Semi-active vibration control using a rotary magnetorheological damper – experimental verification

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Abstract: From all kinds of vibrations, torsional ones are very important as naturally associated with fundamental rotational drive system motion. The aim of this paper is to present semi-active control of torsional vibrations of the working machine drive systems, which is performed using a rotary damper with the magnetorheological fluid. The main purpose of this research is minimization of vibration amplitudes in order to increase fatigue durability of the most responsible machine elements. The special control strategies are proposed for steady-state torsional vibrations suppression. The semi-active control method is based on a principle of on-line selection of optimum damping coefficient values, which is realized by application of the magnetorheological fluid. The analysis performed in the paper combines experimental verification using the laboratory test rig with theoretical computations. This experiment was designed in the form of the laboratory drive system co-operating with two asynchronous motors generating properly programmed driving and retarding electromagnetic torques.

Keywords: semi-active control, torsional vibrations, rotary dampers, magnetorheological fluid, electromechanical drive system

1. Introduction

Active vibration control opens new opportunities to improve reliability and efficiency of machines drive system. Torsional vibrations are kind of vibration, which are very important, because they are naturally related with the basic rotational motion. The torsional vibrations create huge problems in the control associated not only with proper control torque generation, but also with convenient ways of imposing the control torques on quickly rotating drive system parts. 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 piezoelectric actuators. But in such cases relatively small values of control torques can be generated and thus the piezoelectric actuators can be usually applied to low-power drive systems. In [2, 3] there is proposed the semi-active control technique based on rotary dampers with the magnetorheological fluid. In such devices the damping effect is created due to friction in the magnetorheological fluid film between the freely rotating inertial ring and hub attached directly to the torsionally vibrating shaft. Due to properties of magnetorheological fluid, they have ability to realize adjustable viscosity. Such devices 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 main purpose of this paper is an experimental verification of the presented in [2, 3] theoretical concept of semi-active control. The investigated object is the laboratory machine drive system equipped with one rotary damper with the magnetorheological fluid. Thus, for this purpose the appropriate 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.

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

In the considered laboratory drive system, which is imitating operation of the working rotating machine, power is transmitted from the servo-asynchronous motor to the driven machine tool in the form of electric brake using two multi-disk elastic couplings with built-in torque-meters, electromagnetic overload coupling and by the shaft segments.

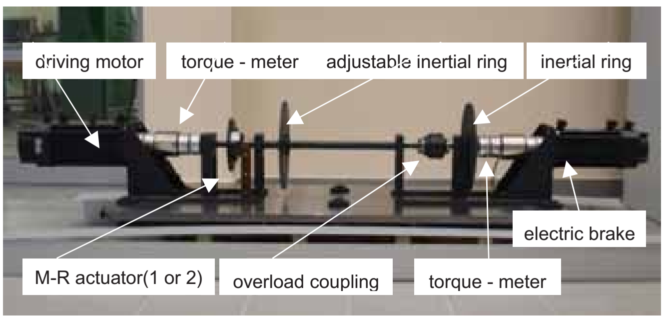


Fig. 1. The laboratory drive system
Rys. 1. Laboratoryjny układ napędowy

The schematic diagram of structure of the real damper installed in the test-rig is presented in fig. 2. Using special sliders control voltage is applied to the magnetorheological damper. External magnetic field acts on the fluid inside the damper, which results in adjustment of characteristics of the magnetorheological fluid, so as to provide the control of torsional vibrations process. Since the average rotational speeds of the rings and of the shaft are similar, only small wearing effects can be expected and vibrations can be suppressed without influencing much the rigid body motion of the drive system.

In order to perform a theoretical investigation of the semi-active control, which can be applied for this mechanical system, reliable and computationally efficient mechanical models are required. In this paper dynamic investigations of the entire drive system are performed using two independent structural models. Those models consist of torsionally deformable one dimensional beam-type finite elements and rigid bodies, as shown in fig 3.

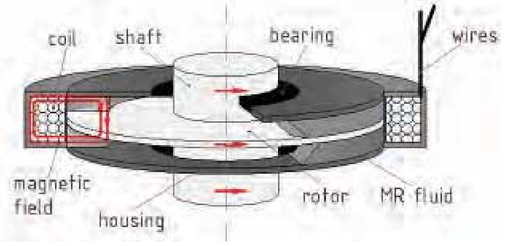
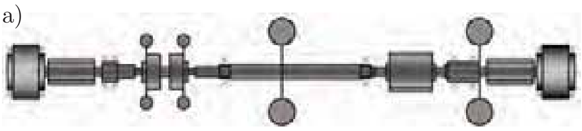


Fig. 2. Rotary damper with the magnetorheological fluid
Rys. 2. Tłumik obrotowy z cieczą magnetoreologiczną

a)



b)

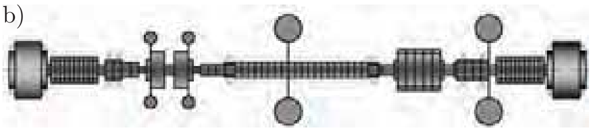


Fig. 3. Mechanical model of laboratory test rig (a: discrete-continuous model and b: the classical beam finite element)
Rys. 3. Mechaniczne modele laboratoryjnego układu napędowego (a: model dyskretno-ciągły, b: klasyczny model MES)

For the purpose of theoretical investigation are used discrete-continuous model and the classical beam finite element (FEM) one. Both models are employed here for eigenvalue analyses as well as for numerical simulations of test-rig torsional vibrations. In the hybrid model, each of cylindrical segments of the stepped rotor-shaft is substituted by the cylindrical macro-element with continuously distributed inertial-visco-elastic properties. However, in case of the finite element model, these continuous macro-elements have been discretized with proper mesh density to assure 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 the 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 Lagrange coordinates. In the assumed model the control damping torques can be regarded as the response-dependent external excitations. Then, by a transformation of them into the space of modal coordinates and upon proper rearrangements the following set of coupled modal equations is yielded:

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

$$\mathbf{D}(\dot{\mathbf{r}}(t))=\mathbf{D}_0+\mathbf{D}_\mathrm{C}(k_j(t),\dot{\mathbf{r}}(t)),\quad j=1{,}2. \tag{1}$$

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 $\mathbf{D}_\mathrm{C}(k_j(t),\dot{\mathbf{r}}(t))$ 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 $\mathbf{D}$ 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 $6\div10$ 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 generalized coordinates, which can be found e.g. in [2].

Since in the investigations, during start-ups and steady-state operation the drive system can be affected by external excitations that are unknown in advance, any effective and practically usable control strategy has to be passive or closed-loop. According to the above, it is assumed here that the feedbacks are the current frequency spectra $\omega_e(f, t)$ and $\omega_r(f, t)$ of the electric and the retarding torques, which can be estimated on-line from the current slip values determined form measurements of the rotational velocities of the appropriate drive system shaft cross-sections. The locally optimum control functions $k_j(t)$ are determined with respect to the frequency response functions $\mathrm{FRF}_e(f, \mathrm{k})$ and $\mathrm{FRF}_r(f, \mathrm{k})$ of the damped drive system excited respectively by the electric motor and by the power receiver. In this paper semi-active control, for which the damping coefficients remain constant during the whole run-up process and the steady-state operation, is proposed. Here, their values are optimum with respect to the frequency response function of the damped drive system excited by the power receiver, as defined by:

$$\mathbf{k}_0 = \arg\min_{\mathbf{k}} \max_{f} \mathrm{FRF}_r(f, \mathbf{k}). \tag{2}$$

This control strategy is going to be tested numerically, and experimentally verified below.

1.2. Modeling of the electrical external excitation generated by the servo-asynchronous motor

In order to develop a proper control algorithm for the given torsionally vibrating drive system, the electromagnetic external excitation produced by the driving motor should be described possibly accurately. In this way, 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 [4, 5]. Then, the electromagnetic torque generated by such a motor can be expressed by the following formula

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

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, [4, 5].

From the aforementioned system of voltage equations as well as from formula (3), it follows that the coupling between the electric and the mechanical system is nonlinear in character, which leads to very complicated analytical description resulting in rather difficult 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.

In the computational examples there are investigated start-ups and following after them steady-state operation of the considered laboratory test-rig presented in fig. 1. This system is accelerated from a standstill to the nominal operating conditions characterized by the rated retarding torque $M_n = 12$ Nm at the constant rotational speed 1680 rpm. The retarding torque produced by the working machine has been assumed as linearly proportional to the current shaft rotational speed with a superimposed stepwise fluctuation component of also velocity dependent amplitude:

$$M_r(\Omega_r(t)) = \frac{M_n}{\Omega_n} \Omega_r(t) \cdot \big(1 + h \cdot \mathrm{sgn}\big(\sin(\Omega_r(t))\big)\big), \tag{4}$$

where: Ω_n denotes the nominal angular velocity, $\Omega_r(t)$ is the current angular velocity of the power receiver working tool (this angular velocity plays a role of the excitation frequency), h denotes the step fluctuation ratio. Temporary negative values of the retarding torque $M_r(t)$ is assumed to be equal to zero. The retarding torque (4) as well as the electromagnetic torque (3) generated by the servo-asynchronous motor are assumed to be uniformly distributed along the mechanical model elements representing the machine working tool and the motor rotor, respectively.

2. Experimental verification

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. 4. This system consists of the voltage amplifier controlled by the real-time computer using the appropriate converting system and one active element – the damper with the magnetorheological fluid. Such measurement-control system enables us to monitor and register all measurement results. This is possible using the control-communication unit by means of the TCP/IP protocol.

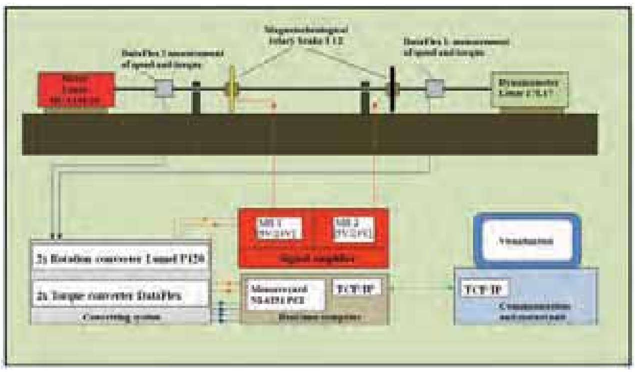


Fig. 4. Measurement system scheme of the test-rig
Rys. 4. Schemat układu pomiarowego stanowiska badawczego

Measurements are carried out with frequencies above 15 kHz. This allows accurate detection of even very rapid changes of the torques. The torques are measured in the non-contact way. The measurements are performed in bandwidth of 300 kHz. Simultaneously, the real-time computer carries out the FFT analysis from the time intervals of one second and records the values of the two major peaks of the FFT. This allows the observer to record the value of the dominant peak, depending on the value of control signal on voltage amplifiers. On the basis of the on-line measurement results of the dynamic torques transmitted by the shown in fig. 1 shaft segments adjacent to the torque meters, the appropriately designed control algorithm determines in real time the current values of damping coefficients of the magnetorheological fluid in rotary damper.

The electromagnetic torque generated by the servo-asynchronous motor is determined using formula (3) during numerical simulations of the start-up and steady-state operation process regarding torsional vibrations of the mechanical system mutually coupled with oscillations of electrical currents in the motor windings. For the retarding torque described by (4) there was assumed $h = 1$, which implies its fluctuation amplitude two times greater than the average retarding torque value. The time history plots of the driving and retarding torques normalized by the rated torque M_n during the start-up and the beginning of nominal operation are illustrated in fig. 5 by the blue and red lines, respectively.

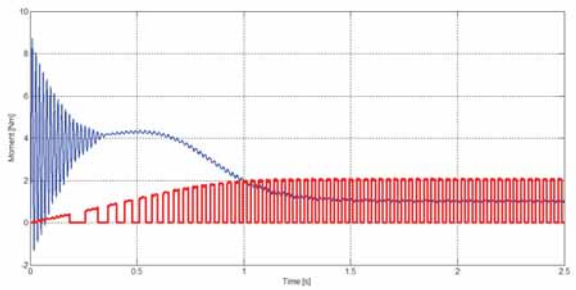


Fig. 5. An example of the time course of electromagnetic (red line) and retarding torque (blue line)

Fig. 5. Przykładowy przebieg momentu napędowego (linia niebieska) i oporowego (linia czerwona)

From these plots it follows that the initial value of the electromagnetic torque generated by this motor is 3.9 times greater than the nominal torque $M_n = 12$ Nm, the maximum quasi-static value of this torque exceeds M_n ~4.65 times and the initial amplitude of the decaying with time dynamic component oscillating with frequency 60 Hz is 5.8 times greater than M_n.

2.1. Measurement and numerical results

The measurement results of dynamic torsional responses have been registered for the steady-state operating conditions at constant nominal rotational speeds. The measurements are performed respectively for the passive object (without control) and semi-actively controlled drive system, both excited by the harmonic fluctuating component of the retarding torque within the frequency range 0–100 Hz. Fig. 6 presents exemplary time-histories obtained for the passive system (blue line) and the semi-actively controlled one (red line), both for the excitation frequency 54 Hz corresponding to the first natural laboratory system frequency. In fig. 7 there are shown plots of dynamic response amplitudes of the passive object (blue line) and semi-active system (red line) determined by means of measurements, fig. 7a, and by numerical simulations, fig. 7b.

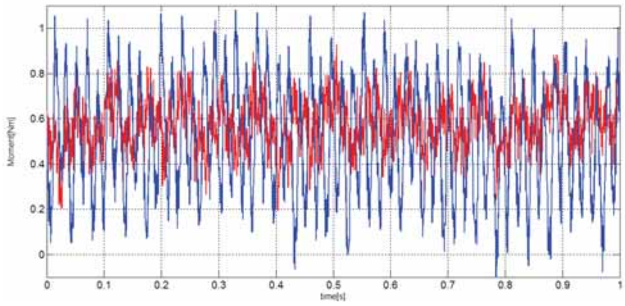


Fig. 6. Measured time-histories of the dynamic torque transmitted by the input-shaft

Rys. 6. Zmierzone przebiegi dynamicznego momentu skręcającego w funkcji czasu

a)

b)

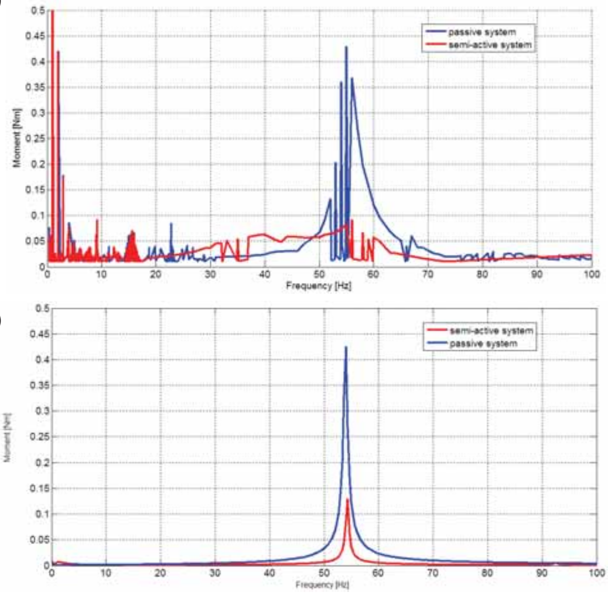


Fig. 7. Measurement and numerical results – amplitude characteristics of the test rig dynamic responses (passive – blue line, semi-actives – red line)

Rys. 7. Charakterystyki amplitudowe odpowiedzi dynamicznej układu pasywnego – linia niebieska i półaktywnego – linia czerwona (a – zmierzone, b – wyznaczone numerycznie)

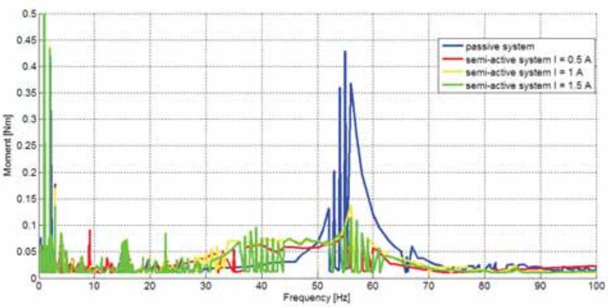


Fig. 8. Measurement and numerical results – amplitude characteristics of the test rig dynamic responses (0 A – blue line, 0.5 A – red line, 1 A – yellow line, 1.5 A green line).

Rys. 8. Charakterystyki amplitudowe odpowiedzi dynamicznej układu pasywnego – linia niebieska i półaktywnego – linia czerwona przy 0,5 A, linia zółta 1,0 A a linia zielona przy 1,5 A)

This research also enabled us to determine the optimal control current value for the magnetorheological damper control signal. Then, for greater control current values, worse dissipation effects realized by the rotary damper are observed, presented in fig. 8. In this case the optimal current value is 0.5 A.

3. Conclusions

In the paper a semi-active control of steady-state torsional vibrations of the laboratory test rig driven by the asynchronous motor, has been experimentally and computationally verified using the rotary damper with the magnetorheological fluid. As it follows from the measurement and numerical examples, in both cases the optimum control carried out by means of the applied damper with the magnetorheological fluid can effectively reduce the steady-state torsional 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. In the next step of research in this field, other control strategies are going to be experimentally and theoretically verified. In future also a larger number of magnetorheological dampers are expected to be used in laboratory and theoretical investigations.

References

1. Przybyłowicz P.M., *Torsional vibration control by active piezoelectric system*, "J. of Theoretical and Applied Mechanics", 33(4), 1995, 809–823.
2. Szolc T., Jankowski Ł., Pochanke A., Magdziak A., *An application of the magneto-rheological actuators to torsional vibration control of the rotating electro-mechanical systems*, The 8th IFToMM International Conference of Rotor Dynamics, 12–15.09.2010, KIST, 488–495.
3. Pręgowska A., Konowrocki R., Szolc T., *Experimental verification of the semi-active control concepts for torsional vibrations of the electro-mechanical system using rotary magneto-rheological actuators*, Proc. of XXV Symposium on Vibrations in physical systems, 2012, 329–334.
4. White D.C., Woodson H.H., *Electromechanical energy conversion*, Wiley, New York 1959.
5. Laschet A., *Simulation von Antriebssystemen*, Springer-Verlag, Berlin, London, New York, Paris, Tokyo 1988.

Półaktywne sterowanie drganiami skrętnymi przy wykorzystaniu obrotowego tłumika z cieczą magnetoreologiczną – weryfikacja doświadczalna

Streszczenie: Drgania skrętne są jednym z najważniejszych rodzajów drgań, ponieważ są naturalnie związane z podstawowym ruchem roboczym wielu maszyn i urządzeń. Prezentowany artykuł jest poświęcony półaktywnemu sterowaniu drganiami skrętnymi napędu maszyny roboczej przy wykorzystaniu obrotowych tłumików z cieczą magnetoreologiczną o zmiennych właściwościach dyssypacyjnych. Głównym celem prezentowanych badań jest zmniejszenie poziomu amplitudy drgań skrętnych w celu zwiększenia trwałości zmęczeniowej najbardziej odpowiedzialnych elementów maszyny roboczej. Półaktywne sterowanie sprowadza się do wyznaczenia optymalnej wartości współczynnika tłumienia, który jest realizowany przy wykorzystaniu właściwości filmu z cieczy magnetoreologicznej. Teoretyczna koncepcja sterowania została doświadczalnie zweryfikowana za pomocą stanowiska doświadczalnego w postaci laboratoryjnego układu napędowego, na którym zmienne napędowe i oporowe momenty wymuszające są generowane odpowiednio przez silnik i hamownicę asynchroniczną.

Słowa kluczowe: półaktywne sterowanie, drgania skrętne, obrotowe tłumiki z cieczą magneto reologiczną

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