JOURNAL OF THEORETICAL
AND APPLIED MECHANICS
49, 4, pp. 1169-1181, Warsaw 2011

LABORATORY STAND FOR TESTING SELF-POWERED VIBRATION REDUCTION SYSTEMS

Bogdan Sapiński, Jacek Snamina, Łukasz Jastrzębski

University of Science and Technology, Department of Process Control, Kraków, Poland

e-mail: deep@agh.edu.pl; snamina@agh.edu.pl; lukasz.jastrzebski83@gmail.com

Antoni Staśkiewicz

Cracow University of Technology, Department of Mechanics, Kraków, Poland

e-mail: as@mech.pk.edu.pl

The study describes the laboratory stand for testing semi-active vibration reduction systems comprising a magnetorheological (MR) damper, powered from an electromagnetic generator. The design objectives, mechanical structure and parameters of the test stand are discussed. Dynamic parameters of the stand are estimated basing on numerical simulation data. The key elements of the test stand are presented: the vibration reduction system, the vibration generation system and the data acquisition system. Selected results of functional tests are provided.

Key words: MR damper, electromagnetic generator, vibration reduction

1. Introduction

Various types of vibration reduction systems are employed to mitigate for the effects of structural damage of buildings and structures due to vibrations. These vibration reduction systems utilize the dissipation of energy of excitations, such that its remaining portion should be transmitted by the structure.

In the case of tall structures, such as high buildings, vibration reduction systems might be categorized depending on the applied damping devices and depending on the manner they are mounted in the building. According to the first classification, we get passive, semi-active and active systems. The other categorization gives us base-insulating and stiffness-control (bracing systems). Semi-active systems use various types of damping devices providing for controllable damping force.

The research on potential applications of MR dampers in semi-active seismic protection systems began in the 1990s. The first results obtained for base-insulating sensors were reported in (Spencer *et al.*, 1996; Dyke and Spencer, 1996; Dyke *et al.*, 1996), focusing on modeling, simulation and laboratory testing of semi-active seismic protection systems equipped with MR dampers providing the damping force of the order of several kN. Further results, obtained for vibration reduction systems equipped with MR dampers generating considerably larger forces (of the order of tens of tons), were reported in the work by Yang (2001). Similar tests were done for MR bracing systems (Hiemenez and Werely, 1999; Hiemenez *et al.*, 2000). Control algorithms in MR damper systems were investigated in Yosioka *et al.* (2002). Control systems explored in those studies were feedback systems comprising a sensor, a controller, and an external source of energy to power the MR damper.

The work by Cho *et al.* (2005) gives the conceptual design of the base insulating system where the motion of the structure is associated with the MR damper force. In this system, the sensor, controller and a current driver are replaced by an electromagnetic generator wherein the motion of the structure is "converted" into the voltage signal inducing the current flow in the MR damper coil. The current activates the magnetic field that controls the MR damper force. In this approach, some portion of energy of the vibrating plant is utilized for control of the damping force. Experimental data for the system comprising an MR damper of the RD-1097-1 type (http://www.lord.com) are summarized in the work by Cho *et al.* (2007).

This study briefly describes the laboratory stand for testing the self-powered vibration reduction system at the Laboratory of Adaptronics of the Department of Process Control in AGH-UST. The design objectives, mechanical structure and parameters of the test stand are discussed. Dynamic parameters of the stand are estimated basing on numerical simulation data. Results of functional tests are provided, which seem to confirm the adequacy of the system design.

2. Design objectives, mechanical structure and basic parameters of the stand

The test stand is designed to imitate a simple model of the first floor in the structure, acting as the insulating base protecting the entire structure from the effects of ground movements. This concept determines the series structure of the test laboratory stand (Fig. 1), and the vibration reduction system is placed parallel to the spring representing the elastic capabilities of the insulating base. The mass of the platform represents the mass of the building. The applied shaker allows for reconstructing the ground motions in accordance with the prescribed motion profiles. The movements of the platform emulate the motion of the considered building or structure.

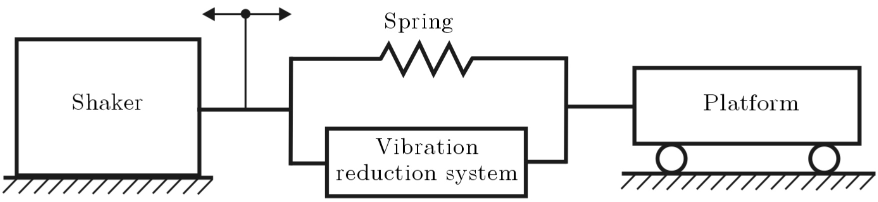

Fig. 1. Simplified mechanical structure of the stand

The selection of parameters of the laboratory stand consists in finding the mass of the platform and the spring stiffness such that the engineered vibration reduction system should be effectively used. The following parameters of the vibration reduction system are taken for calculations:

- amplitude of relative velocity of the generator components at which it generates the voltage required to endure the effective operation of the MR damper; basing on the predicated and experimental data this amplitude is taken as 0.2 m/s;
- rms value of the damper force for the assumed amplitude 0.2 m/s; the rms force 460 N is read off the experimental characteristics (Sapiński, 2010).

The stand should enable the testing of natural and excited vibration in the frequency spectrum as broad as possible. The parameters of the stand were chosen such that the dimensionless damping ratio should be 0.5. Then the motion of the platform becomes oscillatory damped motion and the amplitude-frequency characteristic will have a maximum.

Thus, the selected parameters can be treated as the baseline for further testing. Additionally, taking into account the calculated forces and parameters of the shaker, the amplitude of the excitation executed by the shaker is taken as 0.004 m. Other parameters of the stand are:

- natural vibration frequency under which the generator generates voltage required to ensure the effective operation of the MR damper (taking into account the approximate displacement amplitude) – 5 Hz;
- mass of the platform at which the dimensionless damping ratio reaches the predetermined value, for the assumed damper force and calculated natural frequency of vibration; mass of the platform – 100 kg;

- stiffness ratio of the spring based on the calculated mass of the platform and natural frequency; stiffness ratio – 10^5 N/m.

The assumed structure and parameters were used as the staring point for the further stage of the research program.

3. Numerical simulation of the stand

Numerical simulations were performed to evaluate the dynamic parameters of the stand. Schematic diagrams of the mechanical and electrical sub-systems are shown in Figs, 2 and 3 (Snamina and Sapiński, 2011). The diagram of the mechanical sub-system (Fig. 2) presents its main components as well as the coordinate of the body position (platform) x, kinematic excitation z, the generator force F_g, the MR damper force F, the spring force F_s. The model of the electric sub-system comprises the connected coils of the generator and MR damper. R_g and L_g denote the resistance and inductance of the generator coil, whilst R_g and L_g are the resistance and inductance of the MR damper control coil, e denotes electromotive force and i – current in the generator-damper circuit.

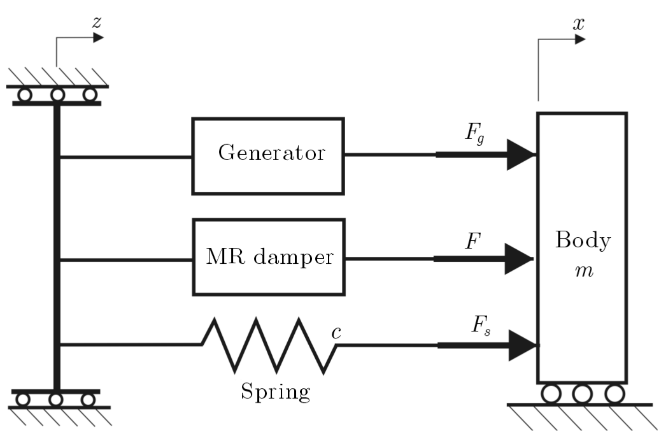

Fig. 2. Schematic diagram of the mechanical sub-system

The force generated by the MR damper is given by the formula (Guo *et al.*, 2006; Kwok *et al.*, 2006; Maślanka *et al.*, 2007)

$$\begin{aligned} F = (c_1|i| + c_2)\tanh\Big[\beta\Big(\Big(\frac{dz}{dt} - \frac{dx}{dt}\Big) + p_1(z-x)\Big)\Big] \\ + (c_3|i| + c_4)\Big(\Big(\frac{dz}{dt} - \frac{dx}{dt}\Big) + p_2(z-x)\Big) \end{aligned} \tag{3.1}$$

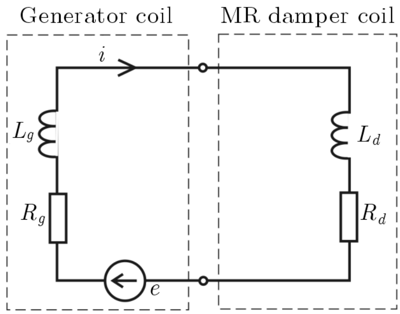

Fig. 3. Schematic diagram of the electric sub-system

where: c_1, c_2, c_3, c_4 are constants in the MR damper model, and β, p_1, p_2 are scaling parameters.

Admitting the state variables: x – body coordinate, w – velocity of the body, i – current in the generator-MR damper circuit, the state equations can be written in the form (Snamina and Sapiński, 2011)

$$\begin{aligned}
\frac{dx}{dt} &= w \\
\frac{dw}{dt} &= \frac{1}{m}\Big\{c(z-x)+\kappa i+(c_1|i|+c_2)\tanh\Big[\beta\Big(\Big(\frac{dz}{dt}-w\Big)+p_1(z-x)\Big)\Big] \\
&\quad +(c_3|i|+c_4)\Big(\Big(\frac{dz}{dt}-w\Big)+p_2(z-x)\Big)\Big\} \\
\frac{di}{dt} &= \frac{1}{L_g+L_d}\Big[\kappa\Big(\frac{dz}{dt}-w\Big)-(R_g+R_d)i\Big]
\end{aligned} \tag{3.2}$$

The calculations were performed for the following parameters of the system: $m = 100\,\mathrm{kg}$, $c = 105\,\mathrm{N/m}$, $R_g = 0.4\,\Omega$, $L_g = 7.5\,\mathrm{mH}$, $R_d = 5\,\Omega$, $L_d = 100\,\mathrm{mH}$, $\kappa = 24\,\mathrm{N/A}$. For those parameters, the natural frequency of the system is equal to 5 Hz. The parameters of the model used in the damper RD-1005-3, estimated on the basis of former laboratory tests, are: $c_1 = 800\,\mathrm{N/A}$, $c_2 = 40\,\mathrm{N}$, $c_3 = 3745\,\mathrm{Ns/Am}$, $c_4 = 322\,\mathrm{Ns/m}$. A predicted and measured damper force versus velocity is shown in Fig. 4.

The simulation results for the applied kinematic excitations of frequency 4.5 Hz and amplitude 3.5 mm are shown in Figs. 5 and 6. Figure 5 presents the time histories of voltage and current in the generator-MR damper circuit. The phase shift of current with respect to voltage is about $35°$. This parameter is of great significance to ensure the performance of the system.

The plot of damper force (Fig. 6b) reveals fast changes in those times when the relative velocity is close to zero (see Fig. 5b). That is the consequence of the adopted model of the MR damper.

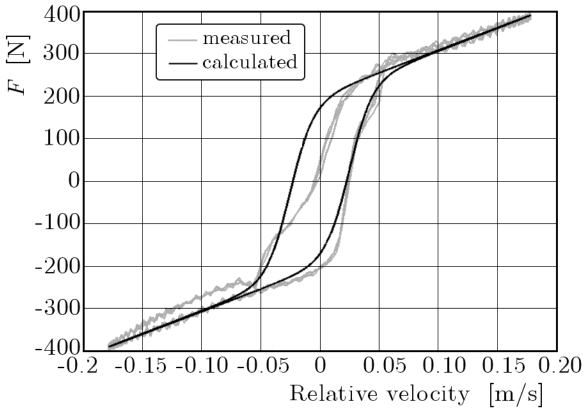

Fig. 4. Damper force F versus relative velocity $\dot{z} - \dot{x}$

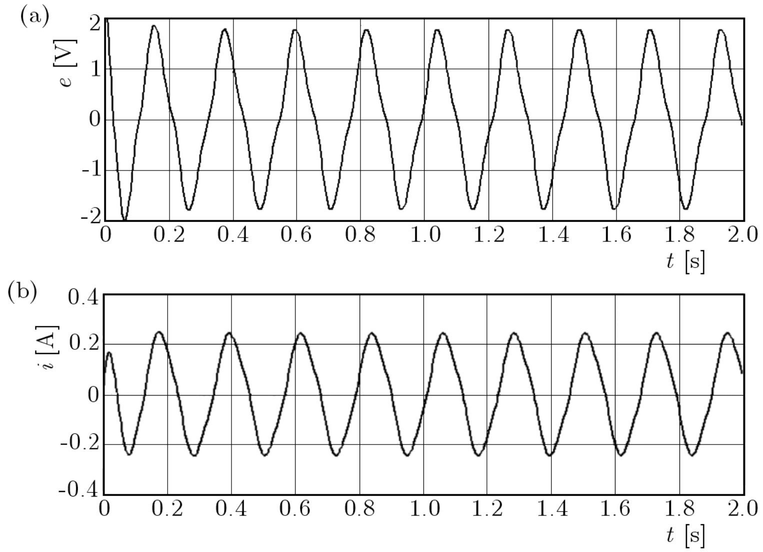

Fig. 5. Time histories of: (a) electromotive force e, (b) current i in the generator-MR damper circuit

4. Description of the stand

The design of the test stand (see Fig. 1) should ensure the displacement of its all mobile components in one direction. That requires precise guiding systems to move the platform and the vibration reduction system. The base is fixed to the ground with a rigid frame, made of steel C-profiles C50. The plates bolted to the upper section of the frame uphold the linear guiding systems. The guides

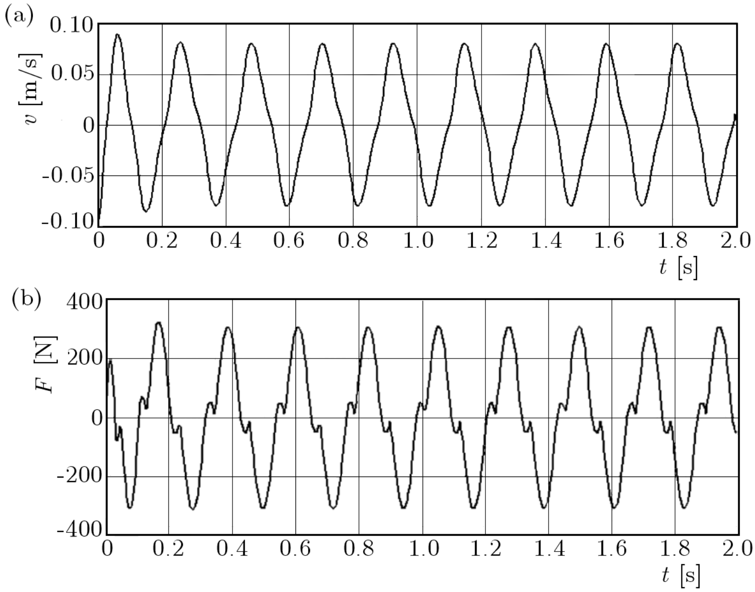

Fig. 6. Time histories of: (a) relative velocity $v = \dot{x} - \dot{z}$, (b) damper force F

and linear bearings allow the displacement of the vibration reduction system and of the platform. Mobile elements of the structure are connected in series with joints, so as to alleviate for irregularities of the guides position. The test stand is shown in Fig. 7.

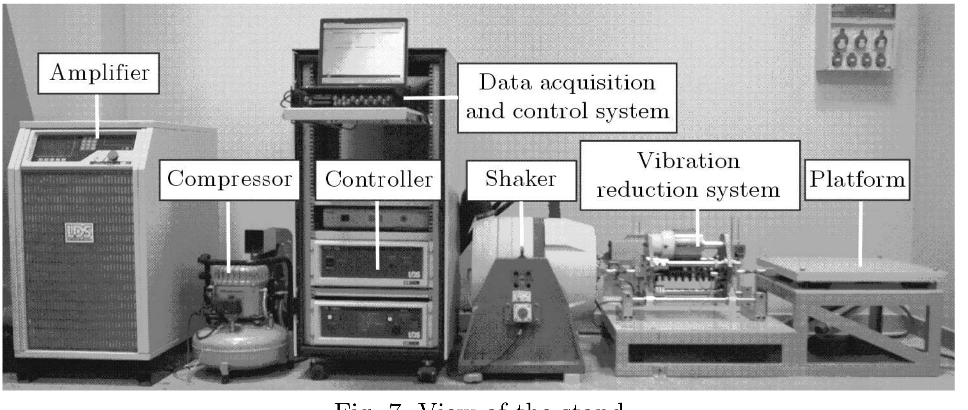

Fig. 7. View of the stand

The stand incorporates an electromagnetic shaker equipped with a controller, amplifier and a compressor, vibration reduction system and a spring mounted parallel to it, platform, data acquisition and the control system. On one end the vibration reduction system is attached to the shaker, on the other- to the mobile platform. The main function of the shaker is to generate the linear displacement so as to induce motion of the vibration reduction system and of the platform. The platform comprises three boards arranged horizontally to which trolleys are attached that slide along the guides, thus enabling the platform movement along the horizontal axis.

The vibration reduction system incorporates a MR damper of the RD-1005-3 type manufactured by Lord Corporation and an electromagnetic generator (Sapiński, 2010) whose ends are fixed between two base boards, and a spring is provided between the two boards.

The schematic diagram of the measuring and control system is shown in Fig. 8. Actually, it comprises two systems, one for generating mechanical vibration, the other used for acquisition of measurement data.

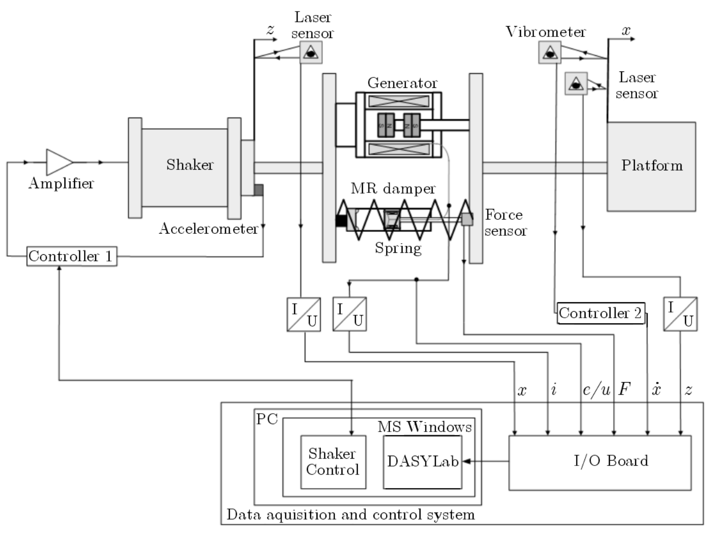

Fig. 8. Schematic diagram of the measurement and control system of the stand

The vibration generation system incorporates a shaker V780 of LDS, a power amplifier and a controller connected to a computer via a USB port. The shaker is controlled using the feedback signal from the piezoelectric accelerometer 357B33 of PCB Piezotronics. The Dactron Shaker Control software allows the displacement pattern to be preset, to imitate the seismic movement profile.

The data acquisition system comprises the hardware (laser and piezoelectric sensors with conditioners, a computer with an I/O board National Instruments DAQPad-6052E connected via the FireWire port) and software elements (DASYLab version 10.0).

The parameters that can be registered include: displacement z (applied excitation), displacement of the platform x, velocity of the platform $\dot{x}$, damper force F, terminal voltage u and current i in the generator-MR damper circuit. Displacements are measured with laser sensors FT 50 RLA of SENSOPART; velocity measurements are taken with the laser vibrometer OFV-505 of Polytec with a PFV 5000 controller. The damper force is measured using the piezoelectric sensor 208-C03 of PCB Piezotronics connected with a signal conditioner 480B21. Current in the control coil is measured with a current-voltage converter, incorporating a reference resistor $(0.1\,\Omega)$ and an operational amplifier. Thus, the measured quantities are converted into voltage signals in the range $(-10, +10)$ V and fed to the I/O board.

5. Testing of the stand

The purpose of the testing program was to check the performance of the test stand and capabilities. Selected results of functional tests are given in terms of the transmissibility coefficient $(T_{xz}(f) = X(f)/Z(f))$ (Fig. 9) and time patterns of the measured quantities: $z(t)$, $x(t)$, $e(t)$, $u(t)$, $i(t)$, $F(t)$ under the applied excitation z with the amplitude 3.5 mm and frequency 4.5 Hz (Figs. 10-12).

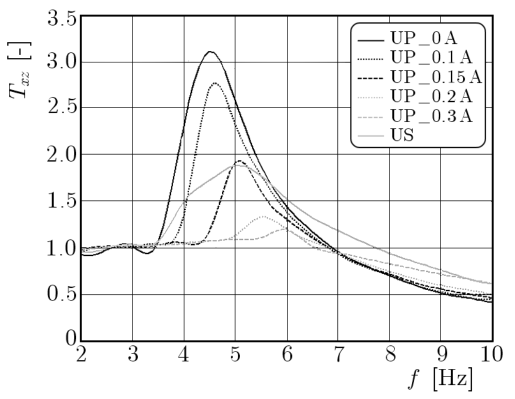

Fig. 9. Transmissibility coefficient T_{xz}

The plots of the transmissibility coefficient of a passive vibration reduction system UP (for the current level in the control coil: 0, 0.1, 0.15, 0.2, 0.3 A) and of a self-powered system US reveal that:

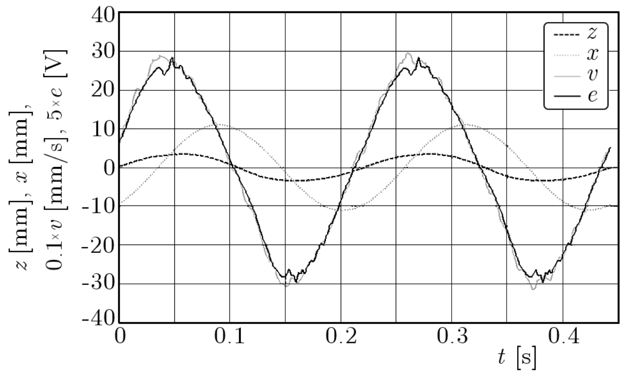

Fig. 10. Time histories of the electromotive force e, displacement of the platform x and relative velocity v under a sine excitation z with the amplitude 3.5 mm and frequency 4.5 Hz

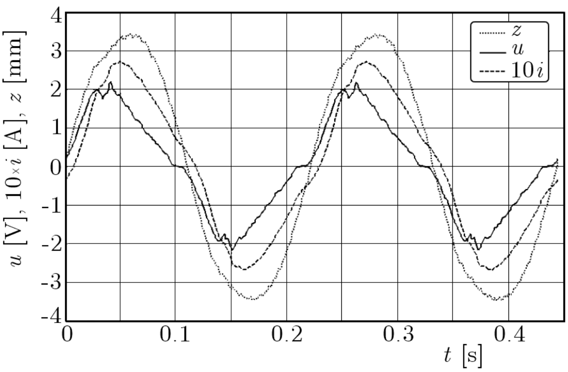

Fig. 11. Time histories of the voltage u and current i in the generator-MR damper electric circuit under a sine excitation z with the amplitude 3.5 mm and frequency 4.5 Hz

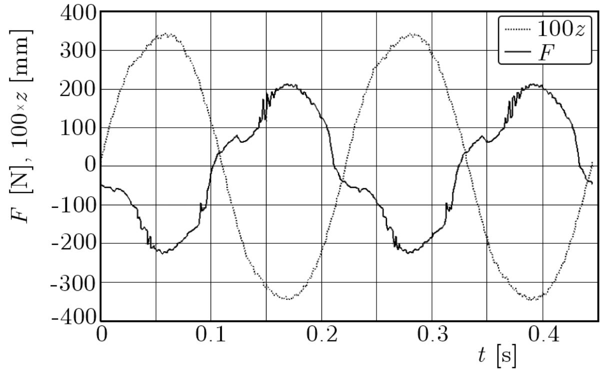

Fig. 12. Time histories of the damper force F under a sine excitation z with the amplitude 3.5 mm and frequency 4.5 Hz

- the resonance frequency for the UP_0 A system equals 4.5 Hz,
- an increase in the current level in the control coil in the passive system leads to an increase of the resonance frequency and to reduction of the resonance gain,
- the resonance frequency for the self-powered system US is 5 Hz,
- the transmissibility coefficient in the US system for the resonance frequency is comparable to its value obtained for the passive system UP_0.15 A.

Figure 10 shows time histories of the electromotive force e registered for the system UP_0 A. It was observed that the electromotive force e was proportional to the relative velocity $v = \dot{x} - \dot{z}$ and remained in phase with it. Figure 11 presents time histories of the voltage u and current i in the generator-MR damper electric circuit for the self-powered system US. The phase shift between these two quantities was dependent on the frequency of excitation. Similarly, Fig. 12 shows time histories of the damper force F in the US system. It was observed that the increase in the excitation frequency led to an increase of the current in the control coil, bringing forth the increase of the damper force.

6. Summary

The study briefly describes the laboratory stand for testing the semi-active vibration reduction systems. The design objectives, mechanical structure and parameters of the test rig are discussed. Dynamic parameters of the stand are estimated basing on numerical simulation data. The key elements of the stand are presented: vibration reduction system, the vibration generation system and the data acquisition system. Selected results of functional testing are provided.

Functional tests evidenced good performance of all sub-assemblies, confirmed the adequacy predictions of the variability range of key mechanical and electric parameters. The measurement data are in agreement with the calculation results obtained at the stage of design. Simulated time patterns of the generator terminal voltage u and current i in the generator-MR damper electric circuit (Fig. 5) are in line with the registered measurement data (Fig. 11). The shape of the simulated time histories of the damper force (Fig. 6) agrees well with those obtained during the functional testing (Fig. 12) but the value of the damper force obtained from calculations is larger. It is probably associated with the friction force present in the linear guiding systems.

Acknowledgement
The study is a part of the research project No N501 366934.

References

1. Cho S.W., Jung H.J., Lee I.W., 2005, Smart passive system based on a magnetorheological dampers, *Smart Materials and Structures*, **1**, 707-714
2. Cho S.W., Jung H.J., Lee I.W., 2007, Feasibility study of smart passive control system equipped with electromagnetic induction device, *Smart Materials and Structures*, **16**, 2323-2329
3. Dyke S.J., Spencer B.F. Jr., 1996, An experimental study of MR dampers for seismic protection, *Smart Materials and Structures* (special issue on large civil structures)
4. Dyke S.J., Spencer B.F., Sain M.K., Carlson J.D., 1996, Modeling and control of magnetorheological dampers for seismic response reduction, *Smart Materials and Structures*, **5**, 565-575
5. Guo S., Yang S., Pan C., 2006, Dynamic modeling of magnetorheological damper behaviours, *Journal of Intelligent Materials Systems and Structures*, **17**, 1, 3-14
6. Hiemenez G.J., Choi Y., Werely N.M., 2000, Seismic control of civil engineering structures utilizing semi-active MR bracing systems, *Smart Structures and Materials: Smart Systems for Bridges*, 217-228
7. Hiemenez G.J., Werely N.M., 1999, Seismic response of civil structures utilizing semi-active MR and ER bracing systems, *Proceedings of the 7th International Conference Electrorhelogical Fluids and Magnetorheological Suspensions*
8. Kwok N.M., Ha Q.P., Nguyen T.H., Samali B., 2006, A novel hysteretic model for magnetorheological fluid dampers and parameter identification Rusing particie swarm optimization, *Sensors and Actuators A*, **132**, 441-451
9. Maślanka M., Sapiński B., Snamina J., 2007, Experimental study of vibration control of a cable with an attache MR damper, *Journal of Theoretical and Applied Mechanics*, **45**, 893-917
10. Sapiński B., 2010, Vibration power generator for a linear MR damper, *Smart Materials and Structures*, **19**, 1050-1062
11. Snamina J., Sapiński B., 2011, Energy balance In self-powered MR damper-based vibration reduction system, *Bulletin of the Polish Academy of Science. Technical Sciences*, **59**, 1, 75-80

12. Spencer B.F. Jr., Dyke S.J., Sain M.K., Carlson J.D., 1996, Phenomenological model of a magnetorheological damper, *ASCE Journal of Engineering Mechanics*, USA

13. Yang G., 2001, Large-scale magnetorheological dampers for vibration mitigation: modeling, testing and control, Doctoral Dissertation, The University of Notre Dame

14. Yoshioka H., Ramallo J.C. Spencer B.F. Jr., 2002, Smart base isolation strategies employing magnetorheological dampers, *Journal of Engineering Mechanics*, **128**, 5, 540-551

15. http://www.lord.com

Stanowisko laboratoryjne do badań samozasilającego się układu redukcji drgań

Streszczenie

W artykule przedstawiono stanowisko badawcze układu redukcji drgań z tłumikiem magnetoreologicznym (MR), który jest zasilany z generatora elektromagnetycznego. Omówiono założenia projektowe i przyjęto strukturę mechaniczną stanowiska oraz wykonano obliczenia jego podstawowych parametrów. Opisano budowę wchodzących w skład stanowiska układów: redukcji drgań, wytwarzania drgań i akwizycji danych pomiarowych. Wykonano obliczenia symulacyjne układu pozwalające na oszacowanie istotnych dla działania stanowiska wielkości mechanicznych i elektrycznych. Przedstawiono wybrane wyniki testów funkcjonalnych stanowiska.

Manuscript received December 15, 2010; accepted for print February 17, 2011