ARCHIVE OF MECHANICAL ENGINEERING

Volume 66 2019 Number 3

DOI: 10.24425/ame.2019.129681

Key words: proportional control, composite beam, MFC, dynamics, FEM

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Use of an active element for dynamic control of a thin-walled laminated beam with kinematic excitation

The paper describes the dynamics of a composite cantilever beam with an active element. The vibrations of the kinematically excited beam are controlled with the use of a Macro Fiber Composite actuator. A proportional control algorithm is considered. During the analysis, actuator is powered by a time-varying voltage signal that is changed proportionally to the beam deflection. The MFC element control system with the implemented algorithm allowed for changing the stiffness of the tested structure. This is confirmed by the numerical and experimental results. Resonance curves for the beam with and without control are determined. The results show a very good agreement in qualitative terms.

1. Introduction

The dynamics of composite structures is fundamental for describing the behavior of these structures. Systems made of thin-walled composite structures equipped with active elements are particularly interesting in this respect. The basic task of these elements is affect the behavior of an entire structure. The most widely used active elements include: PZT elements, PVDF piezoelectric polymers, shape memory alloy, electrorheological fluids and magnetostrictive materials. The most modern active elements considered in various types of research are Macro Fiber Composite (MFC) elements which have recently found many interesting applications. In papers [1, 2], the design and practical applications of MFC elements are presented. The author of [3] described and investigated the properties of MFC

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elements. Experimental and numerical studies were conducted on unmanned aerial vehicles equipped with active elements of MFC type. The effect of active elements and their location on the behavior of a flying vehicle in the wind tunnel was studied. In [4], the effectiveness of an MFC active element in energy recovery was determined. The effect of system load on the generated voltage of the active element was described. Experimental and numerical tests were carried out. The method of efficient modeling of MFC active elements characterized by the d_{33} effect in the Abaqus environment was presented in [5]. The author tested a cantilever laminated beam with an embedded actuator. Simple and inverse piezoelectric effects were examined by FEM and laboratory tests. Numerical and experimental modal analyses of an active three-blade rotor were presented in [6]. The frequencies and modes of vibrations of a fixed system were analyzed. In [7], the authors proposed a way of modeling MFC elements in dynamic studies. The investigated object was a rotor composed of a hub and three laminated active blades. The study checked changes in the natural frequency of vibrations, depending on the rotational speed and constant voltage applied to the active element. The numerical and experimental dynamics of a three-blade rotor made of active laminated blades was considered in [8]. The rotor was assumed to rotate at a constant speed. The authors determined the effect of the hub's rotational speed and the piezoelectric effect on the dynamics of the blades. Researchers [9] described the dynamic behavior of a composite beam rotating at a constant angular velocity and excited by an MFC activator. The numerical research was conducted by the implicit method, whereas selected cases were verified experimentally. The numerical and experimental results show a very high agreement and are presented in the form of amplitude-frequency characteristics.

A review of the literature on the dynamics of rotating beams and the control of active structures was given in [10]. Both numerical and experimental studies were considered. In [11, 12], the authors gave an overview of methods for vibration reduction in rotating structures. They presented numerical, analytical and experimental research. In [13], analytical studies were performed on vibrations reduction of a freely supported beam with an activator using the open loop and closed loop control system. The results allowed them to determine the effect of the piezoelectric actuator position on the effectiveness of beam vibration reduction. Laminated beams with PZT elements are studied in [14]. Theoretical studies were carried out using appropriate control algorithms to reduce vibrations of a cantilever beam. Sodano [15] investigated the use of MFC components for damping vibrations. An appropriate algorithm with feedback was used to control an active cantilever aluminum beam. It was shown that MFC actuators could be effectively used to damp vibrations as well as to control active structures.

In this paper, the dynamics of a composite cantilever beam with an active element is studied. The MFC element is used to control the structure, which is forced by kinematic excitation. A simple proportional algorithm is proposed, and the MFC voltage signal is proportional to the deflection of the structure.

Experimental and numerical analyses are performed. An important aspect of this study is the numerical analysis made by the finite element method with the use of the Abaqus software.

2. Experimental study

The experiments were performed on a beam with two embedded elements. The beam had the layup-ply configuration [±45/90]s and was made of a glass-epoxy unidirectional prepreg. A Macro Fiber Composite (MFC) element and a strain gauge were fixed on the opposite sides of the beam surface. The first one was applied as an actuator and the other one was used as a sensor. The actuator was the MFC element type M-8528-P1 with the d_{33} piezoelectric effect. This means that the piezoelectric element is deformed in the same direction as the lines of the electric field. All dimensions of the system are shown in Fig. 1.

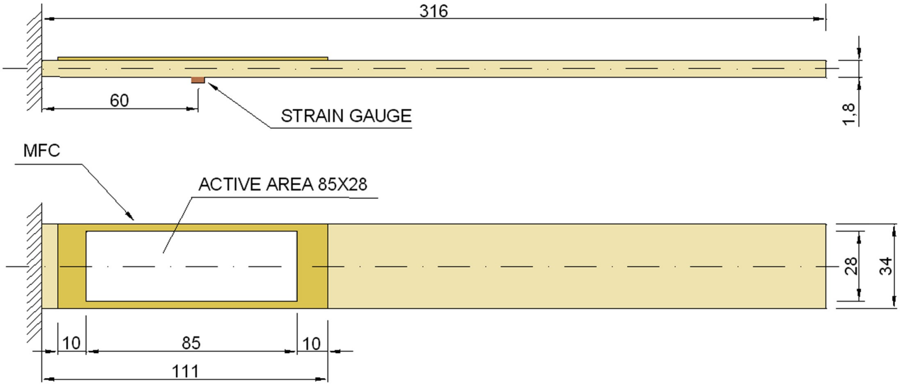
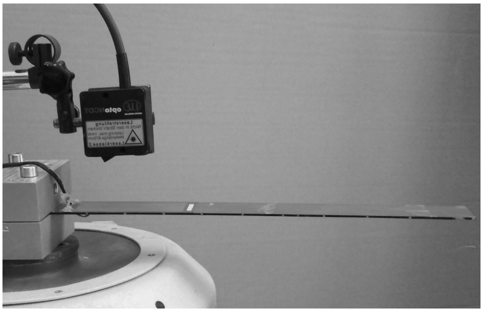
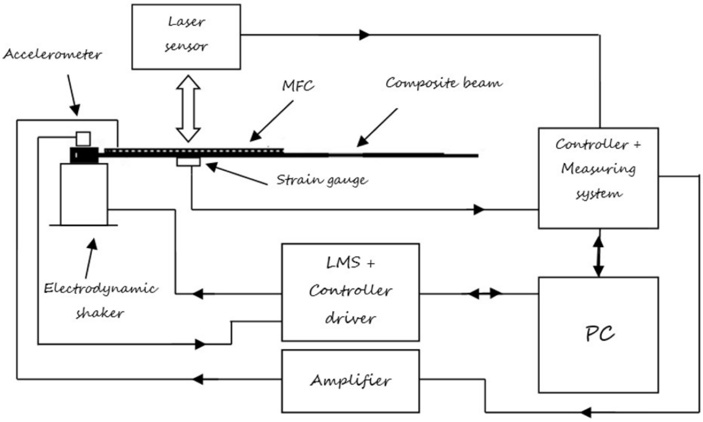


Fig. 1. Geometry of the active beam (dimensions in mm)

The beam was mounted on an electrodynamic shaker by means of a special grip. In the tests, the system was excited kinematically. The amplitude of the shaker armature motion was maintained constant (1 mm), while the frequency was slowly changed (the so-called sine test in a LMS system). The range of excitation frequency corresponded to the first bending vibrations mode. The beam response recorded by the strain gauge was used as the input to the controller, which was realized as the proportional algorithm (P). The output signal from the controller was converted with a high voltage amplifier and used to supply the MFC patch. Additionally, the displacement of one point of the beam structure was measured by a laser sensor. The measuring range of the laser sensor was ±5 mm, therefore this point was located nearby the clamped beam's end (about 60 mm). A laboratory test stand is shown in Fig. 2a, while Fig. 2b shows the scheme of the experimental setup.



(a) Test stand



(b) The scheme of the experimental setup

Fig. 2. Experimental analysis

3. FEM analysis

In the first step of the numerical analysis, a beam model was built by the finite element method (Fig. 3). The numerical model consists of a host structure (laminated beam) and an active element (MFC). The modeling was done using the Abaqus software package. The laminated beam was modeled using SC8R continuum shell finite elements with 6 layers in the configuration [±45/90]s. To this end, the layup-ply technique was used. Due to the complex structure of the MFC element, its equivalent model was proposed. The active element was a homogeneous isotropic structure, which was modeled using C3D20RE solids elements. The properties of MFC equivalent model were determined in static studies [9]. All material parameters are given in Table 1.

In the second step of the numerical analysis, a modal analysis was performed. The Lanczos algorithm was used to determine the first natural frequency of the cantilever laminated beam with the MFC element. Next, a dynamic analysis was performed. The time domain analysis was carried out using the implicit procedure.

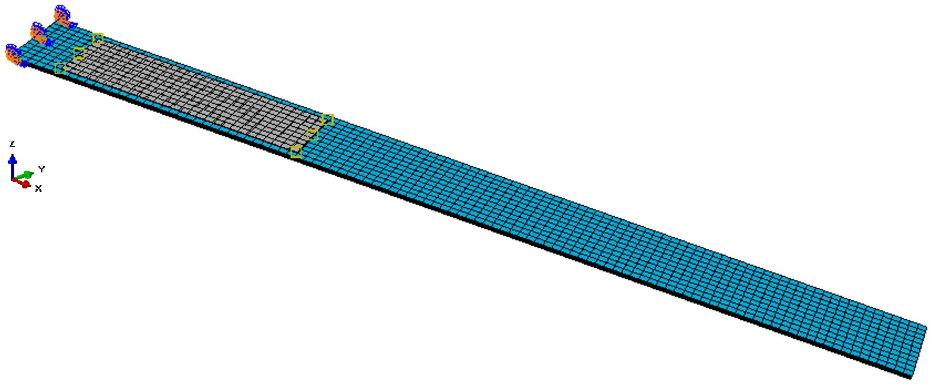


Fig. 3. FE model of the cantilever beam with the MFC element

Kinematic excitation was defined with mechanical boundary conditions. For the clamped end of the beam, the displacements of nodes were set equal to zero in the x- and y-directions and periodically changed in the z-direction (Fig. 3). The voltage control signal applied to the actuator was defined in the electrical domain with electrical boundary conditions. The voltage was calculated using the control algorithm from beam response. In practice, the development of an UAMP procedure was required. The procedure allowed for getting a time-varying voltage signal that was developed in the form of a subroutine.

Table 1.

Properties of the used elements

Laminate		
Longitudinal modulus	46.4	[GPa]
Transverse modulus	14.9	[GPa]
Shear modulus	5.2	[GPa]
Poisson’s ratio	0.27	[-]
Density	2032	[kg/m^3]
Macro Fiber Composite		
Young’s modulus	6.75	[GPa]
Poisson’s ratio	0.31	[-]
Piezoelectric constant	$1.02 \cdot 10^{-7}$	[m/V]
Density	5440	[kg/m^3]

4. Results

As a result of the experimental tests, resonance curves are determined for the beam with and without control. For structure with kinematic excitation, relative or/and absolute motion is observed. The strain gauge signal provides information about relative motion, i.e., the level of beam oscillations. However, absolute motion is the sum of relative beam motion and beam excitation. The total displacement of the selected point of the structure was recorded by a laser sensor. The knowledge of both signals allows for verification of the numerical model. The experimental resonance curves are compared with the numerical results (Fig. 4). The curves show the vibrations amplitudes (A) of beam absolute motion versus excitation frequency (f), where both variables were reduced to a dimensionless form. The characteristics show quite a good agreement between the numerical and experimental results. It is worth noting that the resonant frequency of the system was about $f_0 = 9$ Hz. However, A_0 is the resonance vibrations amplitude.

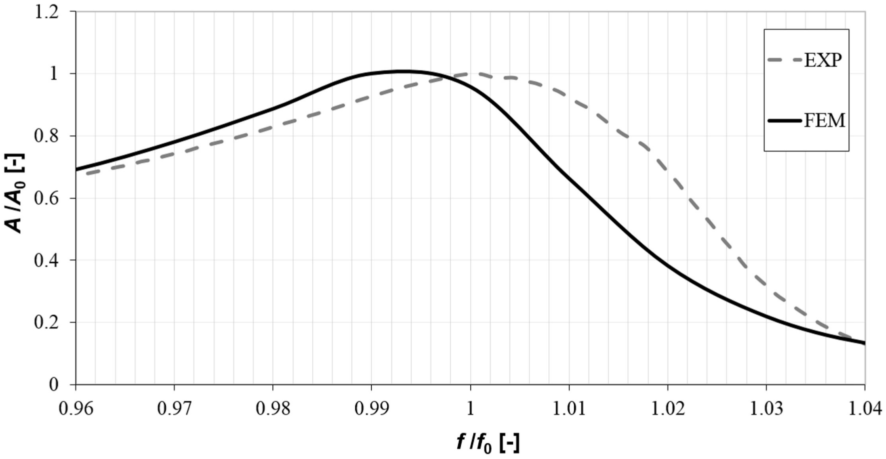


Fig. 4. Resonance curves describing the system without control in absolute motion

Finally, the system with activated control was tested. During the analysis of beam dynamics with kinematic excitation, an MFC element was applied to control vibration. The piezoelectric actuator was powered by a time-varying voltage signal. Generally, this signal was calculated from relative beam response using the proportional algorithm. The resonant characteristics of the system with control were determined and compared with the results obtained for the system without control. The curves describing the system in relative motion are presented in Fig. 5. The application of the active element control system (proportional algorithm) allowed us to change the stiffness of the tested system, as shown in Fig. 5. As a result, the curve peaks obtained during control are shifted to the left in the plot. The same dependence was observed in the experimental and numerical analyses. However, in the experiments, the maximum amplitude of vibrations was reduced by 67%, while in the numerical analysis the vibration amplitude was reduced by approximately 20%.

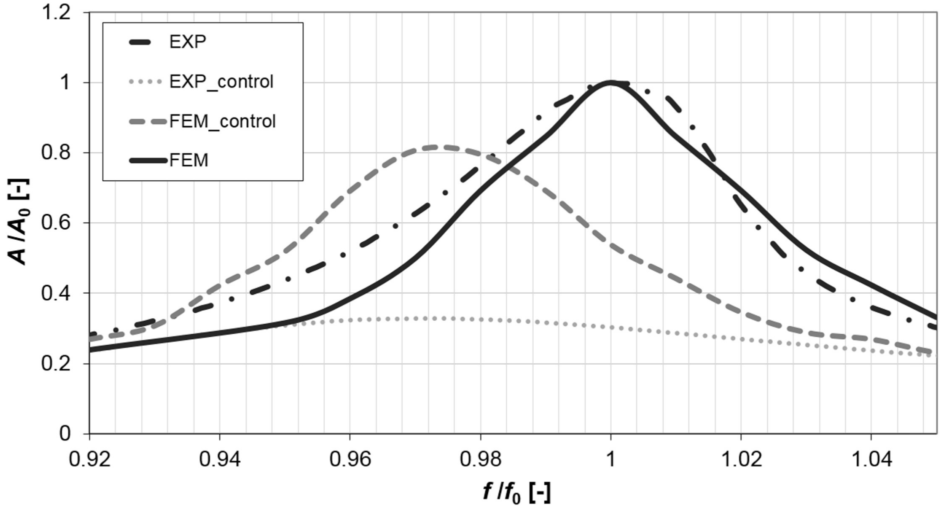


Fig. 5. Resonance curves describing the system in relative motion

5. Conclusions

The dynamics of a composite cantilever beam with an active element was investigated numerically and experimentally in this paper. The kinematically excited beam was controlled with the use of a Macro Fiber Composite element. The proportional control algorithm was employed. During the analysis with kinematic excitation, the MFC element was powered by a time-varying voltage signal that was changed proportionally to the beam deflection. Resonance curves of the beam with and without control were determined. The MFC active element control system with the implemented proportional algorithm allowed us to change the stiffness of the tested system. This is confirmed by the numerical and experimental results shown in Fig. 5, where the curve peaks are shifted to the left (stiffness reduction of about 3%). The results show a very good agreement in qualitative terms. However, no quantitative agreement was obtained. Therefore, the maximum amplitude of vibrations in the experimental tests was reduced by 67%, while in the numerical analysis only by approximately 20%. These discrepancies can be explained by the fact that the numerical model is idealized and does not include many factors. For example, in the real system there may be time delays in the feedback loop that were not modeled numerically in this study. In addition, the properties of the measuring system and the actuator supply system wherein inertial systems may occur were not taken into account in this study. Future studies will involve numerical model modification in order to take into account inertial elements in the system.

Acknowledgements

The project/research was financed in the framework of the project Lublin University of Technology-Regional Excellence Initiative, funded by the Polish Ministry of Science and Higher Education (contract no. 030/RID/2018/19).

Calculations were carried out at the Academic Computer Centre in Gdansk, Poland.

Manuscript received by Editorial Board, May 14,2019;
final version, September 09, 2019.

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