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Analysis of rotating systems using actuators with magnetorheological fluid to vibrations minimization – theoretical and experimental investigation

1 Introduction

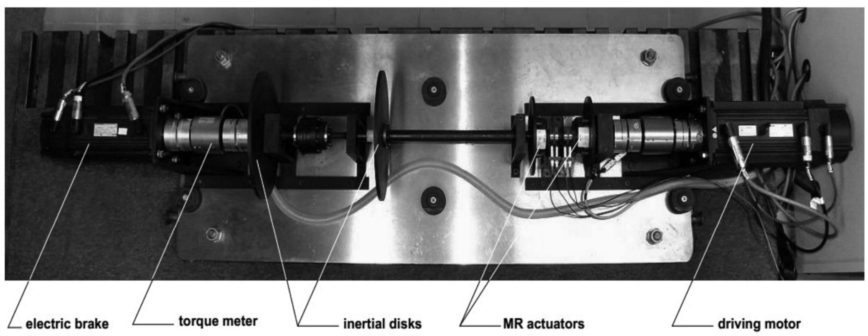
In general torsional vibrations are the source of additional oscillatory angular displacements superimposed to nominal rotational motions of mechanical structure. From among various kinds of vibrations occurring in the rotating systems the torsional ones are very important as naturally associated with their fundamental rotational motion. Unfortunately, it effective minimization is rather difficult to obtain. They provided problems with proper control torque generation and convenient technique of it imposing on quickly rotating shaft segments. The applied so far passive methods are not sufficiently effective in majority of practical applications. In the field of attenuation of torsional vibrations one can find not so many published results of research, beyond some attempts performed by active control using piezo-electric actuators, [1]. In such cases relatively small values of control torques can be generated. Therefore piezo-electric actuators can be usually applied to low-power drive systems. Moreover, even if a relatively big number of the piezo-electric actuators are attached to shafts only higher eigenmodes can be controlled. It results in control of the most important fundamental eigenmodes, which is often not sufficiently effective. Alternative solutions based on actuators with magnetorheological fluid (MRF) applications we can find in [2-3]. The control torques are generated by MRF of adjustable viscosity. This approach enables us an effective suppression of vibrations in several mechanical systems.

In this paper, analysis of the damped rotating system using MRF actuators during steady-state operation is presented. The theoretical investigations are carried out with structural hybrid model and FEM model of the mechanical part of rotating machine. While the electric part of the system is modeled using the circuit model of the electric motor. Some theoretical and numerical results have been confirmed by experimental measurements.

The paper is organized as follows: Section 1 reviews ways of vibration minimization in the rotating systems. Section 2 describes mathematical electromechanical model of a real object, which is imitating an operation of the coal pulverize. Section 3 presented control problem for the torsional vibrations. Section 4 treat about experiment, measurements part and discusses the received results. Finally, some conclusions are given in Section 5.

2 Mathematical model

In order to perform a theoretical investigation reliable and computationally efficient electromechanical models are required. The real structure under consideration is presented
in Fig. 1. To increase a reliability of the obtained computational results, the investigations are performed in two ways. First one by means of the hybrid and second with the classical finite element (FEM) structural electromechanical models. These two models consisting of rigid bodies and continuous or discretized deformable finite elements, respectively, are employed here for eigenvalue analyses as well as for numerical simulations of torsional vibrations of the drive system coupled with electrical vibrations in the motor windings.



Rys. 1. Stanowisko laboratoryjne
Fig. 1. The laboratory test-rig

In the assumed model the control damping torques generated by one MRF actuators can be regarded as the response-dependent external excitations, like in [4]. After a transformation of them into the space of modal coordinates this system of coupled modal equations can be expressed in following way

$$\mathbf{M}_0\ddot{\mathbf{r}}(t)+[\mathbf{D}_0+\mathbf{D}_C(k_j(t),\dot{\mathbf{r}}(t))]\dot{\mathbf{r}}(t)+\mathbf{K}_0\mathbf{r}(t)=\mathbf{F}(t,\dot{\mathbf{r}}(t)),\ j=1,2, \tag{1}$$

where $\mathbf{M}_0$, $\mathbf{K}_0$ and $\mathbf{D}_0$ denote the constant diagonal modal mass, stiffness and damping matrices, respectively. The matrix $\mathbf{D}_c$ plays here a role of the semi-active control matrix. The symbol $\mathbf{F}(t,\dot{\mathbf{r}}(t))$ denotes the response external excitation vector partially depended on the electromagnetic torque generated by the electric motor and the retarding torque produced by the driven imitated rotating machine.

In order to minimized torsional vibration in the drive system, the electromagnetic torque, which is generated by the motor should be described possibly accurately. Then, the mechanical engineers applied external excitations produced by the motor as a ‘a priori’ assumed time or the rotor-to-stator slip function [5-6]. In many cases, such simplifications yield sufficiently useful results for engineering applications, but very often they can lead to remarkable inaccuracies, since many qualitative dynamic properties of the mechanical systems, e.g. their mass distribution, torsional flexibility and damping effects, are being neglected. To obtain more satisfactory precision in the case of the symmetrical three-phase servo-asynchronous motor electric current oscillations in windings are described by the six circuit voltage equations, which are transformed into the system of four Park's equations in so called '*αβ-dq*' reference system, which can be found e.g. in [7]

$$
\begin{bmatrix} \sqrt{\frac{3}{2}}U\cos(\omega_e t) \\ \sqrt{\frac{3}{2}}U\sin(\omega_e t) \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} L_1+\frac{1}{2}M & 0 & \frac{3}{2}M & 0 \\ 0 & L_1+\frac{1}{2}M & 0 & \frac{3}{2}M \\ \frac{3}{2}M & 0 & L_2'+\frac{1}{2}M & 0 \\ 0 & \frac{3}{2}M & 0 & L_2'+\frac{1}{2}M \end{bmatrix} \cdot \begin{bmatrix} \dot{i}_\alpha^s(t) \\ \dot{i}_\beta^s(t) \\ \dot{i}_d^r(t) \\ \dot{i}_q^r(t) \end{bmatrix} +
$$

$$
+\begin{bmatrix} R_1 & 0 & 0 & 0 \\ 0 & R_1 & 0 & 0 \\ 0 & \frac{3}{2}pM\Omega(t) & R_2' & p\Omega(t)\left(L_2'+\frac{1}{2}M\right) \\ -\frac{3}{2}pM\Omega(t) & 0 & -p\Omega(t)\left(L_2'+\frac{1}{2}M\right) & R_2' \end{bmatrix} \cdot \begin{bmatrix} i_\alpha^s(t) \\ i_\beta^s(t) \\ i_d^r(t) \\ i_q^r(t) \end{bmatrix}, \tag{2}
$$

where U denotes the power supply voltage, ω_e is the supply voltage circular frequency, L_1, L_2' are the stator coil inductance and the equivalent rotor coil inductance, respectively, M denotes the relative rotor-to-stator coil inductance, R_1, R_2' are the stator coil resistance and the equivalent rotor coil resistance, respectively, p is the number of pairs of the motor magnetic poles, $\Omega(t)$ is the current rotor angular speed including the average and vibratory component. Moreover, variable M is the relative rotor-to-stator coil inductance, p denotes the number of pairs of the motor magnetic poles and i_α^s, i_β^s are the electric currents in the stator reduced to the electric field equivalent axes α and β and i_d^r, i_q^r are the electric currents in the rotor reduced to the electric field equivalent axes d and q, [7].

The electromagnetic torque generated by such a motor can be expressed by the following formula

$$
T_{el} = \frac{3}{2}pM\left(i_\beta^s \cdot i_d^r - i_\alpha^s \cdot i_q^r\right). \tag{3}
$$

From the form of Park's equations (2) and from formula (3) it follows that the coupling between the electric and the mechanical system is non-linear in character, particularly for significantly varying motor rotational speed $\Omega(t)$. Such a coupling leads to very complicated analytical description resulting in rather harmful computer implementation. The electromechanical coupling phenomena can be realized here by means of the step-by-step numerical extrapolation technique, which for relatively small direct integration steps for equations (1) results in very effective, stable and reliable results of computer simulation. The electromechanical drive systems modelling is widely described in [8].

3 Control problem for the vibrations

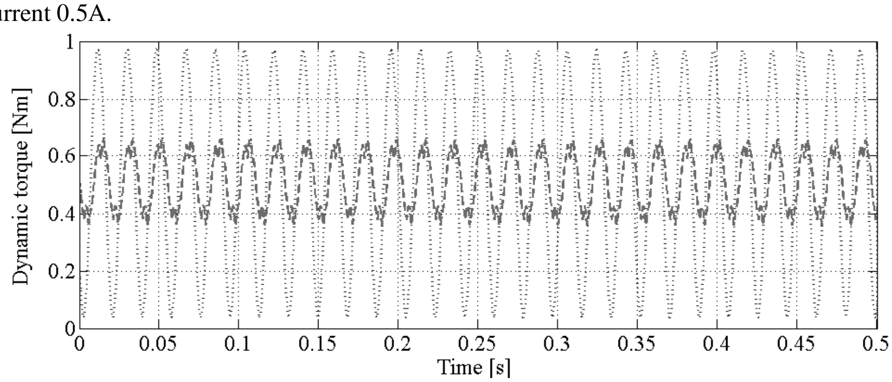
From the theoretical investigations carried out in [9] follows that for the rotating system affected by dynamic loading characterized by clear, predominant frequency components the open-loop semi-active control is similarly effective in suppression of vibration amplitudes as the closed-loop one. This allow us to reduced control problem to a selection of the 'a priori' assumed optimal damping parameters, damping torque generated by MRF actuators and damping coefficient $d(i(t))$. To provided an optimum value of the control current applied to the rotary actuators, the following criterion has to be assumed

$$d_0 = \arg\min_d \max_f \mathrm{FRF}_r(f, d), \tag{4}$$

were d_0 is optimal damping coefficient with respect to the frequency response function (FRF) of the damped drive system excited with the predominant excitation frequency f of the transmitted loading.

From calculated frequency response function follows that for the system dynamic parameters it course rapidly falls with the rise of the damping coefficient to rich its minimum for ca. 1.0 - 1.5 Nms/rad per one rotary damper at the excitation frequency 54.4 Hz corresponding to the first torsional eigenvibration mode. As it follows from the identified in [9] characteristics of the rotary magnetorheological actuators the optimal value of the damping coefficient $d_0(i_0) \approx 1.0$ - 1.5 Nms/rad. It can be realized by the control current $i_0 \approx 1.0$ A.

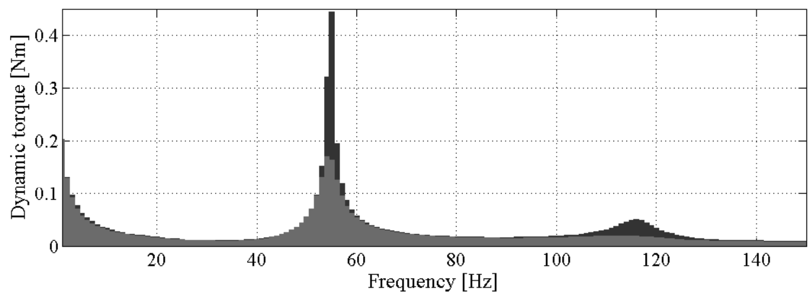
Figure 2 presented time histories plot of the calculated dynamic torques registered between the driving motor and the magnetorheological actuators, obtained respectively for the undamped system (blue line) and damped system (red line), both under resonant excitation with the above-mentioned frequency about 54Hz. The course of damped system has been measured with an optimally controlled actuators with the control current 0.5A.



Rys. 2. Obliczony przebieg momentu dynamicznego (układ niettłumiony- kolor niebieski, układ tłumiony- kolor czerwony)

Fig. 2. Calculated time histories of the dynamic torque (undamped system- blue, damped system- red)

In figure 3 there are shown plots of dynamic response amplitudes of the undamped object (blue line) and damped system (red line) determined by means of measurements. In figure 3 we can see the significant peak corresponding to ca. 54.5 Hz, which is related to the resonance frequency with the system first, fundamental 'elastic' natural frequency. Furthermore the second eigenform is also remarkable in the vicinity of 112 Hz.



Rys. 3. Obliczone amplitudowe charakterystyki momentu dynamicznego (układ niełumiony- kolor niebieski, układ tłumiony- kolor czerwony)

Fig. 3. Amplitude characteristic of the dynamic torque obtained using computation (undamped system- blue, damped system- red

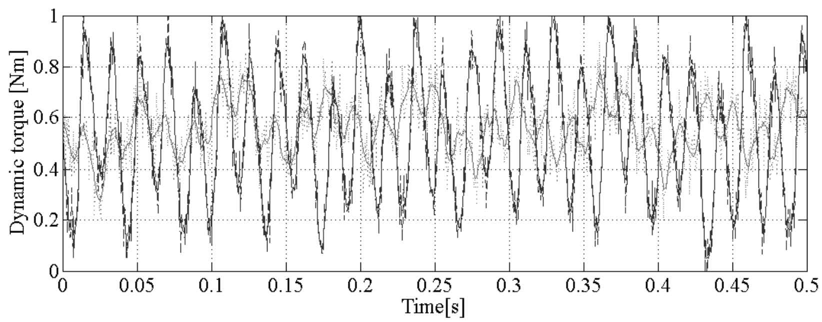
4 Experimental verification

The experimental verification of the theory was performed on the test stand. The laboratory drive system presented in Fig. 1 imitates the functioning of an industrial rotating machine. The attenuation of torsional vibration occurs via rotary inertia fixed to the main system with MRF actuators and control of MRF properties. In this system power is transmitted from servo-asynchronous motor to driven machine in a form of a programmable electric brake. Moreover, this drive system is equipped with two inertial disks with adjustable mass moments of inertia and the possibility of axial positioning. This enables us to tune the drive train to the proper natural frequencies. The control voltage is applied to the MRF actuators with sliders. The external magnetic field acts on the fluid inside the actuators. As a result the characteristics of the MRF fluid are changed, in a way that controls the torsional vibrations.

The measurement-control system consists of voltage amplifier controlled in real time by a computer using the appropriate converting systems. This enables us to monitor and register all the measurements. This is possible through the use of a control-communication unit by means of the TCP/IP protocol. The measurements were taken with frequencies above 15 kHz. This allowed accurate detection of even very rapid changes in torque. The torques are noncontact measurement. The computer carried out real time FFT analyses of the interval lengths per second and recorded the values of the two major peaks of the FFT. This allowed us to keep the value of the dominant peak, depending on the amplifiers of the control signal voltage.

The laboratory test-rig Fig. 1 was experimentally tested in nominal, steady-state operating conditions for the supply voltage frequency set-up to $\omega_e/2\pi$= 60Hz, where on the rated torque the fluctuating sinusoidal component has been imposed.

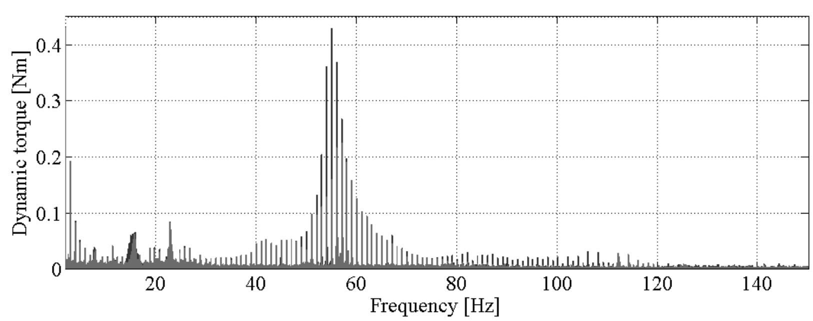
The measurements are performed respectively for the undamped object (without control) and controlled drive system both excited by the harmonic fluctuating component of the retarding torque within the frequency range 0 - 150Hz.



Rys. 4. Zmierzony przebieg momentu dynamicznego (układ niethumiony- kolor niebieski, układ tłumiony- kolor czerwony)

Fig. 4. Measured time histories of the dynamic torque (undamped system- blue, damped system- red)

In the figure 4 one can observed time histories plot of the measured, averaged dynamic torques registered between the driving motor and the magnetorheological actuators, obtained respectively for the undamped system (blue line) and damped system (red line), both under resonant excitation with frequency about 54 Hz. While in Fig. 5 there is presented the amplitude spectrum determined by means of FFT of the measured dynamic torque, Fig. 4. Apart of the visible disturbances introduced by the excitation signal at ca. 16 and 23 Hz, in figure 5 one can notice the significant peak corresponding to ca. 54.5 Hz, which is related to the resonance effect with the system first, fundamental 'elastic' natural frequency.



Rys. 5. Amplitudowe charakterystyki momentu dynamicznego zmierzone doświadczenie (układ nietłumiony- kolor niebieski, układ tłumiony- kolor czerwony)

Fig. 5. Amplitude characteristic of the dynamic torque obtained using experiment (undamped system- blue, damped system- red)

Both from the experimentally Fig. 4 and theoretically Fig. 2 obtained plots it follows that for the most inconvenient resonant operating conditions the rotary magnetorheological actuators minimized torsional vibration amplitudes in the considered shaft segment more than two times. This result has been confirmed by the experimentally Fig. 5 and theoretically Fig. 3 obtained amplitude characteristics using FFT in the abovementioned excitation frequency range 0-150 Hz. We achieved a very good qualitative agreement between the measured and calculated results, taking into account the similarity of the respective extreme values and corresponding to them frequencies.

5 Conclusions

The main theme of work was to study a behavior of the rotating drive system. From computational and experimental investigations, it follows that the steady-state torsional vibrations of a rotating shaft can be reduced with the magneto-rheological actuators proposed in [4]. The optimum control realized by means of the 'a priori' assumed control current, and in this way damping parameters, can essentially reduce steady-state torsional vibrations of the successive shaft segments. Obtained results show that properly selected control current values, which based on the respective minimum of the frequency response function could reduced by about 50-60% level of resonance vibrations.

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Summary

The analysis of damped rotating systems using actuators with magnetorheological fluid is the subject of interest. The main purpose of this research is minimization of vibration amplitudes in order to increase fatigue durability of the most responsible machine elements. Here, control damping torques are generated by magnetorheological fluid of adjustable viscosity. The theoretical investigations are based on a hybrid and finite element structural model (FEM) of the mechanical structure as well as on sensitivity analysis of the response with respect to the actuators damping characteristics. The analysis performed in the paper combines experimental verification using the laboratory test rig with theoretical computations.

Keywords: rotating systems, torsional vibrations, actuators with a magnetorheological fluid, rotary actuators

Analiza układów wirujących z wykorzystaniem aktuatorów z cieczą magnetoreologiczną do tłumienia drgań - badania teoretyczne i eksperymentalne

Streszczenie

Prezentowany materiał poświęcony jest analizie układów wirujących z wykorzystaniem aktuatorów z cieczą magnetoreologiczną o zmiennych właściwościach dyssypacyjnych. Głównym celem realizowanych badań jest zmniejszenie poziomu amplitudy drgań skrętnych, aby zwiększyć trwałość zmęczeniową najbardziej odpowiedzialnych elementów maszyny roboczej. W praktyce sprowadza się to do wyznaczenia optymalnej wartości współczynnika tłumienia, który jest realizowany przez film z cieczy MRF. W pracy przedstawione są wyniki analiz otrzymanych z modeli teoretycznych, poparte badaniami symulacyjnymi. Potwierdzają one rezultaty badań przeprowadzonych na zbudowanym w tym celu stanowisku eksperymentalnym. Dalsze analizy związane z tą tematyką będą dotyczyły identyfikacji defektów w układach wirujących.

Słowa kluczowe: układy wirujące, drgania skrętne, tłumiki MRF, tłumiki obrotowe

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