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Experimental Verification of the Semi-Active Control Concepts for Torsional Vibrations of the Electro-Mechanical System Using Rotary Magneto-Rheological Actuators

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Abstract

In the paper semi-active control of torsional vibrations of the rotating machine drive system driven by an electric motor is performed by means of rotary actuators with the magneto-rheological fluid. The main purpose of these studies is a minimisation of vibration amplitudes in order to increase the fatigue durability of the most responsible elements, assure possibly precise motion of the driven machine working tool as well as to reduce a generated noise level. For suppression of steady-state torsional vibrations excited by dynamic external torques generated by the motor and by the driven object there are proposed control strategies based on a principle of optimum current damping coefficient values realized by the magneto-rheological fluid. The theoretical control concepts are experimentally verified using the laboratory test rig in the form of drive system co-operating with two asynchronous motors generating properly programmed driving and retarding electromagnetic torques.

Keywords: semi-active control, torsional vibrations, rotary actuators, magneto-rheological fluid

1. Introduction

Active vibration control of drive systems of machines, mechanisms and vehicles creates new possibilities of improvement of their effective operation. From among various kinds of vibrations occurring in the drive systems the torsional ones are very important as naturally associated with their fundamental rotational motion. Torsional vibrations are in general rather difficult to control not only from the viewpoint of proper control torque generation, but also from the point of view of a convenient technique of imposing the control torques on quickly rotating parts of the drive-systems. Unfortunately, one can find not so many published results of research in this field, apart of some attempts performed in [1] by active control of shaft torsional vibrations using piezo-electric actuators. But in such cases relatively small values of control torques can be generated and thus the piezo-electric actuators can be usually applied to low-power drive systems. In [2] there is proposed the semi-active control technique based on the actuators in the form of rotary actuators with the magneto-rheological fluid (MRF). In these actuators between the shaft and the inertial ring, which is freely rotating with a velocity close or equal to the system average rotational speed, the magneto-rheological fluid of adjustable viscosity is used. Such actuators generate control torques that are functions of the shaft actual rotational speed, which consist of the average component corresponding to the rigid body motion and of the fluctuating component caused by torsional vibrations.

The general target of this paper is an experimental verification of the presented in [2] theoretical concept of semi-active control of torsional vibration using the rotary actuators with the magneto-rheological fluid. Thus, for this purpose the proper test-rig has been built, using which the measurement results have been compared with theoretical ones determined by means of two mechanical models of identical structure as the real object.

2. Assumptions for the mechanical models and formulation of the problem

In the considered laboratory drive system imitating operation of the rotating machine power is transmitted from the servo-asynchronous motor to the driven machine tool in the form of electric brake by means of the two multi-disk elastic couplings with built-in torque-meters, electromagnetic overload coupling and by the shaft segments. Moreover, this system is equipped by two rotary magneto-rheological actuators and two inertial disks of adjustable mass moments of inertia and axial positions, which enable us to tune-up the drive train to the proper natural frequency values. The considered real laboratory drive system is presented in Fig. 1.

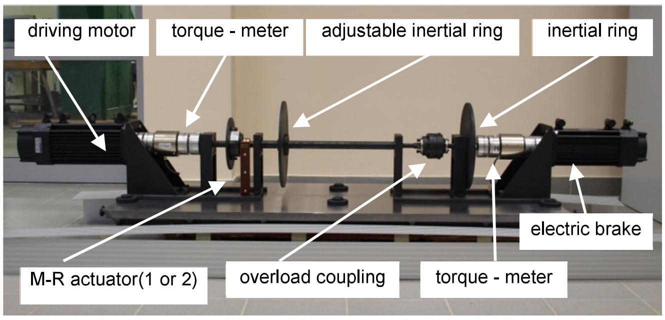

Figure 1. Laboratory drive system

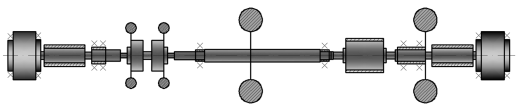

Figure 2. Mechanical model of the laboratory drive system

In order to perform a theoretical investigation of the semi-active control applied for this mechanical system, a reliable and computationally efficient mechanical model is required. In this paper dynamic investigations of the entire drive system are performed by means of two structural models consisting of torsionally deformable one-dimensional beam-type finite elements and rigid bodies, as shown in Fig 2. These are the discrete-continuous (hybrid) model and the classical beam finite element one. Both models are employed here for eigenvalue analyses as well as for numerical simulations of torsional vibrations of the drive train. In the hybrid model successive cylindrical segments of the stepped rotor-shaft are substituted by the cylindrical macro-elements of continuously distributed inertial-visco-elastic properties. However, in the finite element model these continuous macro-elements have been discretized with a proper mesh density assuring a sufficient accuracy of results. In the proposed hybrid and FEM model of the rotating machine drive system inertias of the inertial disks are represented by rigid bodies attached to the appropriate macro-element extreme cross-sections, which should assure a reasonable accuracy for practical purposes. Torsional motion of cross-sections of each visco-elastic macro-element in the hybrid model is governed by the hyperbolic partial differential equations of the wave type. Mutual connections of the successive macro-elements creating the stepped shaft as well as their interactions with the rigid bodies are described by equations of boundary conditions. These equations contain geometrical conditions of conformity for rotational displacements of the extreme cross sections. The second group of boundary conditions are dynamic ones, which contain equations of equilibrium for external and control torques as well as for inertial, elastic and external damping moments.

Similarly as in [2], the solution for forced vibration analysis has been obtained using the analytical - computational approach. Solving the differential eigenvalue problem and an application of the Fourier solution in the form of series in the orthogonal eigenfunctions lead to the set of uncoupled modal equations for time coordinates $\xi_m(t)$. In the assumed model the control damping torques genereded by one rotary actuator with the MRF can be regarded as the response-dependent external excitations. Then, by a transformation of them into the space of modal coordinates $\xi_m(t)$ and upon a proper rearrangements the following set of coupled modal equations is yielded:

$$\mathbf{M}_0\ddot{\mathbf{r}}(t)+\mathbf{D}(k_j(t),\dot{\mathbf{r}}(t))\dot{\mathbf{r}}(t)+\mathbf{K}_0\mathbf{r}(t)=\mathbf{F}(t,\dot{\mathbf{r}}(t)), \qquad (1)$$

where $\mathbf{D}(\dot{\mathbf{r}}(t))=\mathbf{D}_0+\mathbf{D}_C(k_j(t),\dot{\mathbf{r}}(t)),\ j=1,2.$

The symbols $\mathbf{M}_0$, $\mathbf{K}_0$ and $\mathbf{D}_0$ denote, respectively, the constant diagonal modal mass, stiffness and damping matrices. The full matrix $\mathbf{D}_C(k_j(t),\dot{\mathbf{r}}(t))$ plays here a role of the semi-active control matrix and the symbol $\mathbf{F}(t,\dot{\mathbf{r}}(t))$ denotes the response dependent external excitation vector due to the electromagnetic torque generated by the electric motor and due to the retarding torque produced by the driven imitated rotating machine. The Lagrange coordinate vector $\mathbf{r}(t)$ consists of the unknown time functions $\xi_m(t)$ in the Fourier solutions, $m = 1,2,\ldots$. The number of equations (1) corresponds to the number of torsional eigenmodes taken into consideration in the range of frequency of interest. These equations are mutually coupled by the out-of-diagonal terms in matrix **D** regarded as external excitations expanded in series in the base of orthogonal analytical eigenfunctions. A fast convergence of the applied Fourier solution enables us to reduce the appropriate number of the modal equations to solve in order to obtain a sufficient accuracy of results in the given range of frequency. Here, it is necessary to solve only 6÷10 coupled modal equations (1), contrary to the classical one-dimensional rod finite element formulation leading in general to a relatively large number of motion equations in the generalized coordinates.

For the assumed analogous linear finite element model the mathematical description of its motion has the classical form of a set of coupled ordinary differential equations in general coordinates, which can be found e.g. in [2].

In order to develop a proper control algorithm for the given vibrating drive system the electromagnetic external excitation produced by the motor should be described possibly accurately. Thus, the electromechanical coupling between the electric motor and the torsional train ought to be taken into consideration. In the considered case of the symmetrical three-phase asynchronous motor, electric current oscillations in its windings are described by six voltage equations, transformed next into the system of four Park's equations in the so called '$\alpha\beta$-*dq*' reference system, form of which can be found e.g. in [3]. Then, the electromagnetic torque generated by such a motor can be expressed by the following formula

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

where M denotes the relative rotor-to-stator coil inductance, p is 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, [3].

From the abovementioned system of voltage equations as well as from formula (2) it follows that the coupling between the electric and the mechanical system is non-linear in character, which leads to very complicated analytical description resulting in rather harmful computer implementation. Thus, this electromechanical coupling has been realized here by means of the step-by-step numerical extrapolation technique, which for relatively small direct integration steps for motion equations results in very effective, stable and reliable computer simulation.

4. Computational and experimental example

The experimental investigations are going to be carried out by means of the described above test-rig equipped with the proper measurement-control system, the scheme of which is presented in Fig. 3. This system consists of the voltage amplifier controlled by the real-time computer using the appropriate converting system. Such measurement-control system enables us to monitor and register all results of measurements using the control-communication unit by means of the TCP/IP protocol. Basing on the obtained on-line measurement results of the dynamic torques transmitted by the shown in Fig 1 shaft segments adjacent to the torque-meters, the properly developed control algorithm determines in real time the current values of damping coefficients of the magneto-rheological fluid in both rotary actuators.

The measurement results of dynamic torsional responses have been registered for the steady-state operating conditions at constant nominal rotational speeds, respectively for the passive (without control) and semi-actively controlled drive system excited by the harmonic fluctuating component of the retarding torque within the frequency range 0-150 Hz. Fig. 4 presents exemplary time-histories obtained for the passive system (grey line) and the semi-actively controlled one (black line), both for the excitation frequency 54 Hz corresponding to the first natural system frequency. In Fig. 5 there are shown plots of dynamic response amplitudes of the passive (grey line) and semi-active system (black line) determined by means of measurements, Figs. 4a, and by numerical simulations, Fig. 4b.

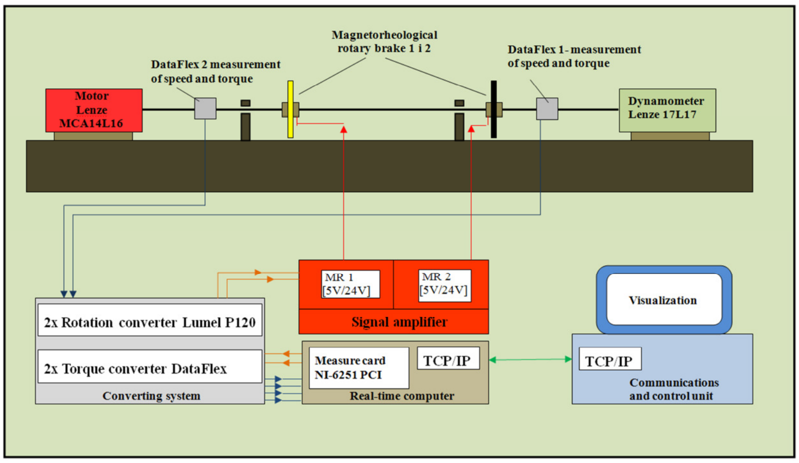

Figure 3. Scheme of the test-rig measurement system

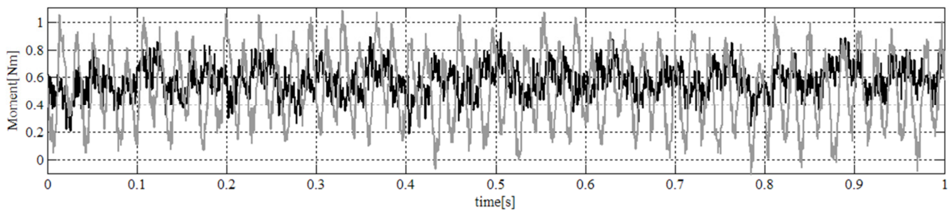

Figure 4. Measured time-histories of the dynamic torque transmitted by the input-shaft

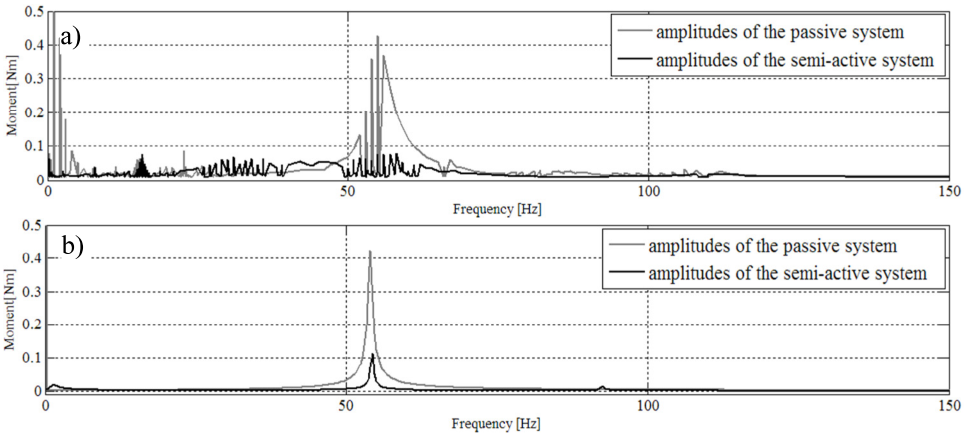

Figure 5. Amplitude characteristics of the dynamic responses of the passive (grey line) and semi-active (black line) system obtained using experiment (a) and simulation (b)

From the experimentally and theoretically obtained plots it follows that the rotary actuators with the magneto-rheological fluid can effectively suppress torsional vibration level, particularly for the resonance oscillation frequencies, e.g. corresponding to the first, fundamental eigenmode, for the properly selected control voltage value based on the respective minimum of the frequency response function determined for the considered mechanical system.

5. Conclusions

In the paper a semi-active control of steady-state torsional vibrations of the laboratory drive system driven by the asynchronous motor has been experimentally and computationally performed by means of rotary actuators with the magneto-rheological fluid (MRF). As it follows from the measurement and numerical examples, in both cases the optimum control carried out by means of the applied actuators with the MRF can effectively reduce the steady-state vibrations of the successive shaft segments to the quasi-static level of the loading transmitted by the drive system, where dynamic amplifications of the responses due to resonance effects have been almost completely suppressed.

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