Marcin BOCHEŃSKI[1], Michał AUGUSTYNIAK[2,3], Mariusz OSTROWSKI[3], Wojciech JARZYNA[3]

Faculty of Mechanical Engineering, Lublin University of Technology (1), Induster Sp. z o. o. (2),
Faculty of Electrical Engineering and Computer Science, Lublin University of Technology (3)

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The influence of control algorithms on the effectiveness of vibration reduction of an active composite beam

Abstract. *The paper presents application of the MFC actuator and selected control algorithms to the suppression of the composite cantilever beam vibrations. The first part concentrates on the identification of the real structure's parameters. The numerical model is based on the Euler-Bernoulli beam theory with a nonlinear curvature component. The second part draws on numerical simulations and leads to the identification of optimal control parameters. Finally, the determined parameters are examined in an experimental laboratory system equipped with a DSP controller.*

Streszczenie. *W artykule przedstawiono zastosowanie siłownika MFC i wybranych algorytmów sterowania do redukcji drgań kompozytowej belki wysięgnikowej oraz wyniki identyfikacji parametrów układu rzeczywistego. Model numeryczny belki zbudowano w oparciu o teorię Eulera-Bernoulli'ego z uwzględnieniem nieliniowej krzywizny. Na podstawie wyników symulacji numerycznych wskazano optymalne nastawy kontrolerów, które następnie zweryfikowano na stanowisku laboratoryjnym wyposażonym w kontroler DSP. (**Wpływ algorytmów sterowania na skuteczność redukcji drgań aktywnej belki kompozytowej**).*

Keywords: MFC actuator, control algorithm, active beam, DSP controller.
Słowa kluczowe: aktywator MFC, algorytm tłumienia drgań, belka aktywna, kontroler DSP.

Introduction

The tendency to improve operational parameters or reduce the noise and vibration of the mechanical structures has been intensively developed in recent years. Embedding piezoelectric actuators and sensors in the system, together with a proper control algorithm, allow to create so called smart structures. This paper is focused on the application of selected control algorithms (P, PID, PPF) to a flexible, geometrically nonlinear composite cantilever beam. The structure can be excited in a double manner. Macro Fiber Composite (MFC) actuator M-8503-P1 plays the role of executor for controller and inductor acting as source of external harmonic disturbance [1, 2]. Displacement of the beam's free end (response of the structure) is measured by the laser vibrometer optoNCDT Micro Epsilon. Numerical results are obtained using MatLab software and experimental tests are performed on DSP MicroDAQ controller with OMAP L137 processor and 16-bit A/D-D/A converters.

Mathematical model

Equation of motion of the beam model presented in Figure 1 has been derived by Hamilton's principle of least action. The model is considered as the Euler-Bernoulli beam with added nonlinear curvature component [3]. Taking into account the beam's gravity force, first bending mode and using Galerkin-Bubnov discretisation procedure, the differential equation for free vibrations takes form:

$$\ddot{x}+\omega_s^2 x+\alpha x^3+\delta\frac{\ddot{x}x^2+x\dot{x}^2}{\sqrt{1-\psi x^2}}+\gamma=0 \tag{1}$$

Next, viscous damping and two external excitation terms are completed:

$$\ddot{x}+2\mu\omega_S^*\dot{x}+\omega_S^{*2}x+\alpha x^3+\delta\frac{\ddot{x}x^2+x\dot{x}^2}{\sqrt{1-\psi x^2}}+\gamma= =U_C k_C\sin(2\Omega t)+U_P k_P\sin(\omega_P t) \tag{2}$$

where U_C, U_P are the supply voltage of the inductor and the MFC, respectively. To determine the parameters of Equation (1) the following material coefficients of the real structure are used: E=25.5×10^{9}Pa, ρ=2100kg/m^3, b=13mm, h=2.2mm, L=220mm.

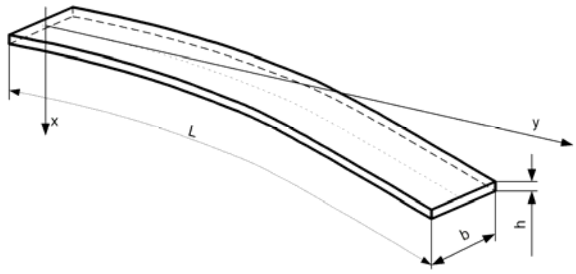

Fig.1. Model of the structure

To identify unknown parameters of Equation (2) such as μ, k_C, k_P the experimental results are used. Block diagram of the laboratory system is presented in Figure 2. Figure 3 shows frequency-amplitude diagrams for different amplitudes and different manner of excitation (inductor or MFC actuator). The amplitudes of inductor and MFC voltage are matched to obtain the same level of beam's response (amplitude of tip deflection). The values of parameters are presented in Table 1, where ω_S is the theoretical beam's natural frequency and ω_S^* is the natural frequency based on laboratory and numerical curves matching.

Table 1. Parameters of Equations (1) and (2)

ω_S	ω_S^*	μ	α	γ	δ	ψ	k_C	k_P
rad/s	*rad/s*	-	*rad/s²*	*m/s²*	-	-	*rad/s²V*	*rad/s²V*
160.7522	150.3283	0.0053	195923	22.2955	10.4383	7.5818	0.214577	0.010606
							0.262234	0.0110273
							0.293505	0.010625
		Average value:					0.256772	0.010501

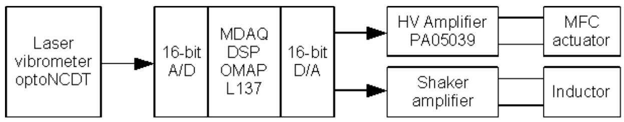

Fig.2. Block diagram of the experimental system

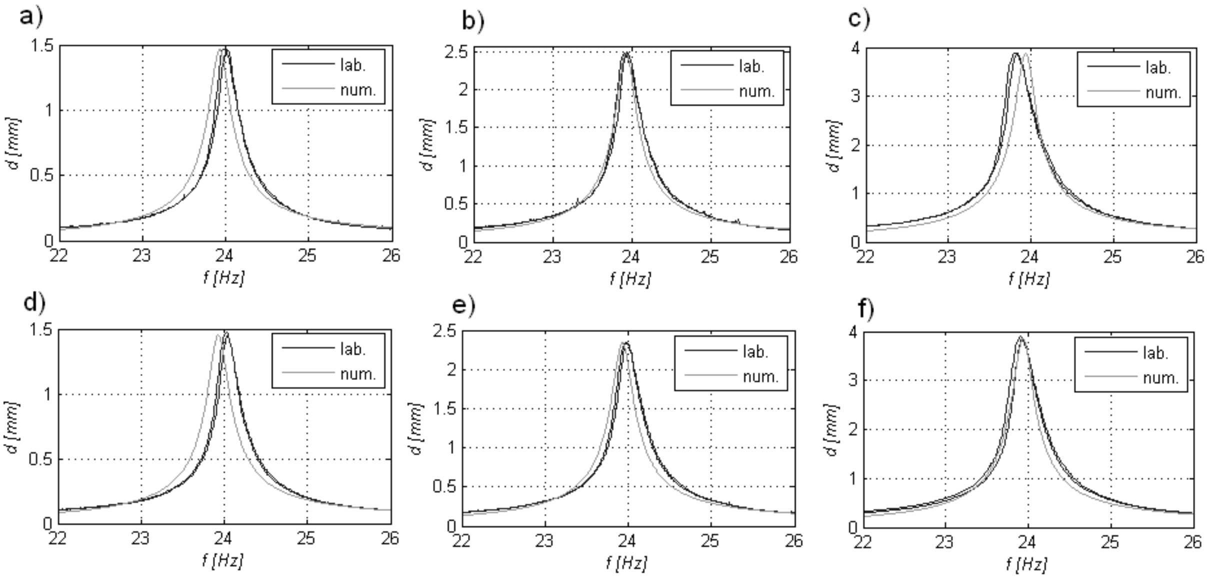

Fig.3. Experimental and numerical frequency-amplitude diagrams; a) U_C=3.76V, b) U_C=5.20V, c) U_C=7.24V, d) U_P=75V, e) U_P=125V, f) U_P=200V

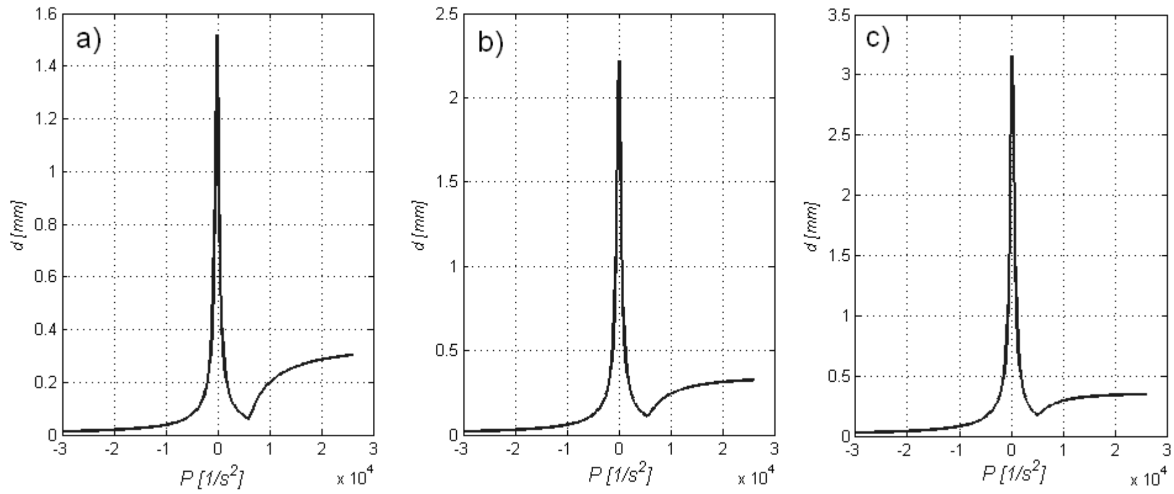

Fig. 4. Beam's tip deflection versus proportional feedback P values – numerical results; excitation by the inductor a) U_C=3.76V, b) U_C=5.20V, c) U_C=7.24V

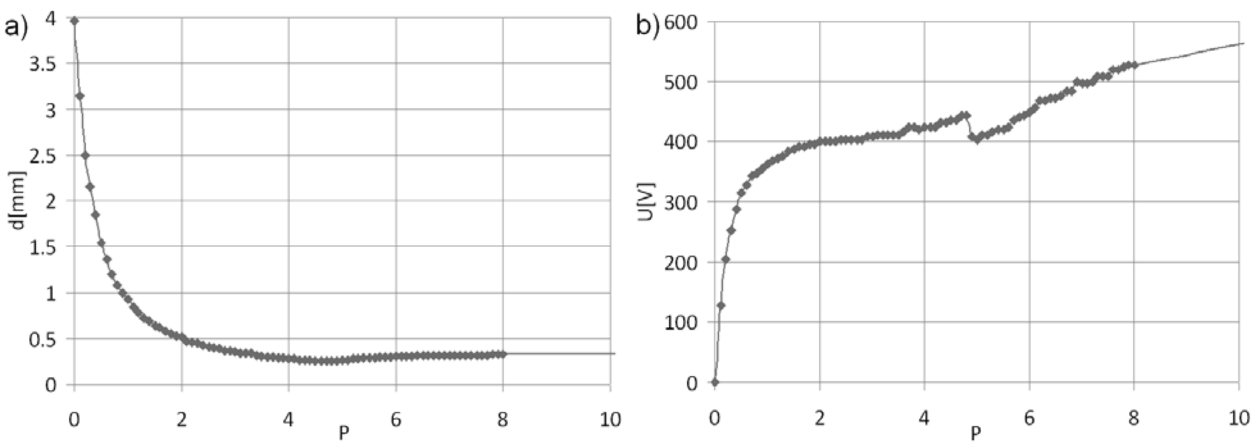

Fig.5. Beam's tip deflection (a) and MFC voltage (b) versus proportional feedback P values – experimental results U_C=7.24V

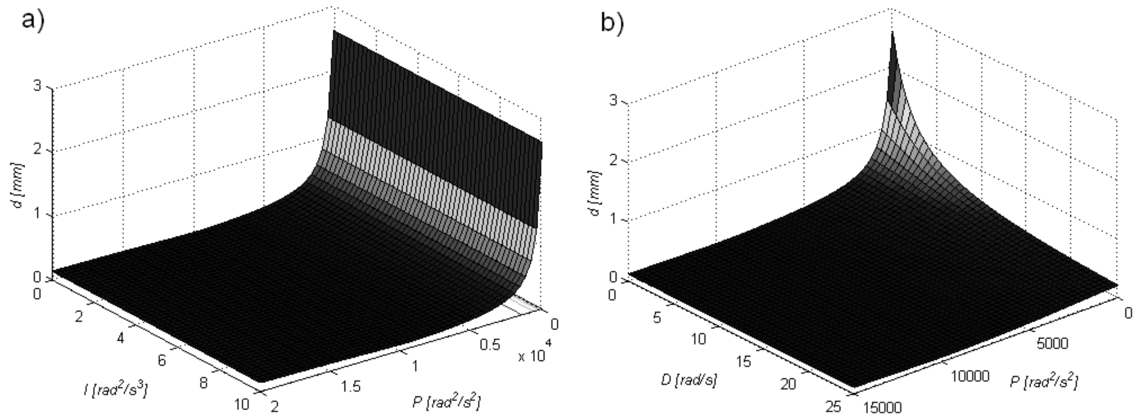

Fig.6. Beam's tip deflection versus *PID* controller parameters – numerical results U_C=7.24V a) influence of *PI, D*=0 b) influence of *PD, I*=0

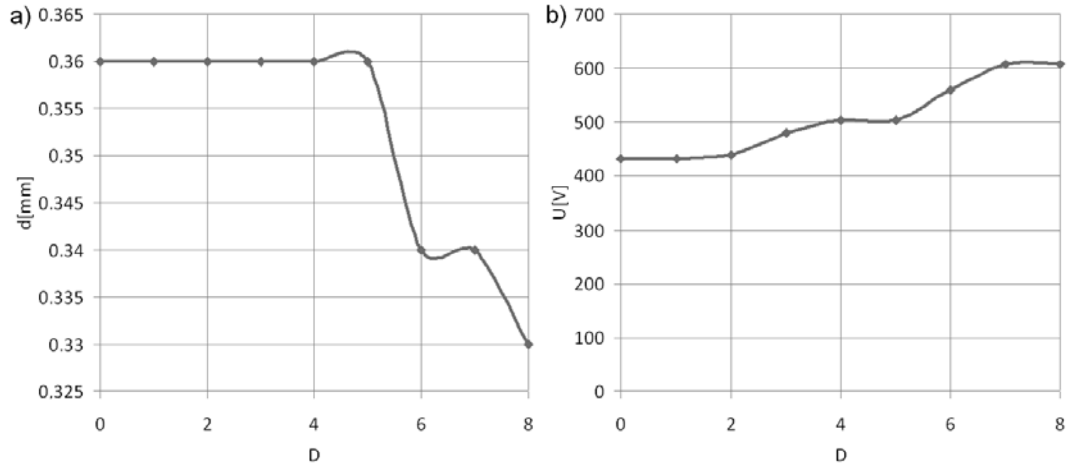

Fig. 7. Beam's tip deflection (a) and MFC voltage (b) versus derivative feedback *D* values – experimental results U_C=7.24V, *P*=3, *I*=0

To the further analysis, three control strategies have been adopted: proportional *P*, classical *PID* and Positive Position Feedback (*PPF*) [4]. The mathematical description of the structure with individual controller is presented in Equations (3)-(5).

- *P*

$$\ddot{x}+2\mu\omega_S^*\dot{x}+\omega_S^{*2}x+\alpha x^3+\delta\frac{\ddot{x}x^2+x\dot{x}^2}{\sqrt{1-\psi x^2}}+\gamma = \\ =U_C k_C\sin\left(2\Omega t\right)+Px \tag{3}$$

- *PID*

$$\ddot{x}+2\mu\omega_s^*\dot{x}+\omega_s^{*2}x+\alpha x^3+\delta\frac{\ddot{x}x^2+x\dot{x}^2}{\sqrt{1-\psi x^2}}+\gamma \\ =U_C k_C\sin\left(2\Omega t\right)+Px+D\dot{x}+I\int xdt \tag{4}$$

- *PPF*

$$\begin{cases}\ddot{x}+2\mu\omega_s^*\dot{x}+\omega_s^{*2}x+\alpha x^3+\delta\dfrac{\ddot{x}x^2+x\dot{x}^2}{\sqrt{1-\psi x^2}}+\gamma \\ \qquad =U_C k_C\sin\left(2\Omega t\right)+g_{PPF}\,y \\ \ddot{y}+2\xi\omega_c\dot{y}+\omega_c^2 y=k_{PPF}\,x\end{cases} \tag{5}$$

Numerical results and experimental verification

For all the numerical simulations the physical limitation of MFC voltage (-500; +1500)V was taken into account. Firstly, values of proportional coupling were tested. Numerical results are presented in Figure 4, while experimental response is shown in Figure 5. Three amplitudes of excitation were tested numerically: U_C=3.76V, U_C=5.20V and U_C=7.24V when frequency of excitation corresponds to the maximum of the amplitude-frequency characteristic of the examined U_C. It is shown that qualitative characteristic is almost independent of excitation's amplitude and for positive values of *P* the local minimum of the beam's deflection occurs. Experimental tests confirm the appearance of this local minimum, but for negative *P* values the system behaved unstably. For positive *P* values numerical and experimental effectiveness of the beam's vibrations reduction is almost the same.

Numerical tests for *PID* controller show that the integrated term does not improve effectiveness of the controller (Fig.6a), which was confirmed by the laboratory tests [5]. The numerical influence of the proportional term is the same as for the *P* regulator. However, increasing the *D* value effectiveness of the *PD* controller (*I*=0) was slightly better (Fig.6b). It was also proved by laboratory tests, but 10% of the beam's displacement reduction caused a 40% MFC voltage increase (Fig.7).

The *PPF* control technique consists in additional electric oscillator connected to the mechanical structure by sensor and actuator. There are two coupling factors: k_{PPF}, g_{PPF} and the controller's damping ratio ξ. The natural frequency of the controller ω_C is the same as the frequency of excitation 2Ω. Gain g_{PPF} affect on change of the controller's amplitude value but not for the beam's vibrations level [3]. Therefore only k_{PPF} and ξ were taken into consideration. Figures 8 and 9 show the beam's tip displacement versus gain k_{PPF} and the controller's damping ratio ξ obtained from numerical results and experimental tests for amplitude of excitation U_C=7.24V, respectively.

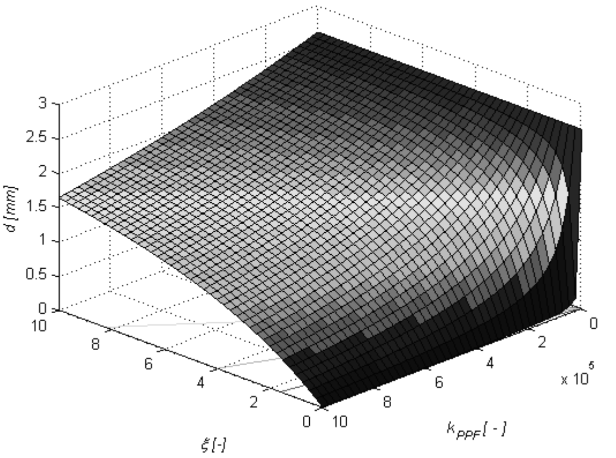

Fig. 8. Beam's tip displacement versus gain k_{PPF} and controller's damping ratio ξ – numerical results $U_C=7.24V$

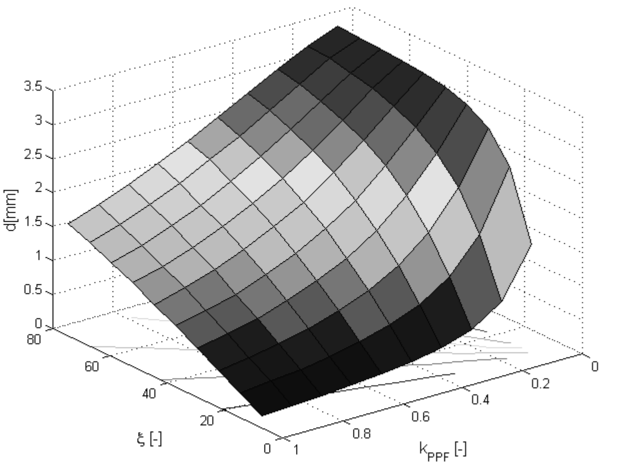

Fig.9. Beam's tip displacement versus gain k_{PPF} and controller's damping ratio ξ – experimental results U_C=7.24V

Owing to different levels of signal value on numerical model and laboratory's electric oscillator, the values of k_{PPF} and ξ have been presented at a different scale, but the effectiveness of the beam's vibrations reduction level is almost the same and the qualitative influence of gain and damping ratio was confirmed by the experimental tests.

Conclusions

The paper presents numerical and experimental results of the influence of selected control algorithms on the dynamical response of an active composite beam. Predominantly, numerical results have been confirmed by laboratory tests. Solely for negative P value and small PPF controller damping ratio ξ the real structure shows a tendency towards instability. The P and PD algorithms are simple, fast and quite often sufficient. The maximum vibrations reduction achieves even 90%. Theoretical effectiveness of PPF is clearly higher, but only for a small damping ratio when an unstable system's response can practically occur. To improve the performance of the laboratory stand, low-pass filters are required.

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***Authors**: dr inż. Marcin Bocheński, Politechnika Lubelska, Wydział Mechaniczny, Katedra Mechaniki Stosowanej, ul. Nadbystrzycka 36, 20-618 Lublin, E-mail: m.bochenski@pollub.pl; mgr inż. Michał Augustyniak, Politechnika Lubelska, Wydział Elektrotechniki i Informatyki, Katedra Napędów i Maszyn Elektrycznych (doktorant), Induster Sp. z o.o., ul. Boh. Monte Cassino 3a, 20-808 Lublin, E-mail: michal.augustyniak@induster.com.pl; inż. Mariusz Ostrowski, Politechnika Lubelska, student specjalności „Napędy Mikroprocesorowe w Automatyce Przemysłowej", Wydział Elektrotechniki i Informatyki, E-mail: vostrowskiv@gmail.com; dr hab. inż. Wojciech Jarzyna, Politechnika Lubelska, Wydział Elektrotechniki i Informatyki, Katedra Napędów i Maszyn Elektrycznych, ul. Nadbystrzycka 38a, 20-618 Lublin, E-mail: w.jarzyna@pollub.pl.*

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