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ANALYSIS OF VIBRATIONS IN A SYSTEM WITH A MAGNETORHEOLOGICAL DAMPER

ABSTRACT *The intense development of modern mechanical equipment with better and better dynamic parameters is connected with the necessity of effective damping the vibrations produced in them. Reduction of vibrations improves the precision as well as application range of this equipment and it becomes more user-friendly. It does not emit vibrations or high intensity noise, harmful to health. Application of vibration dampers with magnetorheological (MR) fluids will allow the vibrations of mechanical equipment to be damped much more effectively as compared with conventional damping systems. A great advantage of those dampers is the possibility to control the damping force simply and in a wide range. This is achieved by variation of the magnetic field in the damper with the MR fluid.*

The paper considers vibration in a system with a single degree of freedom, as shown in Fig.2. A linear electromagnetic magnetorheological damper was used for reducing the vibration amplitude. Two mathematical models of the vibrating system with such a damper were worked out, using the classical approach. In the first model the properties of the damper were described by the Bingham model (Fig.5). The nonlinearities of the damper magnetic circuit are not

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taken into account in this model, hence it is of little use for an analysis of working coditions of real vibration damping systems. In the second model of a vibrating systems (Fig.10) a damper model was used which takes into account the nonlinearity of the magnetic circuit as well as the resistance force resulting from the friction of the damper moving components against sealings.

Simulation tests of the developed damper models and of the systems for automatic vibration reduction were carried out. Selected operation states of the vibrationg system with damper were analysed. Results of the simulation tests are presented as well as conclusions concerning the developed mathematical model of the phenomena and the efficiency of vibration damping by means of controlled magnetoelectric dampers. It was found that application of a damper with a control system reduces vibrations in the system in a decisive way. The MR damper attenuates well vibrations of a system with resonance frequency. High efficiency of vibration damping was achieved which is difficult to be obtained using traditional damping systems.

1. INTRODUCTION

As up-to-date devices dramatically improve their dynamic properties, a need arises to effectively control vibrations in them [5]. By limiting vibrations greater precision of the devices is achieved thereby allowing for a larger variety of applications. The devices also become more user-friendly as they do not emit harmful vibrations or high intensity noise. The use of dampers with magnetorheological fluids (MR dampers) enables more effective vibration control in mechanical devices as is the case in conventional damping systems [4, 6]. One of the advantages of MR dampers is easy adjustment of the damping force in a wide range which is achieved by a change of the magnetic field. The structure of a linear MR damper is shown in Fig.1.

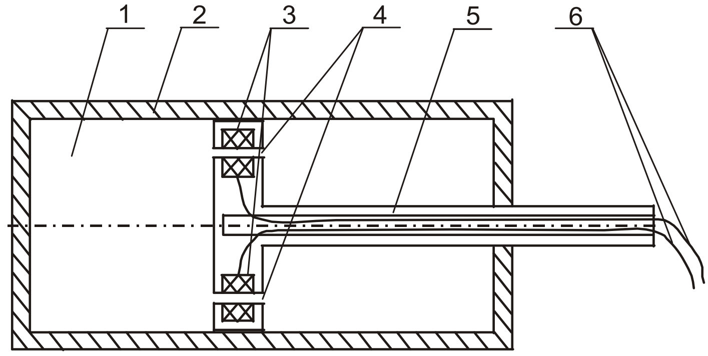


Fig.1. Cross-section of a linear MR damper;
1 – chamber with MRF, 2 – ferromagnetic cylinder, 3 – winding, 4 – working gap, 5 – piston, 6 – electric wires

Magnetorheological fluid is a colloidal suspension of magnetically polarised particles with diameters of 0.5 to 10 μm in a carrier fluid, mostly synthetic oil with a low evaporation rate or water [1]. A typical MRF contains from 20 to 80% of ferromagnetic particles, by weight. The main feature of the fluid is dramatic change of viscosity and, consequently, of shear stress upon the application of a magnetic field. The stress changes during the increase and decrease of magnetic flux density occur in microseconds. The fluids retain their properties in the temperature range from –40°C to 150°C. Relative magnetic permeability of the fluid is small, $\mu_r < 10$ [5, 6].

The paper presents the results of simulation studies into limiting vibrations by means of linear vibration dampers. A one-degree-of-freedom system is considered and the analysis is based on the classical circuit model of phenomena.

2. MATHEMATICAL MODEL OF A VIBRATING SYSTEM WITH AN MR DAMPER

A system consisting of a vibrating mass and an MR damper is considered. For convenience, the analysis of the system is based on its mathematical model. It comprises a model of a vibrating element and a model of an MR damper.

A one-degree-of-freedom vibrating system is shown in Fig.2. It consists of the mass m, the elastic constant spring k and the Newtonian damping element. The Newtonian element has the form a piston placed in a cylinder filled with viscous fluid. The resisting force generated in this component equals the product of the speed $\dot{x}$ of the piston and the damping coefficient c.

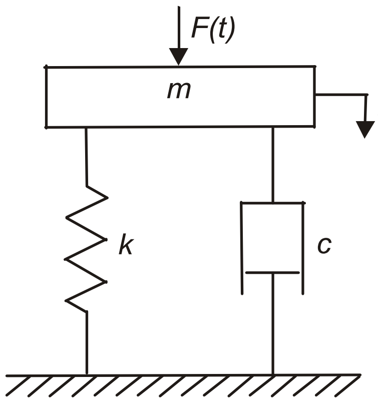


Fig.2. Model of one-degree-of-freedom system

The forces in the system in Fig.2 are described by

$$F(t) = m\ddot{x} + c\dot{x} + kx \tag{1}$$

where $F(t)$ is the external force acting on the system [2]. It is assumed that m = 1000 kg, c/m = 1 and k/m = 40. For such parameters the free vibration frequency in

$$f_o = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{2}$$

equals f_o=1 Hz. Figure 3 shows the relationship between the amplitude X of the vibrations of the mass m and the frequency f of the external force $F(t)=F_m\sin(2\pi ft)$ after solving the equation (1). Vibrations reach the highest amplitude X for the frequency f_o. The amplitude increases with the increase of the amplitude of the force F_m.

It is assumed that the behaviour of an MR damper is described by the Bingham model [1, 3]. A substitute scheme of the damper that corresponds to the model is shown in Fig.4. It consists of a parallel connection of the Newtonian element and the St Venant element. The St Venant component is shown as a slide. It does not change its location x if the external force is weaker than the force of the static friction f_0. The damping force $F_0(t)$ in the Bingham model is described by

$$F_0(t) = f_0 \operatorname{sgn}(\dot{x}) + c_0\dot{x} \tag{3}$$

where c_0 is the damping coefficient.

The force f_0 depends on the current i in the field exciting winding of the damper. The effectiveness of the damper in controlling the vibrations of the mass m has been analysed in the system shown in Fig.5. The vibrations of the mass were generated by the external force $F(t)=F_m\sin(2\pi ft)$. The damper was controlled by a P-type regulator. It influences the force f_0 by changing the exciting current.

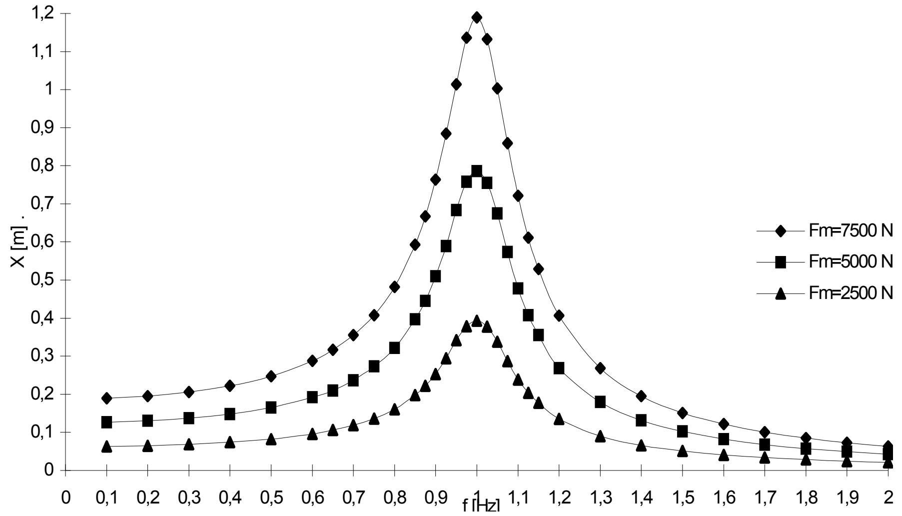


Fig.3. Relationship $X(f)$ for F_m=2500 N, 5000 N and 7500 N

The damper is to generate the resisting force $F_0(t)$. The characteristics $f_0(i)$ and the parameter c_0 of the damper are taken from literature [3]. The calculations were carried out in the Matlab Simulink environment.

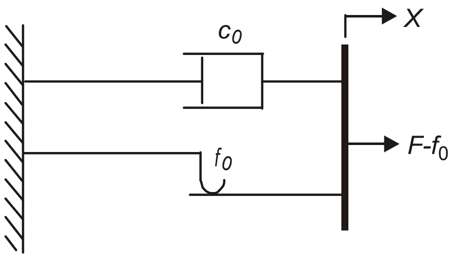


Fig.4. The Bingham model of the damper

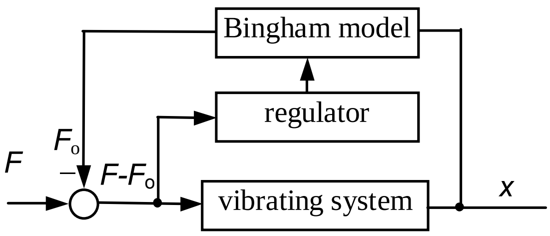


Fig.5. Vibration damping system with the Bingham model

The relationships between the amplitude X of the vibrations of the mass *m* and the frequency f in systems without a damper and with a damper are shown in Fig.6. It was assumed in the calculations that the current in the

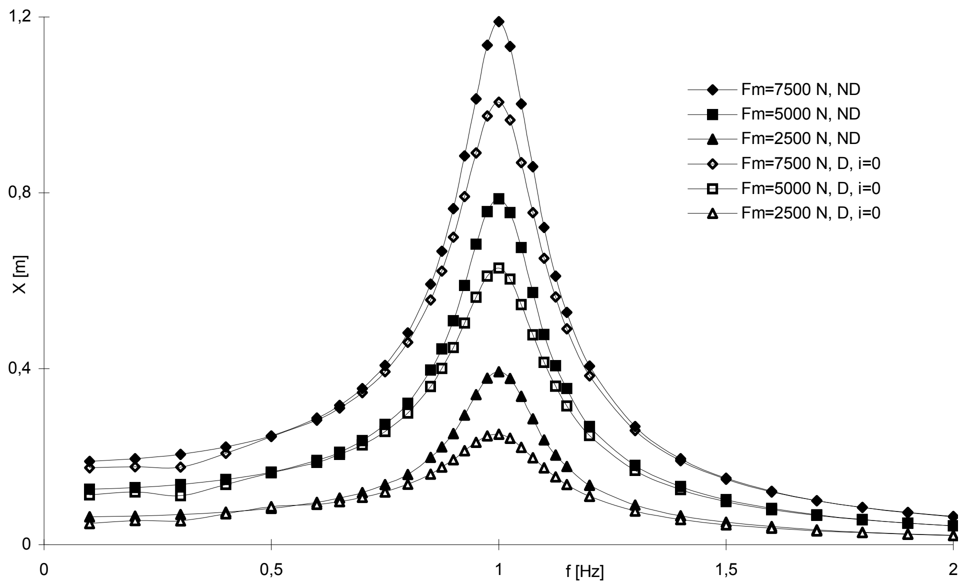


Fig.6. Chart *X*(*f*) for the vibrating system without the damper (ND) and with the damper (D) without the regulator (i=0)

windings of the damper i=0. The study was conducted for three values of the amplitude F_m. It is clearly observable that if the system is equipped with an unregulated damper, the amplitude X of vibrations decreases. A further decrease of the vibration amplitude is achieved if the system is supplemented with a regulator (Fig.7).

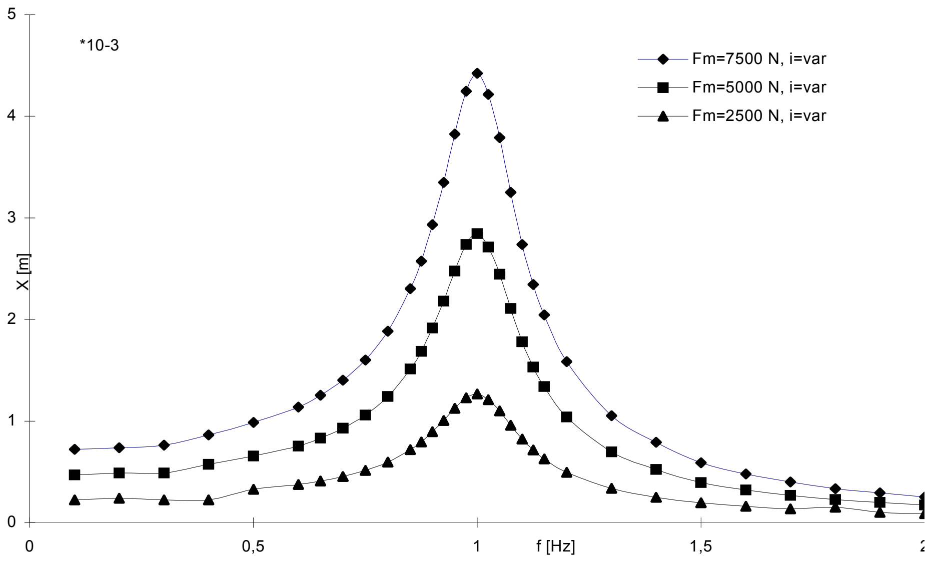
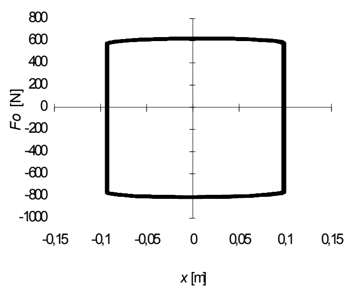


Fig.7. Chart *X*(*f*) for the vibrating system with the damper and the regulator (i=var);

Figure 8a) shows the calculated relationship of the damping force $F_0(x)$ in the system with the damper without the regulator. It appears from the figure that the damping force F_0 reaches the highest amplitude for x = 0. Fig.8b shows the distribution of the damping force $F(V)$ for this system. The force reverses the sign when the piston changes the direction of movement. In Figures 9a) and 9b) the charts $F(x)$ and $F(V)$ for a vibrating system with a damper and a regulator are shown. A significant increase of the resisting force F_0 of the damper can be seen.

a) b)



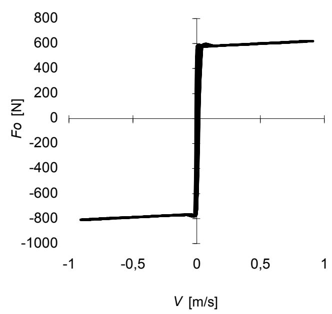


Fig.8. Characteristics a) $F_o(x)$, b) $F_o(V)$ of the system with the damper without the regulator for F_m=5000 N, *f*=1.5 Hz

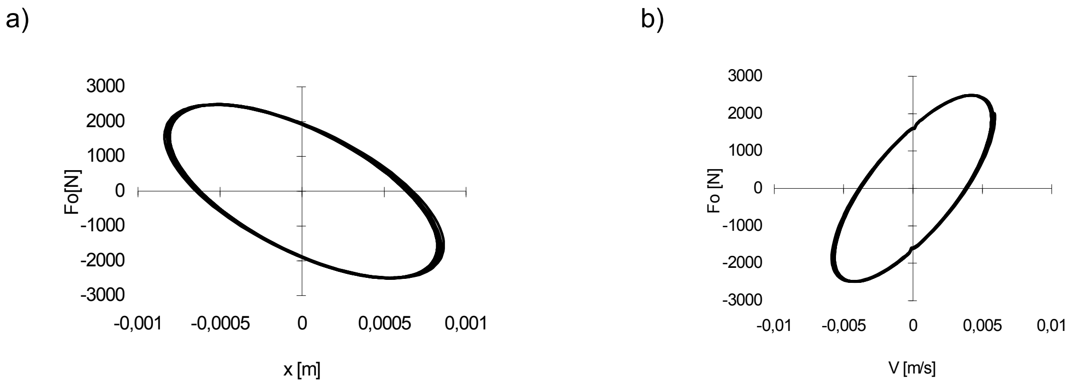


Fig.9. Characteristics a) $F_o(x)$, b) $F_o(V)$ of the system with the damper and the regulator for F_m=5000 N and f=1.5 Hz

In the Bingham model the nonlinearity of the magnetic circuit of the damper is not taken into account.

For that reason a model representing electromagnetic phenomena was used to describe the phenomena in the damper. The model is shown in [4]. The drawback of the regulation system in Fig.5 is the need to measure the difference in forces F-F_0. In a real system the difference is hard to determine and practically it is easier to measure the displacement of the vibrating mass. For reasons mentioned above an alternative vibration damping system was developed. Its structure is shown in Fig.10. The system takes into consideration the nonlinearity of the magnetic circuit and applies a modified model of mechanical phenomena in the damper. It is assumed that there are three types of resisting forces in an MR damper: a) force generated by the magnetic field F_{MR}, b) force F_P connected with the flow of the fluid when no field is present and c) friction force in the seals F_S. It is also assumed that the electric circuit of the damper is supplied from a PWM system. The voltage exciting the windings of the damper is controlled by a PI regulator in the range from 0 to 10 V. The settings of the regulator and the MR damper parameters were chosen experimentally. Vibrations in the mass m were generated by the force $F(t) = F_m\sin(2\pi f)$.

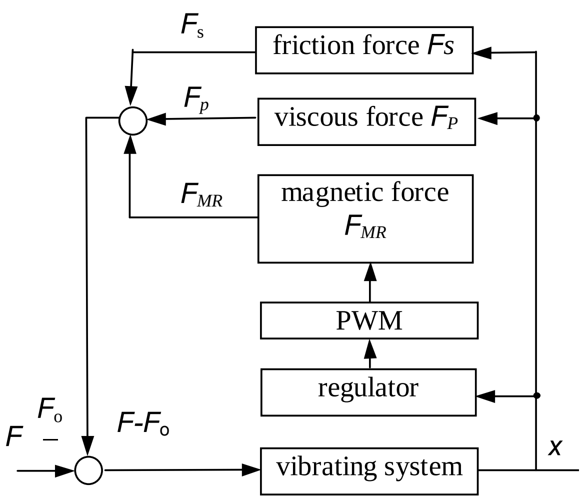


Fig.10. Block diagram of the vibration damping system with the MR damper

Figure 11 shows the calculated characteristics $X(f)$ of the system without a damper and with a damper at the exciting current $i=0$. In such conditions the damper generates a weak damping force that equals the sum of the forces $F_P + F_S$ and behaves as a conventional viscous damper.

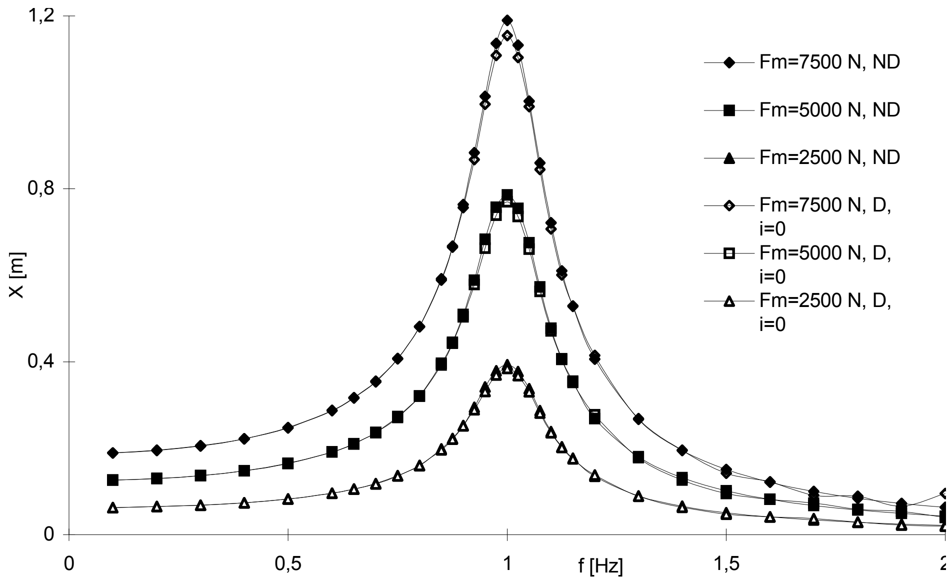


Fig.11. Relationship *X*(*f*) for the system without damper (ND) and with the damper (D) (*i*=0);

Figure 12 shows the relationships $X(f)$ achieved in the system with an MR damper at $i=0$ and the relationships $X(f)$ for the system with the regulated damping force F_o (i=var). It appears from the figure that a system with a damper controlled by a regulator enables a significant reduction of vibrations. This is specifically observable when vibrations of resonance frequency are suppressed. The system is far less effective in controlling the vibrations of the amplitude F_m=7500 N. The reason for this is that the value of the damping force which can be generated by the MR damper is smaller than the force generating vibrations.

Figure 13 shows the relationships of damping force $F_0(x)$ and the velocity $V(x)$ for the current in the exciting winding of the damper $i=0$. The damping force is proportional to the speed of the vibrating mass. Figure 14a) shows the damping force $F_0(x)$ in the system with the regulator. As compared with Fig.13a), an increase of damping force is clearly observable. The force distribution is also different. Figure 14b) shows the characteristic $V(x)$ in the function of the piston velocity for the damper with the regulated damping force. In the model shown it is easy to change the parameters of the damper and the vibrating system and use it for simulating operating conditions of various vibration damping systems.

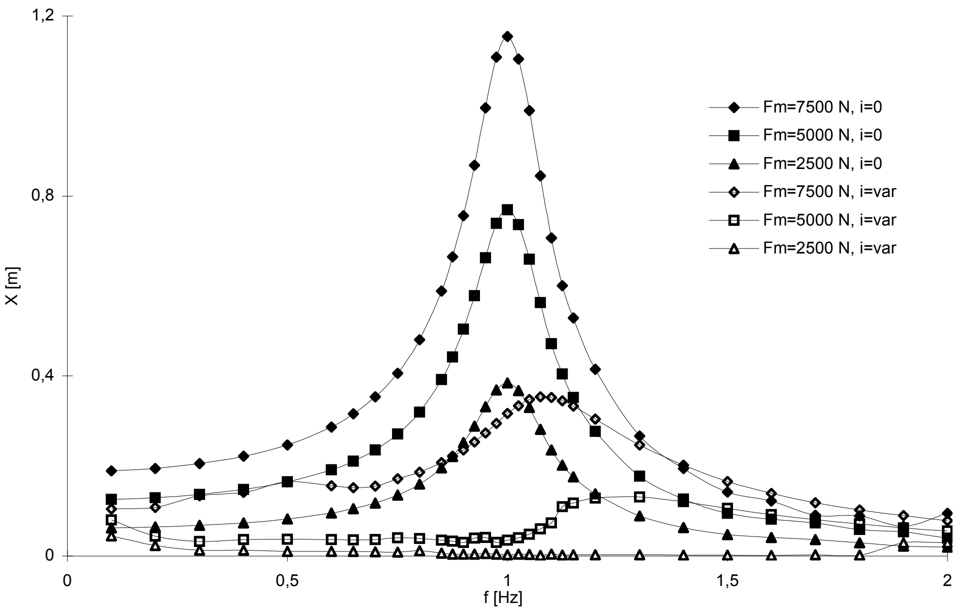


Fig.12. Relationship *X*(*f*) for the system with the damper without the regulator (*i*=0) and with the regulator (*i*=var);

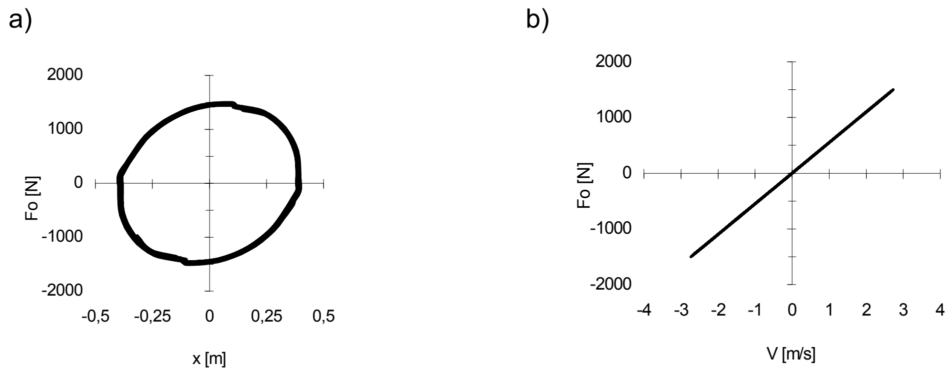
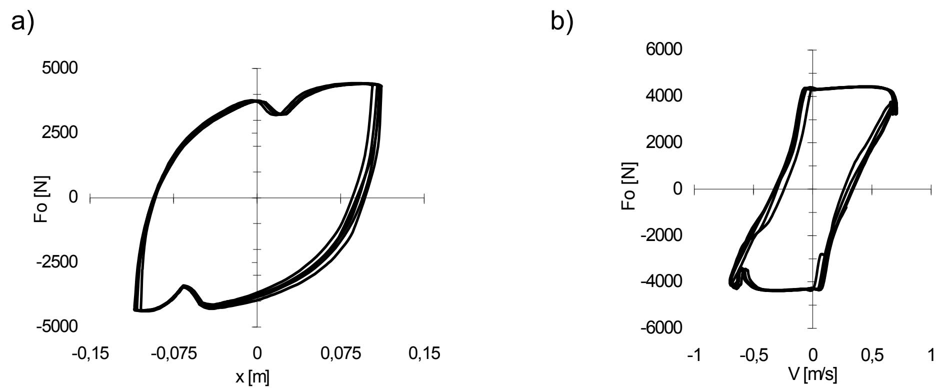


Fig.13. Characteristics a) $F_o(x)$, b)$F_o(V)$ for i=0; F_m=5000 N and f=1.1 Hz



Rys.14. Characteristics a) $F_0(x)$, b) $F_0(V)$ for i=var; F_m=5000 N and f=1.1 Hz

3. CONCLUSION

Vibrations in a one-degree-of-freedom system were considered in the paper. An electromagnetic MR damper was used to limit the amplitude of vibrations. Two mathematical models of the vibrating system with an MR damper were elaborated. In the first model the properties of the damper were described using the Bingham model. The model does not account for the nonlinearity of the magnetic field of the damper and therefore it is not very convenient for the analysis of real operating conditions of vibration damping systems. In the latter model the nonlinearity of the magnetic circuit and the resisting force produced by the friction of moveable elements of the damper against the seals were considered.

Simulations of the models were conducted. Two systems for automatic vibration reduction were considered. It was noted that the application of a regulation system significantly reduces vibrations in the system. An MR damper works well with vibrations of resonance frequency.

If the distribution of forces that are suppressed is known in advance, the appropriate structure and settings of the regulator in the automatic vibration reduction system can be chosen. Consequently, contrary to conventional damping systems high effectiveness of vibration damping can be achieved.

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ANALIZA DRGAŃ W UKŁADZIE Z TŁUMIKIEM MAGNETOELEKTRYCZNYM

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STRESZCZENIE *Z intensywnym rozwojem nowoczesnych urządzeń mechanicznych o coraz to lepszych parametrach dynamicznych, wiąże się konieczność efektywnego tłumienia powstających w nich drgań. W wyniku ograniczenia drgań wzrasta precyzja jak i zakres zastosowań tych urządzeń. Stają się one również bardziej przyjazne dla człowieka. Nie emitują szkodliwych dla zdrowia ludzkiego drgań i hałasu o dużym natężeniu. Zastosowanie tłumików drgań z cieczami magnetoreologicznymi (MR), umożliwi tłumienie drgań urządzeń mechanicznych, ze znacznie większą skutecznością w porównaniu z konwencjonalnymi układami tłumiącymi. Dużą zaletą takich tłumików jest możliwość regulowania w prosty sposób w szerokim zakresie siły tłumiącej. Uzyskuje się to przez zmianę pola magnetycznego w tłumiku z cieczą MR.*

W pracy rozpatrywano drgania w układzie o jednym stopniu swobody przedstawionym na rys.2. Do ograniczenia amplitudy drgań wykorzystano elektromagnetyczny liniowy tłumik magnetoreologiczny. Opracowano dwa modele matematyczne układu drgającego z takim tłumikiem. Posłużono się przy tym klasycznym ujęciem obwodowym. W pierwszym właściwości tłumika opisano za pomocą modelu Binghama (rys.5). W modelu tym nie uwzględnia się nieliniowości obwodu magnetycznego tłumika, jest on zatem mało przydatny do analizy stanów pracy rzeczywistych układów tłumiących drgania. W drugim modelu układu drgającego (rys.10) wykorzystano model tłumika uwzględniający nieliniowość obwodu magnetycznego oraz siłę oporową wynikającą z tarcia elementów ruchomych tłumika o uszczelnienia.

Przeprowadzono badania symulacyjne opracowanych modeli tłumików oraz układów do automatycznego ograniczania drgań. Analizowano wybrane stany pracy układu drgającego z tłumikiem. Przedstawiono wyniki badań symulacyjnych i podano wnioski odnośnie opracowanego modelu matematycznego zjawisk oraz skuteczności tłumienia drgań za pomocą sterowanych tłumików magnetoelek-

trycznych. Stwierdzono, że zastosowanie tłumika z układem regulacji zdecydowanie ogranicza drgania w układzie. Tłumik MR tłumi dobrze drgania układu o częstotliwości rezonansowej. Uzyskano dużą skuteczność tłumienia drgań, trudną do otrzymania za pomocą tradycyjnych układów tłumiących.