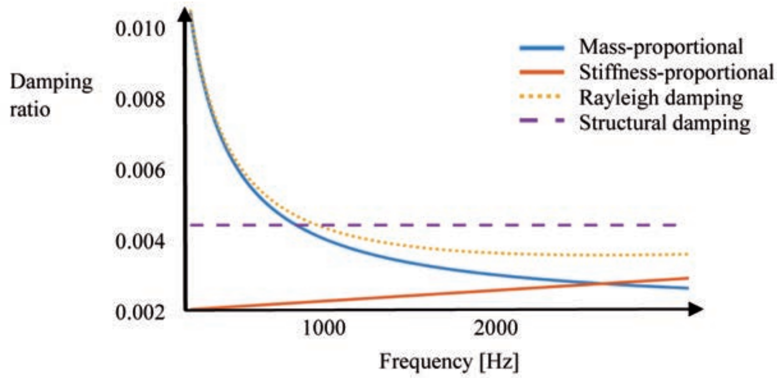

Fig. 6. Damping ratios derived from Rayleigh damping and structural damping model for M05 beam
Source: Authors.

Results

As a consequence of the proposed model updating procedure, the final model was obtained. Table 4 contains a comparison of the natural frequency values determined on the basis of the FEM model and on the basis of experimental data. The supplement to this (Table 4) is Fig. 7 showing the mode shapes visualization for the results obtained on the basis of the model and on the basis of the experiment.

The analysis of the natural frequencies values contained in Table 1 indicates slight discrepancies (at the level of 5%). They result from the properties of the FEM computing environment, which allows only a global change of parameters describing the model. Thus, it is possible to tune the model very precisely to only one mode shape. It should be noted that, despite the differences in frequency values for subsequent mode shapes, its compatibility has been achieved (Fig. 7), and comparable amplitude levels were also obtained (Fig. 8).

Table 4. Comparison of natural frequencies between FEM updated model and experimental results

Mode number	M05		M11	
	Experimental data	FEM model	Experimental data	FEM model
1.	350.40 Hz	349.49 Hz	339.17 Hz	338.5 Hz
2.	350.92 Hz	349.49 Hz	339.25 Hz	338.5 Hz
3.	922.55 Hz	946.26 Hz	896.65 Hz	914.31 Hz
4.	923.69 Hz	946.26 Hz	896.77 Hz	914.31 Hz
5.	1290.08 Hz	1291.60 Hz	1263.83 Hz	1247.00 Hz
6.	1693.96 Hz	1816.20 Hz	1660.74 Hz	1748.90 Hz
7.	1697.04 Hz	1816.20 Hz	1666.14 Hz	1748.90 Hz

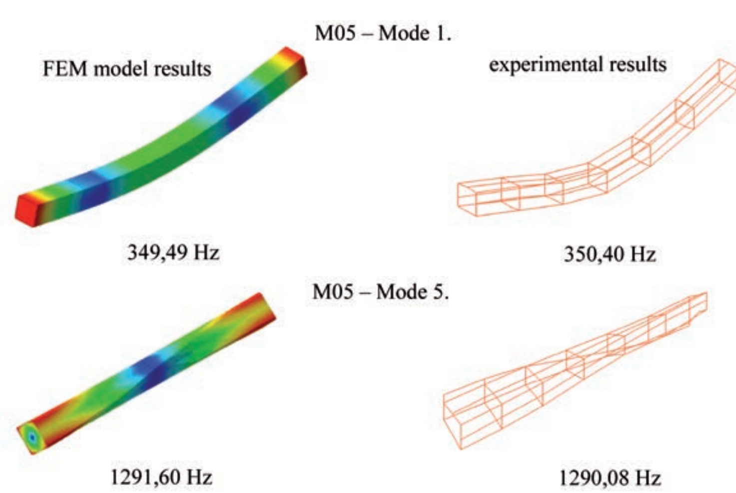

Fig. 7. Mode shapes comparison for M05 beam

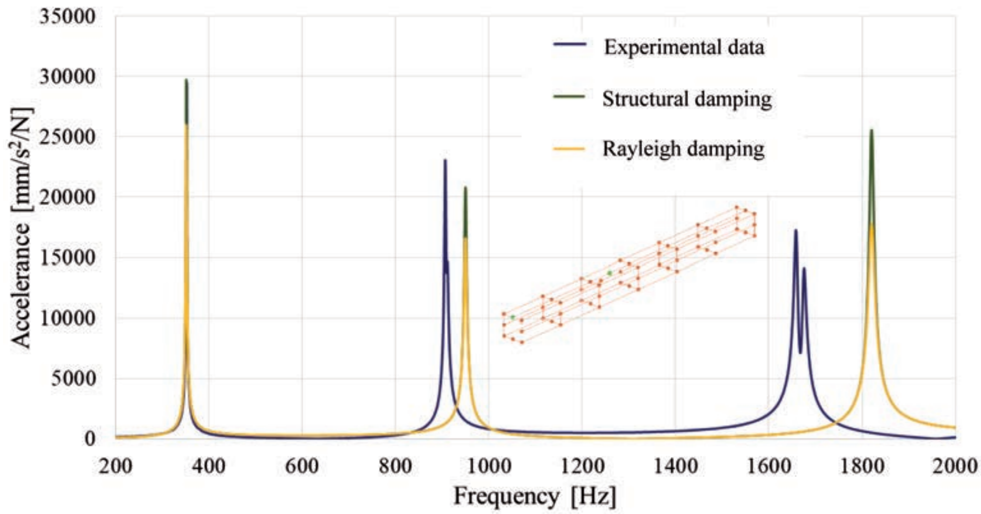

Fig. 8. Frequency response function comparison for M05 beam
Source: Authors.

Summary

In the presented paper, a model identification procedure was proposed. The process of building and updating a FEM model of a steel beam filled with composite material was carried out. The results obtained on the basis of the model were compared with the experimental data, achieving satisfactory model compatibility.

This paper also indicates the limitations of commercial FEM packages and the resulting need to develop a method that allows changing the local parameters of the model (individual elements of the matrix describing the system). Therefore, this should be the next stage of work.

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