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APPLICATION OF PIEZOELECTRIC ELEMENTS TO SEMI-ADAPTIVE DYNAMIC ELIMINATOR OF TORSIONAL VIBRATION

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The paper is concerned with the problem of vibration damping in a torsional system by application of dynamic eliminator equipped with piezoelectric elements. Piezoelectric elements are capable of changing their overall dimensions when exposed to external electric field. If shaped in the form of rings and placed side by side they create a cylindrical shell the radius of which can be controlled by a voltage signal according to the converse longitudinal piezoelectric effect. The dynamic vibration eliminator with such a piezoelectric cylinder mounted in the space filled up with a viscous medium coaxially to the internal cylindrical surface grinded inside of the eliminator housing can adjust its damping coefficient by changing dimensions of the oil gap by applying the voltage to the piezoelectric elements. This property enables the eliminator to adapt its damping to variable excitation frequency so that the optimum damping constant can be obtained. This protects the vibrating system from, what is characteristic for typical dynamic eliminators, disadvantageous growth of the vibration amplitude outside of the resonant zones.

Key words: piezoelectricity, torsional vibration, dynamic eleminator

1. Introduction

Recently much attention has been paid to the problem of application of piezoelectric materials to mechanical systems in which the piezoelectrics are supposed to reduce mechanical vibration or stabilise such systems. Numerous examples of active damping, mainly of transverse vibration in beam-like systems, can be found in literature in the last decade. Success of piezoelectric elements, especially their market representatives in the form of PVDF polymers (polyvinylidene fluoride) and PZT ceramics (lead zirconate titanate) results from their low price, easy accessibility, high reliability and excellent dynamic properties. The time required for piezoelectric elements to respond is negligible relative to vibration periods observed in typical mechanical systems. Finally, they are controlled by voltage – perhaps the easiest signal to be generated and transformed by control, electronic systems. Piezoelectric elements are readily applied to mechanical systems where they introduce artificial, additional active damping by measuring mechanical signal, e.g. vibration, and converting it into an electric one (sensors), then producing a mechanical effect (actuators) just after receiving the appropriately transformed electric signal from the control unit in order to counteract the vibration.

One can come across an abundance of works dealing with the problem of active damping of transverse vibration in one-dimensional continuous systems. The early works by Bailey and Hubbard (1985), Crawley and de Luis (1987), Crawley and Anderson (1990) are worth to be mentioned here together with studies carried out in Poland by Tylikowski (1993), (1997) and Pietrzakowski (1993), (1997). Researchers are also interested in two-dimensional continua, e.g. Kim et al. (1993) examined various flat-shaped piezoelectric actuators to determine the way they interact mechanically with the base structure. Niekerk et al. (1995) considered the use of PVDF actuators for noise attenuation in circular plates. Tylikowski (1997) analysed efficiency of piezoelements in rotating circular plates, where the technical motivation of his work was the attempt to reduce transverse vibration observed in buzz saws. An interesting approach to active damping in shells proposed Tzou (1991) by attracting attention to biomechanical analogies and discussing a concept on piezoelectric neurons and muscles.

Not only transverse vibration damping in mechanical systems is of interest of scientists and engineers. An important class of systems represent torsional systems in which reduction of angular vibration is the goal the researchers aim at. The ability of piezoelectric elements to measure torsional displacement of tubes studied Meng-Kao Yeh et al. (1994). A closed-loop control with coupled piezoelectric sensors and actuators harnessed to active damping of torsional vibration was discussed by Chia-Chi Sung et al. (1994). As it was analysed by Tylikowski (1993) in beam-like systems, where the effect of non-perfect attachment of piezoelements to the base structure was examined, also Przybyłowicz (1995) sought for analogous effects in torsional systems.

This time however, the author intends to give up the concept of vibration damping realised in an active manner, which in fact can be efficient in systems with thin-walled and light shafts, in order to concentrate on applying piezoceramic elements to dynamic eliminators capable of decreasing vibration amplitude in strongly loaded, heavier systems. The concept here is to use PZTs in a dynamic eliminator to make it more flexible to varying loading conditions so that the eliminator effectiveness could be optimised.

Variable frequency of torque exciting the given torsional system makes the dynamic elimination disadvantageous in certain regions if the stiffness and damping parameters of it are structurally fixed. By incorporating piezoelectric elements it is possible to adjust some of these parameters to a changing frequency, obtaining in this way a semi-adaptive eliminator, much more effective from the point of view of its serviceability. Even the simplest strategy toward the concept of its adaptivity can bring encouraging results. The concept presented in this paper consists in affecting by PZT elements the damping coefficient of the torsional dynamic eliminator, making it no longer constant.

2. Model of the system

Consider the discrete-continuous torsional system consisting of three inertia discs I_1, I_2, I_3 and shaft connecting the first and second discs. The system is exposed to mechanical excitation by harmonic torque $M_0 \sin \nu t$ applied to the first disc. The output of the system is reflected by the inertia I_3 vibration of which is to be minimised by dynamic eliminator. Inertia of the eliminator is expressed by I_2 (partly by I_3). The dynamic eliminator, placed between I_2 and I_3, introduces stiffness developing the internal torque M_k and damping producing torque M_c, see Fig.1.

The eliminator is capable of adjusting its damping properties to varying excitation frequency according to the previously prepared program. This program presents a function of the voltage signal that is applied to piezoelectric elements inside of the eliminator. When under voltage these elements change their dimensions what can be the key factor in affecting the damping coefficient. The idea of construction of the eliminator with adaptive damping is shown in Fig.2.

The figure presents the eliminator designed for operating at the end of a shaft (end-tip eliminator). It introduces inertia, which can be tuned up by the additional disc and elasticity in the form of rubber elements (not shown in the figure). The damping is realised in the way as it is done in a typical dashpot, i.e. by oil viscosity generating resistive force between two cylinders in a relative motion. The outer cylinder is posed by the eliminator housing

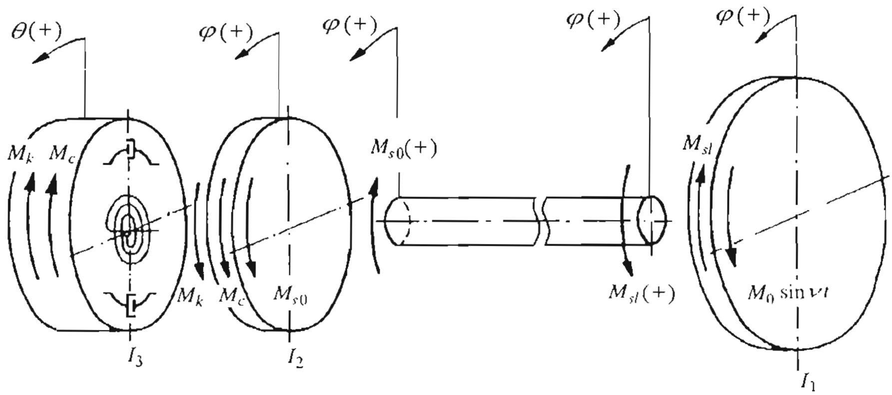

Fig. 1. Model of the system

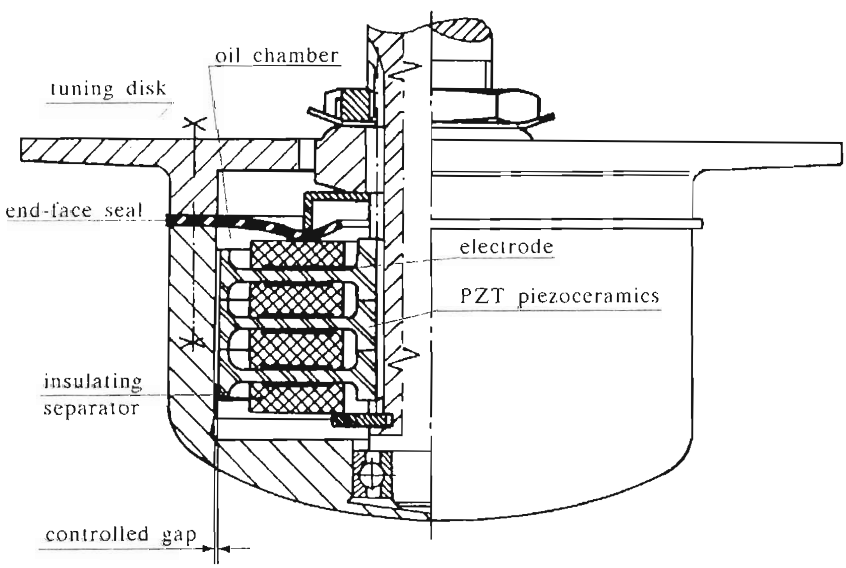

Fig. 2. Scheme of the dynamic eliminator

with a precisely grinded surface. Inside of it the inner cylinder rotates during vibration. Its surface is created by set of piezoelectric rings pressed against each and separated by insulating elements, which additionally reinforce the brittle PZT elements. It should be remembered that the rings must be very thin, so that the electrodes could be placed very close, and thus the electric field possibly high. Radial elongation of the PZT rings is proportional to magnitude of the electric field. On the other hand, it is clear that the PZTs cannot be too thin since lead zirconate titanate is a brittle material and has limited dielectric strength. This is why the number of the rings should be big enough to create a cylinder of considerable length.

The space between the piezoelements and the eliminator housing is filled with the oil of absolute viscosity μ. The oil chamber is closed by the end-face seal, as shown in Fig.2. The oil gap between the cylinders has the nominal size of $g_{t_{\rm nom}}$, i.e. under no voltage. Application of an electric field in the direction opposite to natural polarisation of PZT material results in elongation of the piezoelement. This way the oil gap tightens. The gap can be enlarged by simple change of the electric field sense, which introduces no other effects or difficulties. If dimension of the nominal gap is sufficiently small then application of electic fields of order of 10^6 V/m can vary it to an extent substantially changing the damping constant. To see this in numbers one must concentrate on constitutive equations of piezoelectric materials.

3. Constitutive equations of piezoelectrics

The first-order approximation of constitutive equations of piezoelectric materials can be given in a convenient form, as it was done by Nye (1985) or Damjanović and Newnham (1992):

$$\begin{aligned} \varepsilon_{ij} &= s_{ijkl}^{(E)}\sigma_{kl} + d_{ijk}E_k \\ D_i &= d_{ijk}\sigma_{jk} + \epsilon_{ij}^{(\sigma)}E_j \end{aligned} \tag{3.1}$$

where ε_{ij} and D_i denote the strain and dielectric displacement and σ_{kl}, E_k are the components of the stress tensor and the electric field vector, respectively. The $s_{ijkl}^{(E)}$ coefficients are the elements of the elasticity tensor for a constant electric field. The coefficients d_{ijk} and $\epsilon_{ij}^{(\sigma)}$ denote the linear electromechanical coupling and dielectric permittivity for a constant stress, respectively. Because of symmetry of the stress and strain tensors the third-rank tensor d_{ijk} is also symmetric in indices j and k. This makes it possible to derive a simpler, matrix form notation of Eqs (3.1) by incorporating the following index formula: $ii \to i$ and $23(32) \to 4$, $13(31) \to 5$, $12(21) \to 6$.

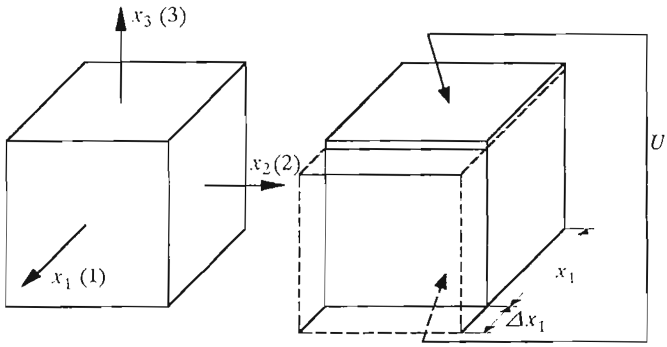

Fig. 3. Converse longitudinal (transverse) piezoelectric effect

From the constitutive equations (3.1) one can obtain a relationship enabling him to calculate basic parameters like mechanical strains or electric fields that appear in the longitudinal piezoelectric effect. This effect is presented in Fig.3. Application of the electric field E_3 results in elongation of the PZT element in the direction x_1. In fact, piezoelectrics made of lead zirconate titanate have the same electromechanical coupling coefficients in both directions x_1 and x_2, i.e. $d_{13} = d_{12} = -170 \cdot 10^{-12}$ m/V, so the PZT cube shown in Fig.3 swells along both perpendicular axes x_1 and x_2. When shaped as a ring the element elongates radially of $\Delta r = \sqrt{\Delta x_1^2 + \Delta x_2^2}$ under E_3. Let us denote the PZT ring thickness by h_3 and its radius by r, see Fig.4. The elongation will be the following

$$\Delta r = \varepsilon r = d_{13} E_3 r = d_{13} \frac{V_3}{h_3} r \tag{3.2}$$

4. Concept of dynamic eliminator with piezoelements

Eq (3.2) results directly from the first one of the constitutive equations given by (3.1) under no stress. For instance, the ring equipped with electrodes distant of 0.6 mm and covering the area on the length of 38 mm is capable

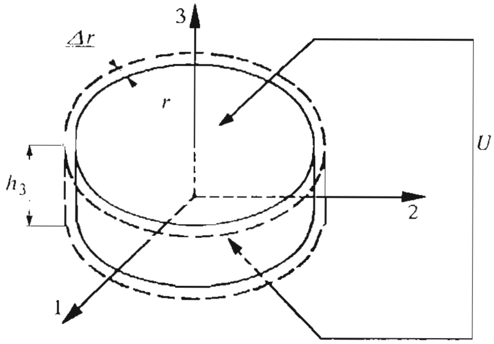

Fig. 4. PZT ring uder transverse electric field

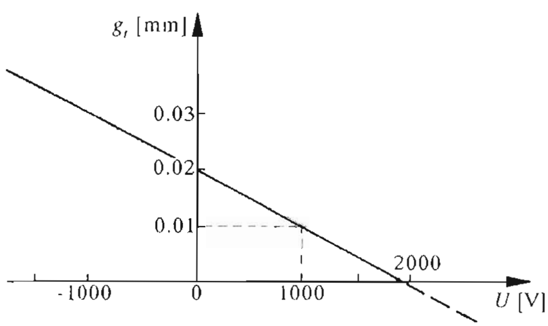

Fig. 5. Gap in the eliminator versus applied voltage

of changing its radius about ± 0.02 mm under the voltage ± 1900 V (in such conditions the thus developed electric field is 25 times below the PZT dielectric strength threshold)., By assuming the nominal gap to be sized of 0.02 mm (as in typical journal bearings) we see that it is easy to change essentially this size, naturally within the range of manufacturing deviations. The main drawback of such an approach lies just in high requirements imposed on dimensional accuracy of the cooperating surfaces (very low deviations) what entails greater costs of after-finishing. But the advantage of precise adaptiveness is the gain. Variablity of the transverse gap g_t in function of voltage is shown in Fig.5. The straight line presented in Fig.5 is represented by the equation: $g_t = g_{t_{\rm nom}} - U\alpha$, where $\alpha = 10.4 \cdot 10^{-6}$ mm/V and U is the applied voltage (in Volts). At this moment it is possible to find the damping coefficient of the dynamic eliminator in function of the applied voltage. From the simplest torsional dashpot model, shown in Fig.6, we can find the viscous damping constant.

On the assumption of the laminar fluid flow between the cylinders one obtains (according e.g. to handbook by Rao, 1990) the following relationship

$$c = \frac{\pi\mu D^3}{32}\Big(\frac{8l}{g_t} + \frac{D}{g_f}\Big) \tag{4.1}$$

where

μ	–	absolute oil viscosity
g_f, g_t	–	front and transverse gaps, respectively (see Fig.6)
D, l	–	inner cylinder diameter and the length immersed in the oil.

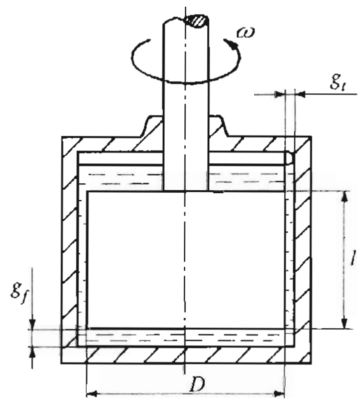

Fig. 6. Torsional damper model

In the dynamic eliminator shown in Fig.2 the front gap is too big to be taken into account. As it introduces negligible damping one can simplify Eq (4.1) to btain finally

$$c = \frac{\pi\mu D^3 l}{4g_t} \qquad\qquad c(U) = \frac{\pi\mu D^3 l}{4(g_{t_{\rm nom}} - \alpha U)} \tag{4.2}$$

Eqs (4.2), which is one of the governing equations of the dynamic eliminator of torsional vibration closes the analysis of its basic parameters as the relationship between the damping coefficient and the voltage is explicitly found.

5. Equations of motion of the system

According to Fig.1, together with the internal torques M_k, M_c, M_{s0}, M_{sl} depicted in it, one can derive the equations of motion of the presented system and formulate corresponding boundary conditions. These equations and conditions have the following form

$$\begin{aligned} I_3\ddot{\theta}(t) &= -k[\theta(t) - \varphi(0,t)] - c[\dot{\theta}(t) - \dot{\varphi}(0,t)] \\ I_2\ddot{\theta}(t) &= k[\theta(t) - \varphi(0,t)] + c[\dot{\theta}(t) - \dot{\varphi}(0,t)] + G^* J_0 \varphi'(0,t) \\ I_1\ddot{\varphi}(l,t) &= M_0 \sin \nu t - G^* J_0 \varphi'(l,t) \end{aligned} \tag{5.1}$$

where

I_i	–	mass moments of inertia of the discs, $i = 1,2,3$
k	–	stiffness constant of the eliminator (between I_2 and I_3)
c	–	adaptive damping coefficient, $c = c(U)$, see Eqs (4.2)
J_0	–	cross-section moment of inertia of the shaft
M_0	–	amplitude of the excitation torque applied to the disc I_1
$\theta(t)$	–	absolute angular displacement of the third disc
$\varphi(0,t), \varphi(l,t)$	–	absolute angular displacements of the shaft beginning point $(x = 0)$ and ending point $(x = l)$, respectively.

The torques denoted in Fig.1 by M_{s0} and M_{sl} are the internal torques developed in the shaft at its ends, i.e. $M_{s0} = G^* J_0 \varphi'(0,t)$ and $M_{sl} = G^* J_0 \varphi'(l,t)$. Eqs (5.1) include also G^* constant being the operator form of the Kirchhoff modulus representing the presence of internal friction in the shaft material. The Kelvin-Voigt model of the internal friction is assumed

$$G^* = G\Big(1 + \beta\frac{\partial}{\partial t}\Big) \tag{5.2}$$

where β is the retardation time corresponding to this model for the shear effect.

Eqs (5.1) must be then completed with the equation of motion of the shaft

$$\frac{\partial^2 \varphi}{\partial t^2} - a^{*2}\frac{\partial^2 \varphi}{\partial x^2} = 0 \qquad a^* = \sqrt{\frac{G^*}{\rho}} \tag{5.3}$$

where ρ is the mass density of the shaft. By predicting a harmonic solution to Eq (5.3) in the form of $\varphi(x,t) = e^{rt}e^{i\nu t}$ we obtain the following characteristic equation

$$r^2a^2(1 + i\beta\nu) + \nu^2 = 0 \qquad a^2 = \frac{G}{\rho} \tag{5.4}$$

which yields the two characteristics roots

$$r_{1,2} = \pm i\frac{\nu}{a\sqrt{1 + i\beta\nu}} = \pm i\frac{\nu}{a^*} \qquad a^* = a\sqrt{1 + i\beta\nu} \tag{5.5}$$

Hence the predicted solution to Eq (5.3) can be written down as

$$\varphi(x,t) = \Big[C_1 \mathrm{e}^{-\mathrm{i}\frac{\nu x}{a\sqrt{1+\mathrm{i}\beta\nu}}} + C_2 \mathrm{e}^{\mathrm{i}\frac{\nu x}{a\sqrt{1+\mathrm{i}\beta\nu}}}\Big]\mathrm{e}^{\mathrm{i}\nu t} \tag{5.6}$$

Assuming that the angular displacement is also a harmonic function, i.e. $\theta(t) = \theta_0 \exp(\mathrm{i}\nu t)$, then substituting it together with Eq (5.6) into Eqs (5.1), we obtain a set of linear simultaneous equations with respect to the complex unknowns C_1, C_2 and θ_0. By putting them down in a matrix form we get

$$\begin{bmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ 0 & a_{32} & a_{33} \end{bmatrix} \begin{bmatrix} \theta_0 \\ C_1 \\ C_2 \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ M_0 \end{bmatrix} \tag{5.7}$$

where

$$a_{11} = \nu^2 I_3 + k + \mathrm{i}\nu c(U)$$

$$a_{12} = a_{13} = a_{21} = -k - \mathrm{i}\nu c(U)$$

$$a_{22} = -\nu^2 I_2 + k + \mathrm{i}\nu c(U) + \mathrm{i}GJ_0\frac{\nu}{a}\sqrt{1+\mathrm{i}\beta\nu}$$

$$a_{23} = -\nu^2 I_2 + k + \mathrm{i}\nu c(U) - \mathrm{i}GJ_0\frac{\nu}{a}\sqrt{1+\mathrm{i}\beta\nu}$$

$$a_{32} = \Big(-\nu^2 I_1 - \mathrm{i}GJ_0\frac{\nu}{a}\sqrt{1+\mathrm{i}\beta\nu}\Big)\mathrm{e}^{-\mathrm{i}\frac{\nu l}{a\sqrt{1+\mathrm{i}\beta\nu}}}$$

$$a_{33} = \Big(-\nu^2 I_1 + \mathrm{i}GJ_0\frac{\nu}{a}\sqrt{1+\mathrm{i}\beta\nu}\Big)\mathrm{e}^{\mathrm{i}\frac{\nu l}{a\sqrt{1+\mathrm{i}\beta\nu}}}$$

6. Results of simulation

Having determined the equations of motion that rule the system dynamics one can observe its behaviour by tracing it on dynamic amplitude-frequency characteristics. Since the shaft can perform a steady rotary motion it is more convenient to examine relative angular displacement measured between the third disc, which is the output of the system, and the first disc, from which the harmonic torque disturbance is transmitted through the shaft. Thus, the ralative displacement $\Delta\varphi = |\theta_0 - \varphi(0)|$ is investigated. Obviously, the system possesses an infinite number of resonances, yet for purely technical reasons, and because of the presence of the internal friction in the shaft, further resonances can be neglected in the analysis (they disappear for stronger internal friction). Admittedly, to explain the problem of semi-adaptive eliminator one requires to consider the first two resonances. The first one, shown in Fig.7, occurs at $\nu_{r1} = 143\,\mathrm{rad/s}$ (it can be shifted by replacing the tuning disk if needed) and reveals typical features characteristic for simple linear vibrating systems. Application of higher damping reduces vibration amplitude. This can be realised by increasing the voltage.

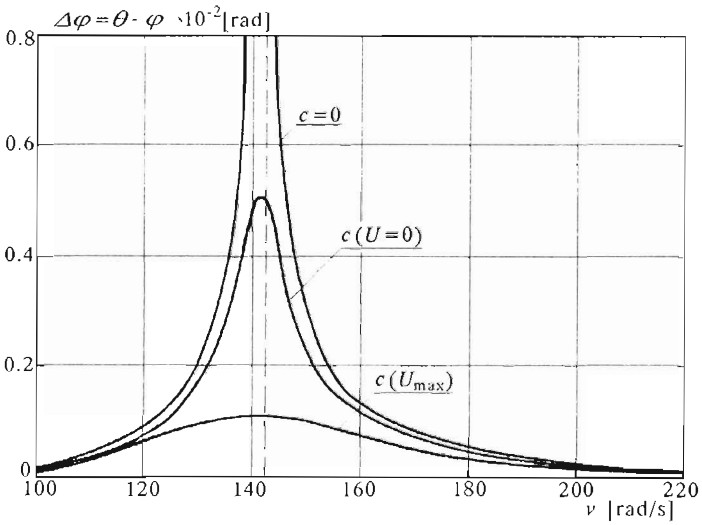

Fig. 7. Damping of first resonance

The nominal oil gap already introduces viscous damping (without voltage signal) which is shown in Fig.7 by the curve denoted by $c(U = 0)$. In this case $g_t = g_{t_{\mathrm{nom}}} = 0.02\,\mathrm{mm}$. Application of voltage ($U_{\max} \approx 1500\,\mathrm{V}$) definitely chokes the vibration amplitude by a drastic growth in damping brought about by a considerable drop in the gap dimension. Yet when kept constant with growing excitation frequency the voltage fulfills its task only in the direct neighbourhood of subsequent resonances. For frequencies far from resonant zones the constant level of damping raises vibration amplitude (which is the cost of good performance near resonances in purely passive dynamic eliminators). Such an effect is presented in Fig.8 showing the second resonance. To make the disadvantageous effect more vivid the corresponding logarithmic characteristics is included as well (Fig.9).

An undamped system (no damping between I_2 and I_3 discs) exhibits virtually no vibration for the excitation frequency ν_e (so-called elimination frequency). Very low vibration amplitude can be gained by switching off the constant voltage signal, or at best, by changing its sign ($U_{\max} \to U_{\min} \approx$

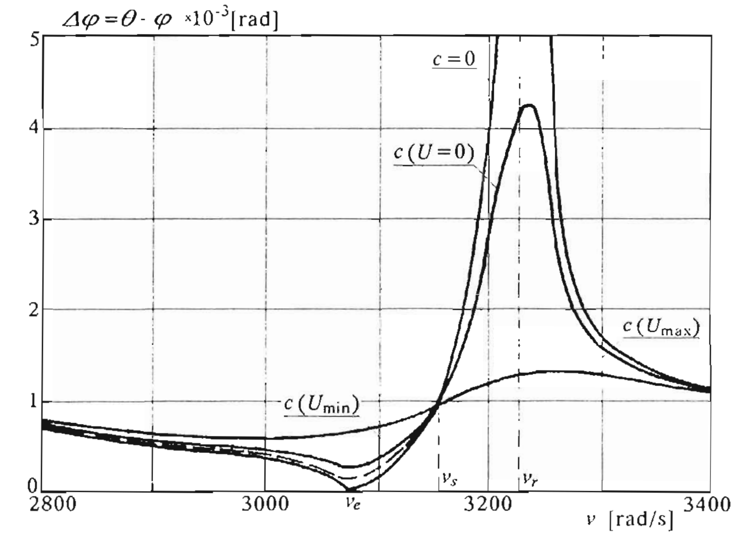

Fig. 8. Damping of second resonance

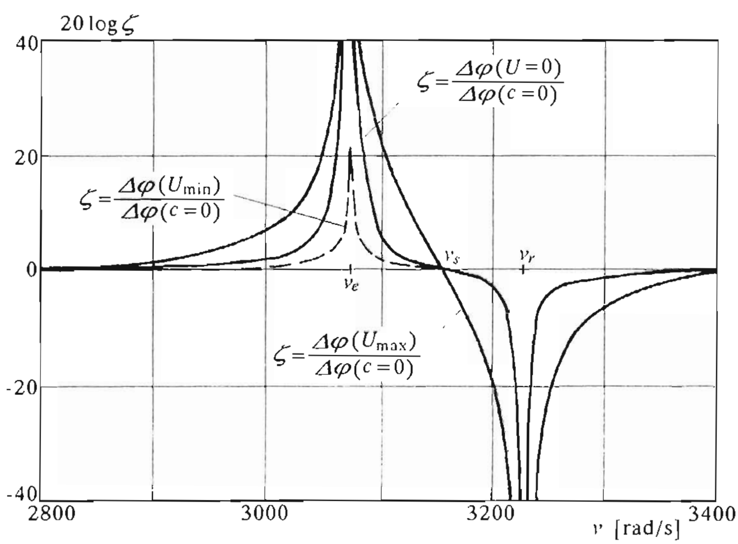

Fig. 9. Second resonance in logarithmic scale

−1500 V). In such a case the system vibrates according to the dashed curves seen in Fig.8 and Fig.9. The opposite sign of the voltage is to be maintained until the frequency ν_s is reached. There intersect all of the curves corresponding to any damping conditions in the system. Then the voltage $U_{\max}$ should be applied again. Obviously, such a situation repeats itself for the third, fourth and further resonances. Even the simplest strategy toward adaptiveness of the dynamic eliminator, i.e. based on applying in turns $U_{\max}$ and $U_{\min}$ signals after subsequent ν_{si} $(i = 1, 2...)$ frequencies, can qualitatively enhance the operation of the dynamic eliminator. Below, given is the table of subsequent frequencies at which the voltage signal should be switched over.

Table 1. Pole reversal frequency ν_s [rad/s]

ν_{s1}	ν_{s2}	ν_{s3}	ν_{s4}	ν_{s5}	ν_{s6}	ν_{s7}	ν_{s8}	ν_{s9}
826	3156	4301	8570	9136	10002	16805	17760	18830

In Fig.10 the voltage signal versus excitation frequency is presented.

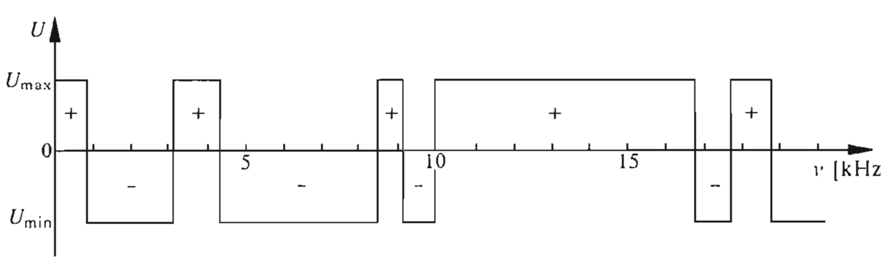

Fig. 10. Voltage as a function of vibration frequency

Function shown in Fig.10 can be realised by embedded in the eliminator body an electronic unit sensitive to frequency. It is worth to emphasise that vibration frequency can be easily measured by an additional piezelectric element working as the accelerometer or just by making use of the same PZT rings assemblied for adapting the oil gap through measuring their torsion according to the so-called shear piezoelectric effect. It would be then enough to replace a changeable electronic card with programmed frequencies pertaining to the given torsional system.

7. Concluding remarks

In the paper a system undergoing torsional vibration with semi-adaptive dynamic eliminator is presented. The adaptivity of the dynamic eliminator is achieved by application of piezoelectric elements which are capable of changing their overall dimensions when exposed to electric field. When applied to a specially devised system they can control size of the gap filled with a viscous medium between moving elements, thus affect the damping properties of such a system. The well known behaviour of classical dynamic eliminators with characteristic and disadvantageous growth in vibration amplitude outside of the resonances can be qualitatively changed by employing the eliminator that is able to develop two different damping coefficients: $c_{\rm min} = c(U_{\rm min})$ and $c_{\rm max} = c(U_{\rm max})$, and adjust them to current frequency of the excitation. Since the program governing the sequence of switching over the voltage signal is independent of the system state, therefore the method can only be regarded as a semi-adaptive approach to dynamic elimination of torsional vibration.

A concept of making the eliminator self-adaptive or, if preferred, semi-acvtive could be the following. The eliminator measures (PZT or PVDF sensors) current vibration amplitude and starts, by design, changing its damping coefficient by e.g. slowly increasing voltage. If the vibration amplitude decreases the electronic unit will raise the signal until $U_{\rm max}$ is reached, then stops, and after a while will repeat the procedure again. If the choice of the voltage increment direction was wrong (vibration amplitude grew) the control unit would immediately change the sign of the voltage and afterwards proceed as mentioned before. Naturally, such a method does not reflect a fully active control, which has to react instantly to varying dynamic conditions. The method requires permanent probing of the vibration amplitude under slight increments of the voltage signal, yet because of its independent action could be understood as a semi-active method.

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Zastosowanie elementów piezoelektrycznych w semi-adaptacyjnym dynamicznym eliminatorze drgań

Streszczenie

Praca poświęcona jest problemowi tłumienia drgań w układzie torsyjnym, gdzie zastosowano eliminator drgań z elementami piezoelektrycznymi. Elementy piezoelektryczne zmieniają swoje wymiary, jeżeli poddane zostaną działaniu zewnętrznego pola elektrycznego. Jeżeli elementy takie ukształtować w formie pierścieni ułożonych jeden obok drugiego stworzą one powłokę walcową, której promień można zmieniać w zależności od przyłożonego napięcia, zgodnie z tzw. odwrotnym wzdłużnym efektem piezoelektrycznym. Dynamiczny eliminator drgań posiadający taki cylindryczny element wykonany z piezoelektryka zamocowany współosiowo z cylindryczną powierzchnią obrobioną w obudowie eliminatora w przestrzeni wypełnionej lepkim czynnikiem może dopasowywać swój współczynnik tłumienia poprzez zmianę wymiarów szczeliny olejowej po przyłożeniu napięcia. Ta właściwość pozwala dostosować tłumienie eliminatora do zmieniającej sie częstości wymuszenia w badanym układzie tak, aby tłumienie to miało wartość optymalną. W ten sposób w danym układzie zapobiega się przed wzrostem amplitudy drgań poza obszarem rezonansu, charakterystycznym dla typowych dynamicznych eliminatorów drgań.

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