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ACTIVE VIBRATION CONTROL SYSTEM FOR 3D MECHANICAL STRUCTURES

SUMMARY

The active vibration control system for 3D mechanical structures is presented in the paper. The steel space framework which consists of 126 steel bars and 44 aluminium joints is a subject of our investigations. It is equipped in two piezoelectric stacks which are a part of chosen vertical bars in a plane X-Z. Bars, joints and piezo-stacks have got glue connections. The mathematical model was decoupled to change Two Input Two Output (TITO) system into two Single Input Single Output (SISO) systems. Such approach allowed us to design simple control laws with help of computer simulation procedure. In the last chapter the investigations on the laboratory stand of the active vibration control system with local controllers PD are described. Experimental results have proved that these PD controllers work good increasing the damping level. Additional damp in the system causes the excellent vibration reduction for the whole mechanical structure.

Keywords*: active vibration control, piezo-stack actuators, decoupled system, active structure*

AKTYWNY UKŁAD STEROWANIA DRGANIAMI
PRZESTRZENNEJ PRĘTOWEJ KONSTRUKCJI MECHANICZNEJ

*Aktywne sterowanie drganiami przestrzennej konstrukcji prętowej zostało przedstawione w poniższym artykule. Obiektem badań jest przestrzenna konstrukcja prętowa zawierająca 126 elementów prętowych, 44 aluminiowe węzły oraz dwa piezostosy. Wspomniane piezoelektryczne aktuatory stosowe zostały wklejone do konstrukcji w wybranych pionowych elementach prętowych w płaszczyźnie X-Z. Globalny model matematyczny obiektu jest układem o dwóch wejściach i dwóch wyjściach (*TITO*), który na potrzeby projektowania praw sterowania został rozprzężony na dwa podukłady o jednym wejściu i jednym wyjściu (*SISO*). Projektowanie praw sterowania zostało przeprowadzone na drodze badań symulacyjnych w środowisku Matlab. W końcowej części artykułu opisano badania eksperymentalne wraz z zaprojektowanymi lokalnymi regulatorami PD. Wyniki tych badań udowadniają, że wspomniane regulatory PD zwiększają poziom tłumienia. Tym samym dodatkowe tłumieniem które pojawiło się w układzie powoduje znakomitą redukcję drgań całej konstrukcji mechanicznej.*

Słowa kluczowe*: aktywne sterowanie drganiami, piezostos, aktywne sterowanie, rozprzężony układ*

1. INTRODUCTION

The vibrations control of the 3D bar structures has a great practical interest in a many industrial applications, e.g. modern structures of the huge space vehicles, aircrafts, solar batteries, robots, rotors, etc. Two demands are required in design of such structures. The excellent dynamic behavior is the first one in order to guarantee the stability of the structure and high precision pointing. The necessity to obtain light structures, which allow reduce the cost is another one. However, these two requirements are often contradictory. The weak internal damping in light structures hinders the accuracy requirements (Premount 2002).

These difficulties can be overcame by applying recently developed advanced materials, for instance, piezoelectric materials. Several researchers have proved that piezoelectric elements can effectively counteract the vibrations (Pietrzko and Mao 2009, Szolc and Pochanke 2010, Pietrzakowski 2004). Of course, the appropriate active damping of these vibrations can be achieved by the optimal location and control of the piezo-elements in the structure.

The vibration control of 3D mechanical structure with two piezo-stacks as actuators and two eddy-current probes as sensors is consider in the paper. It leads to two input and two output (TITO) control system. Exactly saying, control law was designed to eliminate the vibrations of the structure free end in the plane parallel to the fixed base (2D control). By proper excitations the open loop system was decoupled into two single input and single output (SISO) subsystems. For the decoupled system local control laws were designed in the computer simulation procedure. Next, the experimental stand was designed with actuators and sensors located in the structure. The searching procedure of quasi-optimal location was presented in earlier paper (Gosiewski and Koszewnik 2010) while identification procedure of open-loop system dynamics will be described in (Koszewnik and Gosiewski 2011). The experimental results confirm the analytical and simulation considerations.

2. MATHEMATICAL MODEL OF 3D STRUCTURE

Following steps are proposed for the design of the vibration control systems for mechanical structures:

1. Analytical and creative part of the design.
2. Determination of the optimum location of sensors and actuators within the structure (with the use of the finite element method, FEM).
3. Identification of the mathematical model of the vibration process.
4. Control law design.
5. Experimental verification of the design.
6. Return to the first step, if the results are not satisfactory.

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The first three steps was carried out and their results are presented in (Gosiewski and Koszewnik 2010, Koszewnik and Gosiewski 2011).

The laboratory stand with the controlled structure is shown in Figure 1. The identification procedure was carried out for the case when actuators were used as input and sensors as output for different form of excitations. We were looking for such excitation form which allows us to decouple TITO system into two SISO subsystems, one – to control vibration in X direction and second one – to control vibration in Y direction in the sensors plane. It was appeared that the best excitations are obtained when piezo-actuators generate force moments around X and Y axis, respectively. For both excitations the vibrations were measured by both sensors directed in X, Y axis, respectively. As a results of identification procedure we have obtained model in the form of matrix transfer function given in equation (1).

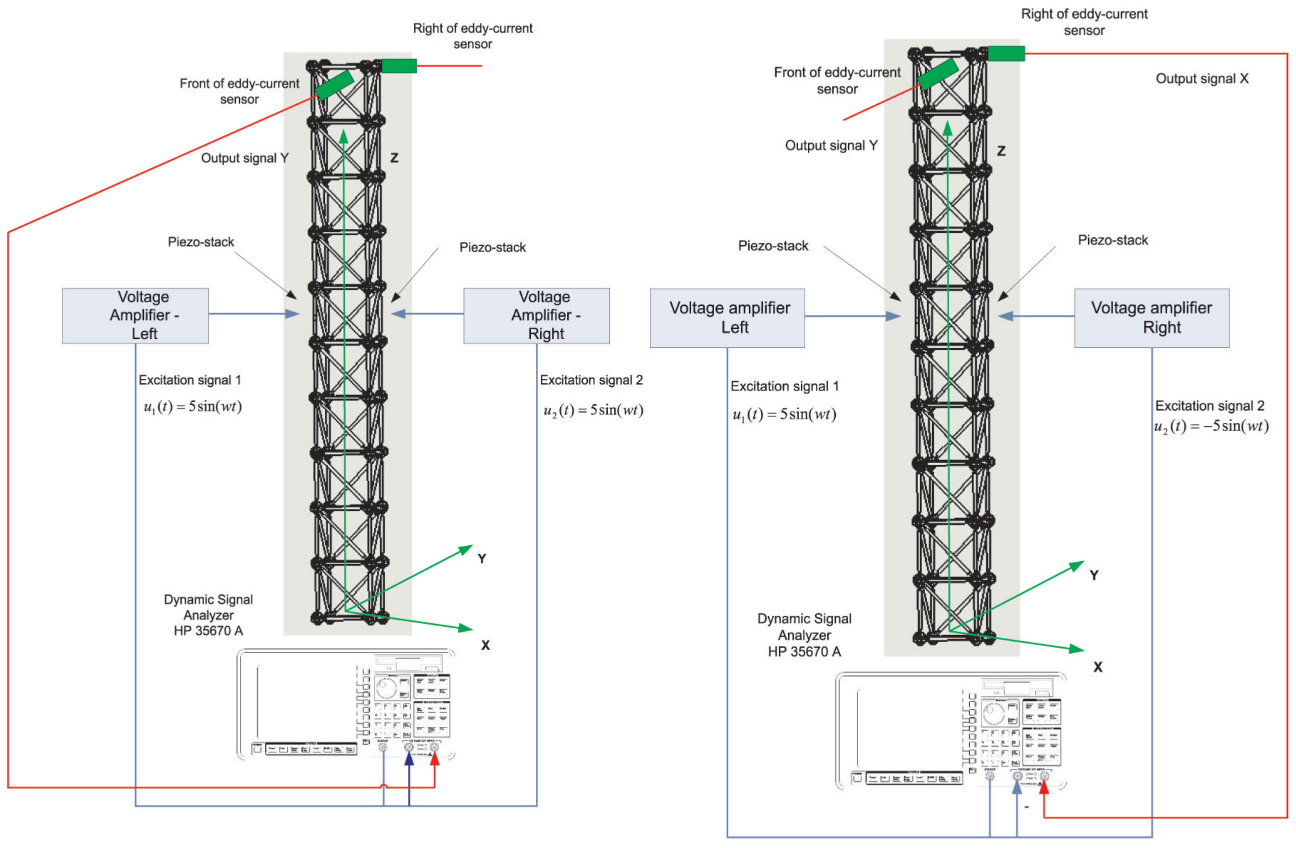

Fig. 1. Laboratory stand for the identification purposes

$$\mathbf{H}(s) = \begin{bmatrix} \mathrm{H}_{XX}(s) & \mathrm{H}_{XY}(s) \\ \mathrm{H}_{YX}(s) & \mathrm{H}_{YY}(s) \end{bmatrix} \tag{1}$$

$$H_{XX}(s) = 0{,}014163 \frac{\left(s^2 + 35{,}84s + 2{,}104e4\right)\left(s^2 + 33{,}18s + 3{,}575e5\right)\left(s^2 + 38{,}03s + 1{,}459e6\right)}{\left(s^2 + 14{,}47s + 9383\right)\left(s^2 + 38{,}99s + 1{,}827e5\right)\left(s^2 + 86{,}64s + 9{,}581e5\right)} \tag{2}$$

$$H_{YX}(s) = 0{,}011018 \frac{\left(s^2 + 19{,}79s + 2{,}572e4\right)\left(s^2 + 20{,}76s + 3{,}163e5\right)\left(s^2 + 68{,}07s + 1{,}033e6\right)}{\left(s^2 + 2{,}692s + 9687\right)\left(s^2 + 22{,}4s + 3{,}096e5\right)\left(s^2 + 116{,}2s + 7{,}793e5\right)} \tag{3}$$

$$H_{XY}(s) = 0{,}002691 \frac{\left(s^2 + 15{,}81s + 1{,}815e4\right)\left(s^2 + 99{,}42s + 2{,}68e5\right)\left(s^2 + 33{,}79s + 1{,}403e6\right)}{\left(s^2 + 2{,}418s + 9629\right)\left(s^2 + 108{,}7s + 2{,}494e5\right)\left(s^2 + 77{,}76s + 9{,}789e5\right)} \quad (3)$$

$$H_{YY}(s) = 0{,}003595 \frac{\left(s^2 + 6{,}165s + 2{,}001e4\right)\left(s^2 + 15{,}49s + 3{,}739e5\right)\left(s^2 + 59{,}8s + 1{,}151e6\right)}{\left(s^2 + 4{,}828s + 1.073e4\right)\left(s^2 + 27{,}74s + 2{,}19e5\right)\left(s^2 + 53{,}92s + 8{,}127e5\right)} \tag{4}$$

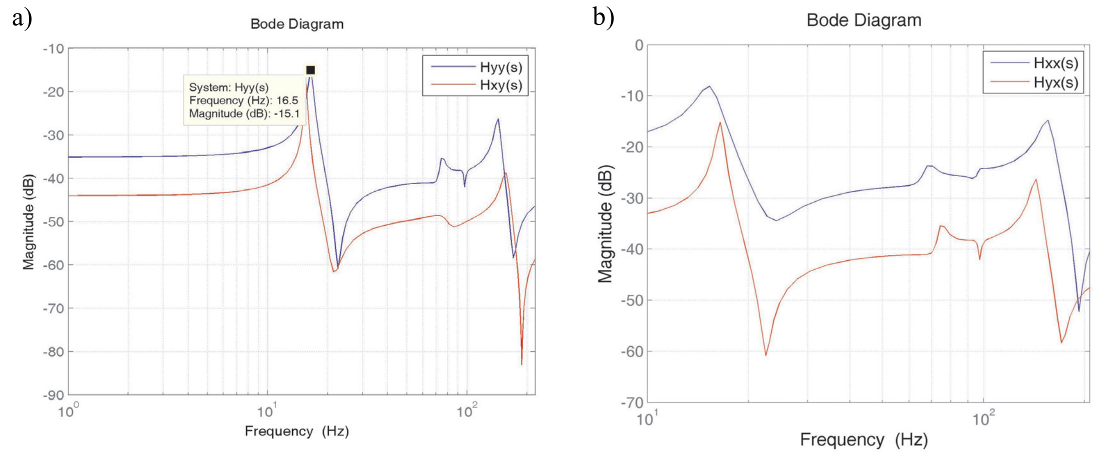

Fig. 2. The comparison the amplitude-frequency characteristics of both directions for the: a) excitation in direction *Y*; b) excitation in direction *X*

In interesting range of frequency 10–200 Hz all elementary transfer functions are of 6th order since they have three resonance peaks. All elementary transfer functions (their amplitudes) are presented in Figure 2.

As we can see in Figure 2 both models have different resonance and anti-resonance frequencies. It is correct, since we analyze structure vibrations in two different planes *X-Z* and *Y-Z*. Furthermore, we can notice, that excitation in one direction has much smaller influence on the vibrations in perpendicular direction. The amplitude of the transfer function H_{XY}(s) is about 10 dB lower than the amplitude of the transfer function H_{YY}(s) in considered range of frequencies. Similar situation we can note also comparing transfer functions H_{XX}(s) and H_{YX}(s). In this case the amplitude of the transfer function H_{YX}(s) is about 20 dB lower than amplitude of the transfer function H_{XX}(s). So, in equation (1) we can omit out-of-diagonal transfer functions. In such a way we have obtained the decupled system. It allows us to design appropriate local control law for transfer function H_{XX}(s) and H_{YY}(s).

3. CONTROL LAW FOR THE STRUCTURE WITH PD CONTROLLERS

TITO system may be controlled in a global way. It would be resulted in TITO controller. It is not simple to design such controller and so more the number a microprocessor calculations increase in power two. Because the vibration process is very fast it is desired to make the control laws as simple as possible.

We have designed the local control laws for subsystems represented by diagonal elements in the matrix transfer function (Eq. (1)). With help of computer simulations the closed-loop system with two classical PD controllers was investigated. One controller was used to damp the vibration in the plane *X-Z* while second – in plane *Y-Z*. High level of damping and very short transient period of the vibrations were the criteria in the design procedure of control laws. The controller parameters were chosen during computer simulation by the comparison of the time plots (responses to step, impulse and sinusoidal force excitations) and the Bode characteristics. So more the plots for closed-loop systems were compared with the plots for open-loop systems.

First, the local controller PD was designed for the transfer function H_{YY}(s). All investigations proved the best PD control law in plane *Y-Z* has the following form $PD_Y(s) = s + 0.05$. Therefore, all presented characteristics are results of the computer simulations for the closed-loop system with such control law. The step answers of the open-loop system and closed-loop system are presented in Figure 3. Please note different time scales in both plots. The closed-loop system is very fast – its control time is 0.04 s while transient period in open-loop system reaches 1.2 s.

The same results are presented in the Figure 4 for the impulse excitation. The transient period of closed-loop systems was reduced several times in comparison with the open-loop system. Both systems were also harmonically excited (opposite applied voltage to piezo-stack *u* = 100sin(2*pi*16.5*t)). To underline differences the excitation frequency equals frequency of the first resonance (f_{1Y} = 16.5 Hz – see left plot in Fig. 2). As we can notice in Figure 5 the vibration amplitude of closed-loop system was reduced almost five times in comparison with the vibration amplitude of the open-loop system.

All time plots showed significant reduction of the *Y-Z* vibrations of the structure when we have used the control subsystem with PD control law. The dynamic behavior of the systems was also check in desired frequency band. The amplitude-frequency characteristics of both open-loop and closed-loop systems are shown in Figure 6. We can notice the strong damping in the case of the closed-loop system particularly in the range of lower frequencies. Stronger damping in full desired frequency range (10–200 Hz) requires of a higher-order control law.

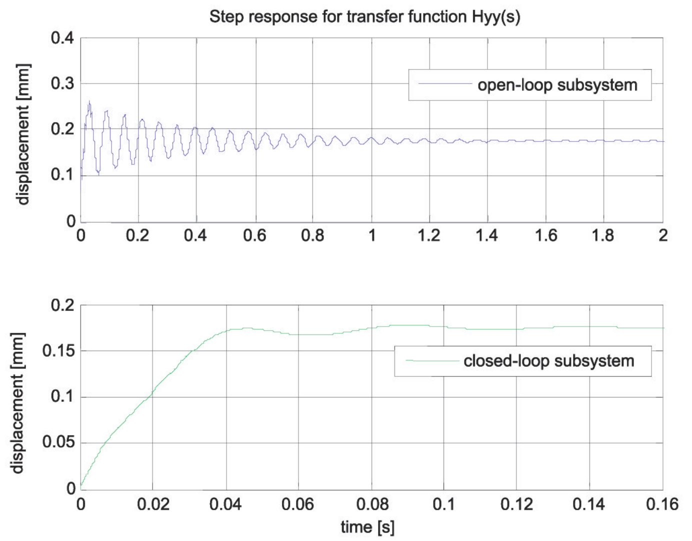

Fig. 3. The comparison of open-loop subsystem and closed-loop subsystem for step excitation in plane *Y-Z*

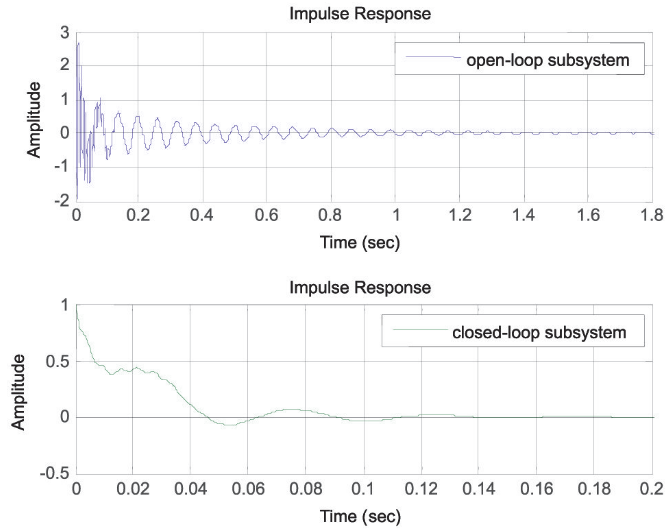

Fig. 4. The comparison of open-loop subsystem and close-loop subsystem for impulse excitation in plane *Y-Z*

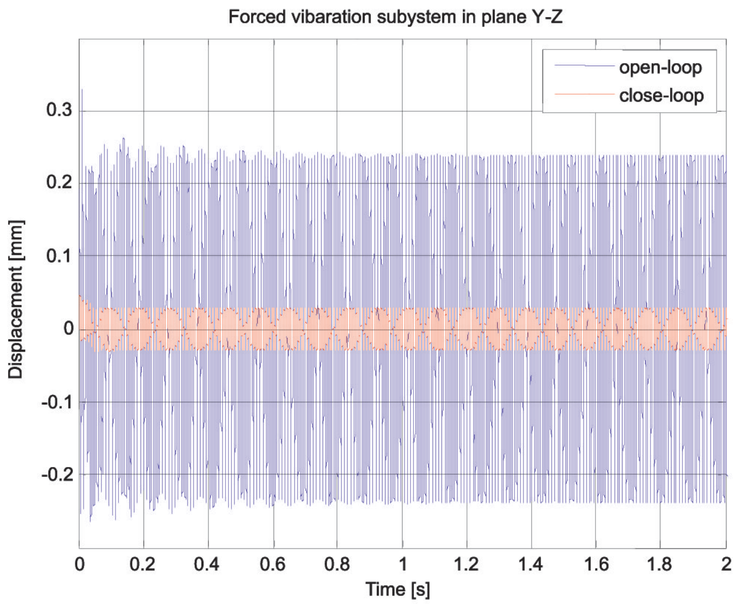

Fig. 5. The comparison of open-loop subsystem and close-loop subsystem for sinusoidal force excitation in plane *Y-Z*

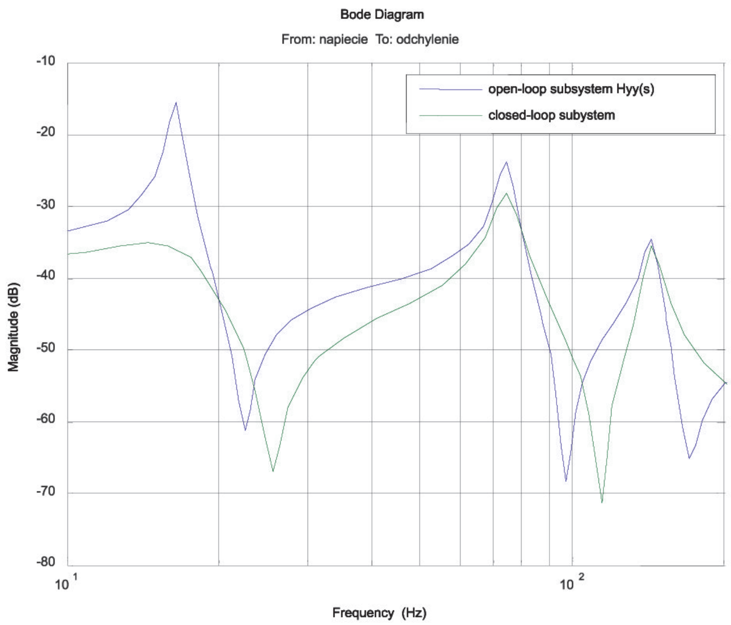

Fig. 6. The comparison of open-loop subsystem and close-loop subsystem

In the second stage similar PD control law was designed for the open-loop model described by the transfer function H_{XX}(s). Also in this case all investigations proved the best PD control law in plane *X-Z* has the following form $PD_X(s) = 0.1s + 3.2.$ Therefore, presented below characteristics result from the computer simulation of the closed-loop system with this control law. Step answers of both systems are shown in Figure 7. The closed-loop system is significantly faster in comparison the open-loop system. Transient period in open-loop system reaches up 0.4 s, while this same parameter for closed-loop system is 0.08 s. Similar results are presented in Figure 8 for the impulse excitation. In this case the transient period of closed-loop systems was reduced four times in comparison with the open-loop system. Additional both systems were also sinusoidal excited results are shown in Figure 9. This time the signal excitation was the sinusoidal force with frequency of the first resonance in direction *X* (f_{1X} = 15.3 Hz – see right plot in Figure 2, applied voltage to the piezo-stack is u = 100sin(2*pi*15.3*t)). As we can see in Figure 9 in this case the amplitude of closed-loop system was reduced six times in comparison with the vibration amplitude of the open-loop system.

Results from Figure 7 to Figure 9 show correct acting the closed-loop system in plane *X-Z*. However we do not know, how this control law will damp the vibrations the structure in consider frequency range 10–200 Hz. Bode plot in Figure 10 is an answer to the question.

Amplitude characteristics in consider frequency range confirm the appearing of the additional damp in the subsystem. Evidently it is significant in vicinity of the two lowest resonances where the amplitude of closed-loop subsystem decreased about 10 dB.

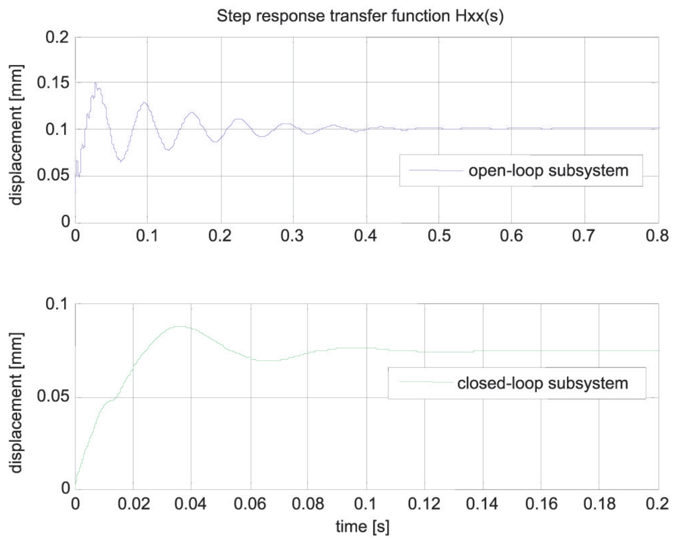

Fig. 7. The step responses of open-loop subsystem and close-loop subsystem for the excitation in plane *X-Z*

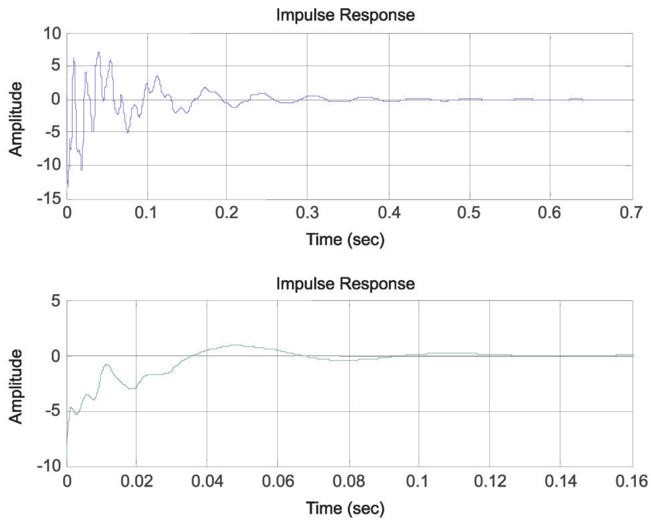

Fig. 8. The impulse responses of open-loop subsystem and close-loop subsystem for the excitation in plane *X-Z*

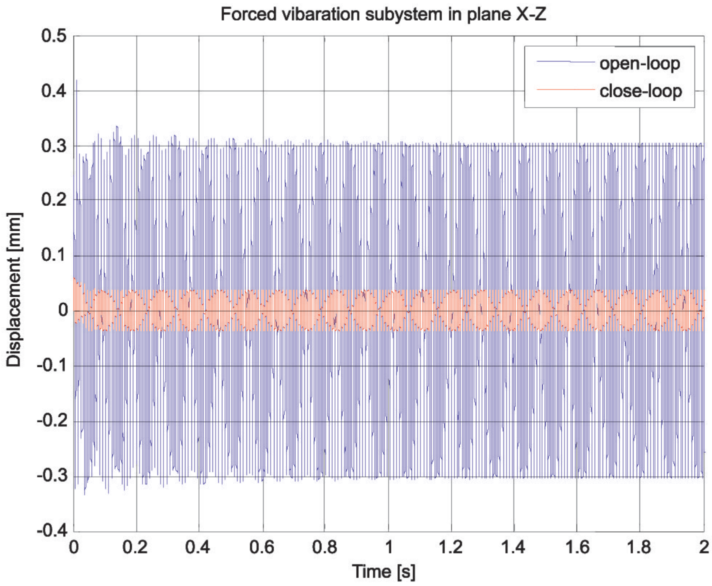

Fig. 9. The comparison of open-loop subsystem and close-loop subsystem responses to the sinusoidal force excitation in plane *X-Z*

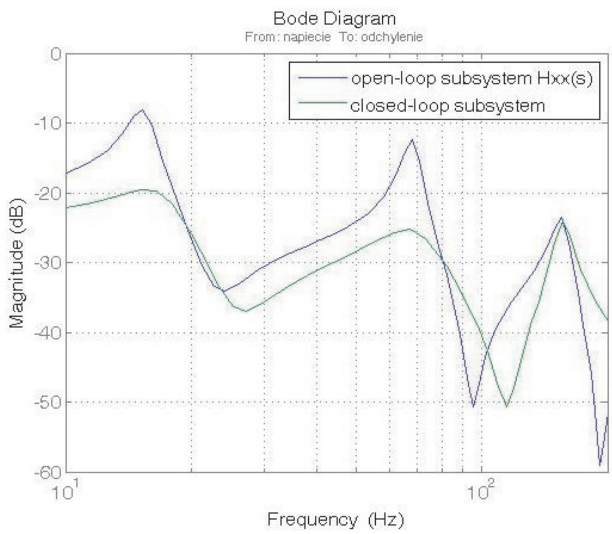

Fig. 10. The comparison of open-loop subsystem and closed-loop subsystem

4. EXPERIMENTAL INVESTIGATIONS

In this chapter the experimental results obtained for open-loop system and closed-loop system with local control laws are discussed. The laboratory stand consists of mechanical bar structure, two eddy-current sensors, two piezo-stacks, two voltage amplifiers and digital signal processor DSP which play the role of the controller. As it was mentioned earlier the local control laws have the following form:

$$u_X(t) = 3.2\left(e(t) + 0.1\frac{de(t)}{dt}\right)$$

and

$$u_Y(t) = 0.05\left(e(t) + \frac{de(t)}{dt}\right).$$

Such control law was implemented at the processor DSP. The block diagram of the whole control system is shown in Figure 11.

First, the structure vibrations were excited by the piezo-stacks with the frequency of the *Y-Z* plane first resonance in (applied voltage: u = 1.2sin(2**pi**16.5**t*) [V]), next, the structure was exited with the frequency of the *X-Z* plane first resonance (applied voltage: u = 1.2sin(2**pi**15.3**t*) [V]). The voltage signals from both eddy-current sensors were recorded and recalculated to obtain the displacements at the free end plane of the structure. During these experiments the piezo-stacks working as a vibration generators were switched off after several seconds (after about 15 s) to obtain transient response of open loop-systems (upper plots in Fig. 12 and Fig. 13). In some experiments the piezo-stacks were switch off as a generator and immediately were

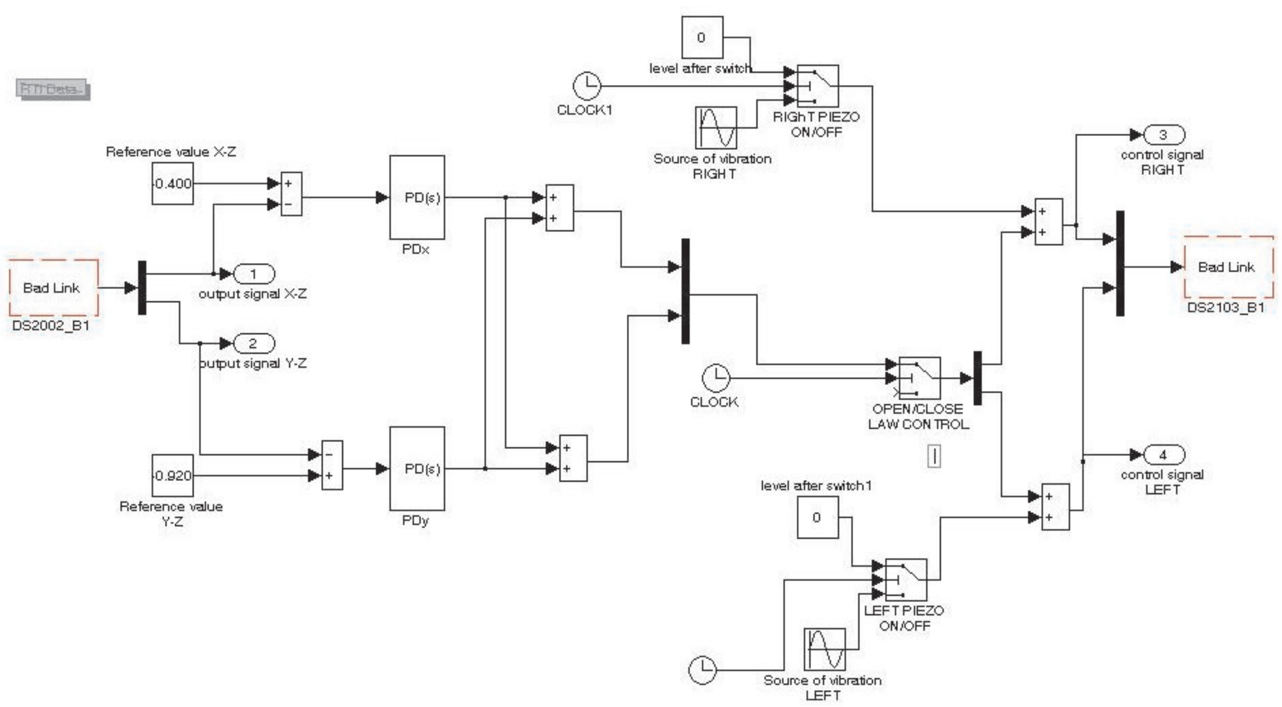

Fig. 11. Block diagram of the control system

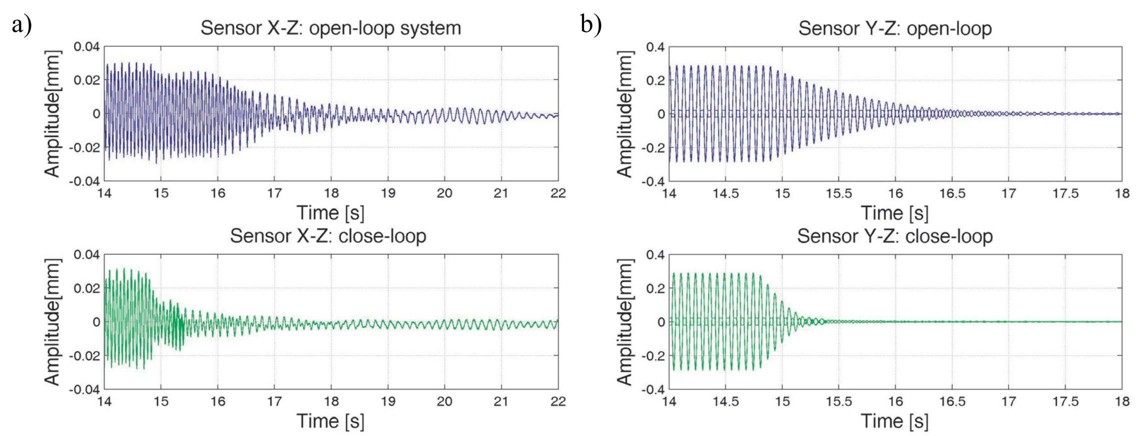

Fig. 12. Transient vibrations of open-loop system and closed-loop system in plane *Y-Z*: a) right sensor (sensor plane *X-Z*); b) front sensor (sensor plane *Y-Z*)

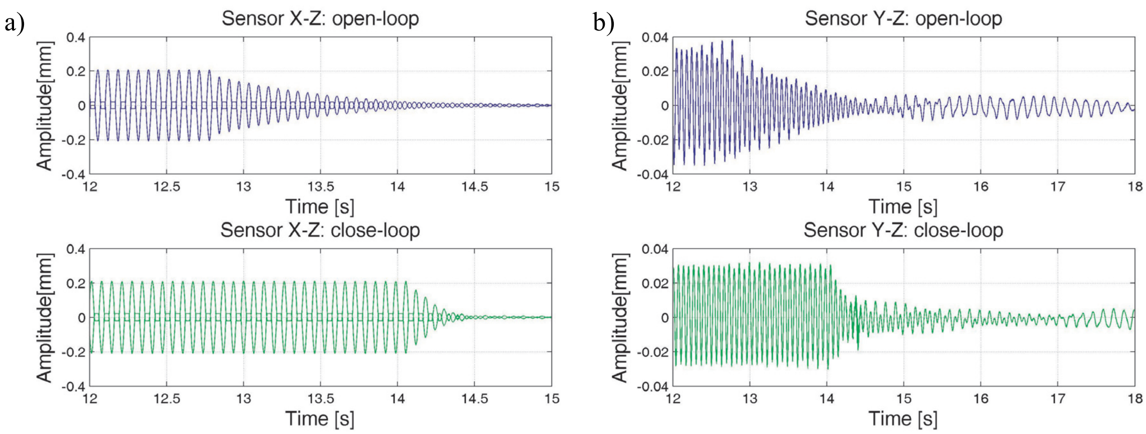

Fig. 13. Transient vibrations of open-loop system and closed-loop system in plane *X-Z*: a) right sensor (sensor plane *X-Z*; b) front sensor (sensor plane *Y-Z*)

switch on as a actuators of closed-loop systems to show the transient period of the closed-loop system (lower plots in Fig. 12 and Fig. 13). The results of such experiments are presented in both directions for the excitation in directions *X* (Fig. 12) and for excitation *Y* (Fig. 13). Please notice again the different scales of the amplitudes in presented plots.

Experimental results have proved that local PD controllers were designed correctly. In both cases the transient vibrations of the closed-loop system significantly faster disappear in comparison with the open-loop system. It was appeared such decoupled closed-loop system with simple controllers is sufficient to have the excellent active damping of the whole structure vibration.

5. CONCLUSIONS

In the paper we have described the process design of 2D control laws for the vibration control of 3D mechanical system. Results of dynamic process identification have showed that the whole system is really decoupled for chosen forms of the excitation. Such excitation approach was used in control loops. *Ipso facto* process of design control laws was reduced to the design of two local controllers each of them for one diagonal transfer function in TITO model of the full system. In such solutions the local control laws have simple form, typical for PD controllers. Control laws parameters were obtained in computer simulations procedure. Next dynamic behavior of open-loop system and closed-loop system was analyzed with help of standard excitations like step function, impulse function and sinusoidal function. All dynamics characteristics show the advantage of the closed-loop system in comparison to the open-loop system. For example transient period time was reduced several times.

Next, the experimental stand was designed and obtained control laws were implemented to processor DSP. Experimental results completely confirmed the computer simulations and they show proper operation of the closed-loop system with two local PD controllers. The active system significantly increases the damp level in structure vibrations. The transient period is much shorter than in case of the open-loop system.

All carried out investigations and obtained results have proved that designed 2D active vibration control system perfectly control the vibrations of 3D flexible bar structure, especially for the first mode of vibration.

References

Bishop R.H., Dorf R.C. 2005, *Modern Control Systems*. 10th ed., Pearson, Prentice-Hall.

dSpace GmbH 2005, *dSpace Solutions for Control*. Paderborn.

Gosiewski Z., Koszewnik A. 2010, *The influence of the piezoelements placement on the active vibration damping of active truss.* Archives of Control Sciences, vol. 20, No.1, pp. 77–88.

Hansen C., Snyder S. 1997, *Active Control of Noise and Vibration*. Champan & Hall.

Kelly S. Graham 2007, *Advanced Vibration Analysis*. CRC Press, Taylor & Francis.

Koszewnik A., Gosiewski Z. 2011, *Frequency Domain Identification of the Active 3D Mechanical Structure for the Vibration Control System*. Paper registered for MSM.

Ogata K. 2002, *Modern Control Engineering*. 4th ed., Prentice-Hall.

Pietrzakowski M. 2004, *Active damping of transverse vibration using distributed piezoelectric elements*. Mechanika.

Pietrzko S.J., Mao Q. 2009, *Reduction of structural sound radiation and vibration using shunt piezoelectric materials*. Solid State Phenomena, 147–149.

Premount A. 2002, *Vibration Control of Active Structures. An Introduction*. 2nd ed., Kluwer Academic Publisher.

Szolc T., Pochanke A. 2010, *Transient and steady-state coupled electro-mechanical vibration analysis of the micro-drive system*. XXIV Symposium Vibrations in Physical Systems, Poznan – Bedlewo.

Tewari A. 2005, *Modern Control Design with Matlab and Simulink.* John Wiley & Sons, Inc., New Jersey.

The MathWorks Inc. 2010, *Control System Toolbox*.