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Semi-active control of torsional vibrations of drive systems by means of actuators with the magneto-rheological fluid

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ABSTRACT

In the paper control of transient torsional vibrations induced by the electric motor during run-ups of the radial compressor drive system is performed by means of 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. The theoretical investigations are based on a hybrid structural model of the vibrating mechanical system as well as on sensitivity analysis of the response with respect to the actuator damping characteristics. For suppression of transient torsional vibrations excited by electro-magnetic torques generated by the asynchronous and synchronous motor there are proposed various control strategies based on actuators in the form of control brakes and control clutches.

KEY WORDS

Semi-active control, transient vibrations, drive system, electric motor, control brakes and control clutches, magneto-rheological fluid

1 INTRODUCTION

Transient torsional vibrations due to start-ups of drive trains driven by electric motors are very dangerous for material fatigue of the most heavily affected and responsible elements of these mechanical systems. Thus, this problem has been considered for many years by many authors, e.g. in [1-3]. Nevertheless, till present majority of these studies are reduced to possibly accurate standard transient vibration analyses taking into consideration additional dynamic effects caused by elastic couplings, dry friction in clutches, properties of the gear stage meshings, e.g. backlash, electro-mechanical couplings and others.

Control of torsional vibrations occurring in the drive systems could effectively minimise material fatigue and in this way it would enable us to increase their operational reliability and durability, e.g. in the form of greater number of admissible safe run-ups and run-outs. Unfortunately, one can find not so many published results of research in this field, beyond some attempts performed by active control of shaft torsional vibrations using piezo-electric actuators, [4]. 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. Moreover, even if a relatively big number of the piezo-electric actuators are attached to the rotor-shafts of the entire drive system, as it follows e.g. from [4], only higher eigenmodes can be controlled, whereas control of the most important fundamental eigenmodes is often not sufficiently effective.

In this paper, the control is going to be realised by brakes and clutches with the magneto-rheological fluid (MRF) applied to attenuate transient torsional vibrations excited during start-ups of the large radial compressor drive system driven by the asynchronous and synchronous motor. The magneto-rheological fluids are functional fluids whose effective viscosity depends on externally provided magnetic field. This feature makes them perfectly suitable for large brakes and clutches with controllable damping characteristic. Besides the ability to generate large damping torques, an important advantage of the MRF-based devices is a low power consumption. External power is needed to supply the electromagnetic coils only, i.e. to modify the dynamic characteristics of the mechanical system, which is the distinguishing feature of the semi-active control. Moreover, the semi-active damping-based approach eliminates a risk of causing instability, which is intrinsically related to the active control paradigm and it can occur in the case of an electrical failure, unexpected control time delays or in the case of an inaccurate modelling. In addition, it should be remarked that as demonstrated in [6], there exist effective control methodologies based on actuators with the MRF, which are characterized by actual control delays reaching few milliseconds only. Thus, this technique enables us to control effectively vibrations of frequency exceeding 100 Hz or more.

2 ASSUMPTIONS FOR THE SIMULATION MODEL AND FORMULATION OF THE PROBLEM

In the considered drive system of the radial compressor, power is transmitted form the electric motor to the impeller by means of the low-speed and high-speed rigid couplings, multiplication single-stage gear and shaft segments, as shown in Fig. 1a.

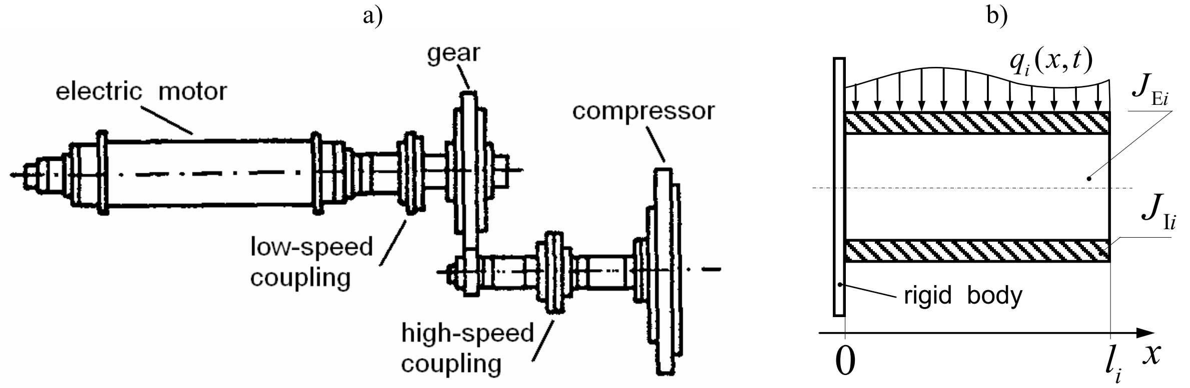

Figure 1: Mechanical model of the compressor drive system (a), visco-elastic continuous macro-element (b).

In order to perform a theoretical investigation of the semi-active control concepts applied for this mechanical system, a reliable and computationally efficient simulation model is required. In this paper dynamic investigations of the entire drive system are performed by means of the one-dimensional hybrid structural model consisting of continuous visco-elastic macro-elements and rigid bodies. This model is employed here for eigenvalue analyses as well as for numerical simulations of torsional vibrations of the drive train. In this model successive cylindrical segments of the stepped rotor-shaft are substituted by torsionally deformable cylindrical macro-elements of continuously distributed inertial-visco-elastic properties, as presented in Fig. 1b. Since in the real drive system of the compressor the electric motor coils and gears are attached along some rotor-shaft segments by means of shrink-fit connections, the entire inertia of such fragments is increased, whereas usually the shaft cross-sections only are affected by elastic deformations due to transmitted loadings. Thus, the corresponding visco-elastic macro-elements in the hybrid model must be characterized by the geometric cross-sectional polar moments of inertia J_{Ei} responsible for their elastic and inertial properties as well as by the separate layers of the polar moments of inertia J_{Ii} responsible for their inertial properties only, i=1,2,…,n, where n is the total number of macro-elements in the considered hybrid model, Fig. 1b. Moreover, on the actual operation temperature T_i can depend values of Kirchhhoff's moduli G_i of the rotor-shaft material of density ρ for each i-th macro-element representing given rotor-shaft segment. In the proposed hybrid model of the compressor drive system inertias of the impeller and gears 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 is governed by the hyperbolic partial differential equations of the wave type

$$G_i(T_i)J_{Ei}\left(1+\tau\frac{\partial}{\partial t}\right)\frac{\partial^2\theta_i(x,t)}{\partial x^2}-c_i\frac{\partial\theta_i(x,t)}{\partial t}-\rho\left(J_{Ei}+J_{Ii}\right)\frac{\partial^2\theta_i(x,t)}{\partial t^2}=q_i(x,t), \qquad (1)$$

where $\theta_i(x,t)$ is the angular displacement with respect to the shaft rotation with the average angular velocity Ω, τ denotes the retardation time in the Voigt model of material damping and c_i is the coefficient of external (absolute) damping. The external active, passive and control torques are continuously distributed along the respective macro-elements of the lengths l_i. These torques are described by the two-argument function $q_i(x,t)$, where x is the spatial co-ordinate and t denotes time.

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 for $x=L_i=l_1+l_2+...+l_{i-1}$ of the adjacent (i-1)-th and the i-th elastic macro-elements:

$$\theta_{i-1}(x,t)=\theta_i(x,t) \qquad \text{for } x=L_i. \tag{2a}$$

The second group of boundary conditions are dynamic ones, which contain linear equations of equilibrium for external and control torques as well as for inertial, elastic and external damping moments. For example, the dynamic boundary condition describing a simple connection of the mentioned adjacent (i-1)-th and the i-th elastic macro-elements has the following form:

$$M_i(t)-I_{0i}\frac{\partial^2\theta_i}{\partial t^2}-M_i^{\mathrm{D}}(t)-G_{i-1}(T_{i-1})J_{\mathrm{E},i-1}\left(1+\tau\frac{\partial}{\partial t}\right)\frac{\partial\theta_{i-1}}{\partial x}+G_i(T_i)J_{\mathrm{E}i}\left(1+\tau\frac{\partial}{\partial t}\right)\frac{\partial\theta_i}{\partial x}=0 \tag{2b}$$

$$\text{for } x=L_i, \quad i=2,3,...,n,$$

where $M_i(t)$ denotes the external concentrated torque, I_{0i} is the mass polar moment of inertia of the rigid body and $M_i^{\mathrm{D}}(t)$ denotes the control damping torque.

In order to perform an analysis of natural elastic vibrations, all the forcing and viscous terms in the motion equations (1) and boundary conditions (2b) have been omitted. An application of the solution of variable separation for Eqs. (1) leads to the following characteristic equation for the considered eigenvalue problem:

$$\mathbf{C}(\omega)\cdot\mathbf{D}=\mathbf{0}, \tag{3}$$

where $\mathbf{C}(\omega)$ is the real characteristic matrix and $\mathbf{D}$ denotes the vector of unknown constant coefficients in the analytical local eigenfuctions of each i-th macroelement. Thus, the determination of natural frequencies reduces to the search for values of ω, for which the characteristic determinant of matrix $\mathbf{C}$ is equal to zero. Then, the torsional eigenmode functions are obtained by solving equation (3).

The solution for forced vibration analysis has been obtained using the analytical - computational approach. Solving the differential eigenvalue problem (1)÷(3) 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 $\xi_m(t)$:

$$\ddot{\xi}_m(t)+(\beta+\tau\omega_m^2)\dot{\xi}_m(t)+\omega_m^2\xi_m(t)=\frac{1}{\gamma_m^2}Q_m(t), \quad m=1,2,... , \tag{4}$$

where ω_m are the successive natural frequencies of the drive system, β denotes the coefficient of external damping assumed here as proportional damping to the modal masses γ_m^2 and $Q_m(t)$ are the modal external excitations. Although each Eq. (4) has its analytical solution, it can be also solved numerically using a direct integration in order to obtain transient torsional response for the passive and the controlled system.

3 CONTROL CONCEPTS OF THE TRANSIENT TORSIONAL VIBRATIONS

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 the quickly rotating parts of the drive system. In this paper two actuating techniques are proposed: The first one applies actuators in the form of control brakes similar to the journal bearings. But in the considered case, instead of the oil film between the shaft and the motionless bushing, the MRF of variable viscosity is used. The second actuating technique, based on the so called control clutches, is realized in an analogous way to the known torsional vibration viscous dampers installed on the reciprocating engine crankshafts. But here, instead of the silicon oil between the shaft and the inertial ring, 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 consists of the average component corresponding to the rigid body motion and of the fluctuating component caused by the torsional vibrations. The optimal control should effectively suppress vibrations without influencing much the rigid body motion of the drive system. In order to satisfy these requirements, some consideration on a dynamic behaviour of the passive drive system during run-ups is necessary. As it follows e.g. from [5], maximal fluctuation of the vibratory component of the shaft rotational speed during run-ups by means of the asynchronous motor occurs at the beginning of the process, when the average rotational speed is relatively small. Then, the actuators of both types can be used to suppress the predominant vibratory component of the shaft rotational speed. However, in the case of run-ups using the synchronous motor, as it follows e.g. from [1,5], the external excitation frequency decreases proportionally to the slip from the double network frequency 100 Hz to zero. Then, the drive system eigenvibration forms of natural frequencies contained in this range can be excited, which leads to transient resonances resulting in severe vibration amplitudes becoming here a target of control. But during run-ups, these resonances usually occur at relatively large average rotational speeds of the drive system. In such cases, the control brakes could even increase the system oscillations by retarding the rigid body motion during run-ups and by keeping longer in this way the slip and the resonant external excitation frequency almost constant, which would lead to a greater fatigue effort of the shaft material and to bigger energy consumption. Thus, in order to avoid these disadvantages, for the system driven by the synchronous motor the actuators in the form of control clutches with the freely rotating inertial rings are much more convenient. Basing on these design solutions of the actuators, various control concepts are proposed.

3.1 Actuators in the form of control brakes

In this case, transient torsional vibrations are going to be suppressed by means of control brakes with the MRF. Assume there are N controllable brakes, each with the damping coefficient $c_j k(t), j=1,2,\ldots,N$, where $k(t)$ is the collective control function and c_j are the brake-specific multipliers. Each brake generates the damping torque

$$M_j^{\mathrm{D}}(t) = -c_j k(t)\Omega(x_j,t) = -c_j k(t)\left[\Omega(t) + \sum_{m=1}^{\infty}\dot{\theta}_m(x_j,t)\right], \quad j = 1,2,\ldots,N, \tag{5}$$

where x_j is the location of the j-th brake. The damping torques $M_j^{\mathrm{D}}(t)$ modify Eqs. (4) by coupling them with each other and with the equation of the rigid body shaft motion. However, by proper determination of the multipliers c_j, the most resonant mode can be decoupled from the influence of the average angular velocity $\Omega(t)$.

The optimum evolution of the control variable $k(t)$ can be determined with respect to the two following objectives:

1. Maximization of the effectiveness of the damping, which is quantified here as the mean square torque above a given safe level. In practice, this objective can be related to the most resonant r-th eigenvibration mode:

$$F_1\left[k(t)\right] = \int \max\left(0, \left|\xi_r(t)\right| - \xi_{r(\mathrm{safe})}\right)^2 \mathrm{d}\,t, \tag{6}$$

   where $\xi_{r(\mathrm{safe})}$ denotes the modal coordinate value yielding the safe level of tangential stress in the most heavily affected shaft.

2. Minimization of the energy dissipated due to the damping:

$$F_2\left[k(t)\right] = \sum_{j=1}^{N} \int M_j^{\mathrm{D}}(t)\omega(x_j,t)\,\mathrm{d}\,t = \sum_{j=1}^{N} \int M_j^{\mathrm{D}}(t)\left[\Omega(t) + \sum_{m=1}^{\infty}\dot{\theta}_m(x_j,t)\right]^2 \mathrm{d}\,t. \tag{7}$$

In practice, these two objectives are contradictory, hence the following compound weighted objective has been used:

$$F_\alpha\left[k(t)\right] = \alpha F_1\left[k(t)\right] + (1-\alpha)F_2\left[k(t)\right], \tag{8}$$

where α is the weighting coefficient used to balance the influence of F_1 and F_2.

The unknown to be optimized is the collective control function $k(t)$. However, in order to avoid the variational formulation of the optimization problem, $k(t)$ can be made dependent on a finite number of parameters. Then, the compound objective function (8) can be minimized using standard numerical approaches. In this paper it is assumed that $k(t)$ is a linear combination of a finite set of base functions, $k(t) = \sum_{i=1}^{K}\hat{k}_i k_i(t)$, which makes it dependent on a finite number K of parameters $\hat{k}_i$ and thus suitable for a constrained optimization using standard numerical techniques. A proper choice of the constraints and of the base functions allows also the MRF-specific damping rise and decay rate restrictions to be satisfied.

Note that the described procedure yields the globally optimum open-loop control. It can later serve as the best reference for possible closed-loop control laws. However, in the considered below case of the asynchronous motor, as illustrated in the computational example, the globally optimum control is a simple hold-and-release strategy, which can be easily realized using an open-loop control.

3.2 Actuators in the form of control clutches

Here, transient torsional vibrations are going to be suppressed by means of control clutches attenuating the difference between the vibratory and the average rotor-shaft motion. Thus, assume that there are N controllable clutches with freely rotating inertial rings, each with the independently controllable damping coefficient $k_j(t)$, j=1,2,…,N. Each clutch generates the following damping torque

$$M_j^D(t) = -k_j(t)\Delta\omega_j(t) = -k_j(t)\left[\Omega(t) + \sum_{m=1}^{\infty}\dot{\theta}_m(x_j,t) - \omega_j(t)\right], \quad j = 1,2,\ldots,N. \tag{9}$$

where x_j is the location of the j-th clutch and $\omega_j(t)$ is the rotational speed of the jth inertial ring, which obeys

$$J_j\dot{\omega}_j(t) = -k_j(t)\left[\omega_j(t) - \Omega(t) - \sum_{m=1}^{\infty}\dot{\theta}_m\left(x_j,t\right)\right]. \tag{10}$$

As mentioned above, in the case of a synchronous motor the external excitation frequency is time-dependent and proportional to the slip. Hence, any effective and practically usable control strategy has to be passive or closed-loop. According to the above, it is assumed here that the feed-back control variable is the excitation frequency $f(t)$, which can be possibly accurately estimated by the current slip value determined form measurements of the rotational velocities of the appropriate several rotor-shaft cross-sections. Then, the locally optimum closed-loop control functions $k_j(t)$ can be determined with respect to the frequency response function (FRF) of the modal equations of motion (4) augmented by the control torques (9) and the equations of motion of the inertial rings (10). In this study, two basic control modes are applied:

1. Active control, which reduces to the closed-loop with respect to the external excitation frequency $f(t)$. The optimum value $\mathbf{k}_0(t)$ of the vector of the damping coefficients $k_j(t)$ can be determined as

$$\mathbf{k}_0(t) = \arg\min_{\mathbf{k}} \mathrm{FRF}(f(t),\mathbf{k}). \tag{11}$$

In practice, there are inevitable control delays, which can be numerically modelled by considering the equation

$$\dot{\mathbf{k}}(t) = d\cdot\left\lfloor \mathbf{k}_0(t) - \mathbf{k}(t)\right\rfloor, \tag{12}$$

where d is the given coefficient describing a control retardation.

2. Passive control, for which the damping coefficients remain constant during the whole run-up process. However, their values are optimum with respect to the resonant frequencies, as defined by

$$\mathbf{k}_0 = \arg\min_{\mathbf{k}} \max_f \mathrm{FRF}(f,\mathbf{k}). \tag{13}$$

Both control modes have been tested numerically, as described in the following section.

4 COMPUTATIONAL EXAMPLES

In the computational example, start-ups of a large radial compressor driven by the asynchronous and the synchronous motor of the same nominal power 5 MW are investigated. This system presented schematically in Fig. 1a is accelerated from a standstill to the nominal operating conditions characterized by the rated retarding torque 31831 Nm at the constant rotational speed 1500 rpm. The values of these quantities are reduced to the motor shaft, where the impeller rotational speed is 4.932 times multiplied by the gear stage. The retarding aerodynamic torque produced by the compressor is described by a parabolic function, as proportional to the square of the impeller actual angular velocity.

The qualitative dynamic properties of the considered drive system have been determined first in the form of an eigenvalue analysis by solving equation (3). In Fig. 2 there are depicted the lowest for this system first four eigenfunctions together with the corresponding natural frequency values. From the viewpoint of vibration

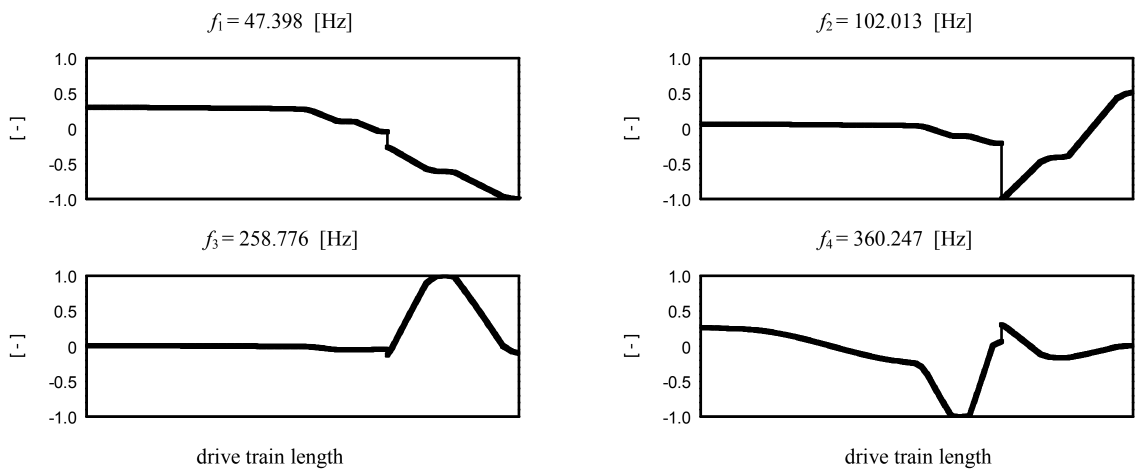

Figure 2: Eigenfuctions with the corresponding natural frequencies of the drive system.

control in the range of the excitation frequencies of interest, the first eigenform of frequency 47.4 Hz seems to be the most important. First of all, this natural frequency value is very close to the synchronous network excitation frequency 50 Hz generated by both considered electric motors. Moreover, the first eigenfuction is characterized by the greatest modal displacements corresponding to the positions of the dynamic excitation source, i.e. the driving motor, and the compressor impeller, which is very essential for imposing the control torques on the vibrating system. The natural frequency of the second eigenform equals to 102.013 Hz and is also close to the maximum excitation frequency 100 Hz generated by the synchronous motor. But as if follows from Fig. 2, its eigenfunction is almost insensitive to the excitation by the motor, since the modal displacements corresponding to the motor positions are close to zero. According to the above, in each considered case of semi-active control, N=3 actuators have been applied: two located on both sides of the motor, i.e. on the left- and right-hand side of the dynamic excitation source, and one near the compressor impeller.

4.1 Run-up by means of the asynchronous motor

The electromagnetic torque generated by the asynchronous motor has been *a priori* assumed according to the respective relations contained in [5]. The time history plots of the normalized driving and retarding torques during the run-up are illustrated in Fig. 3a by the black and grey lines, respectively.

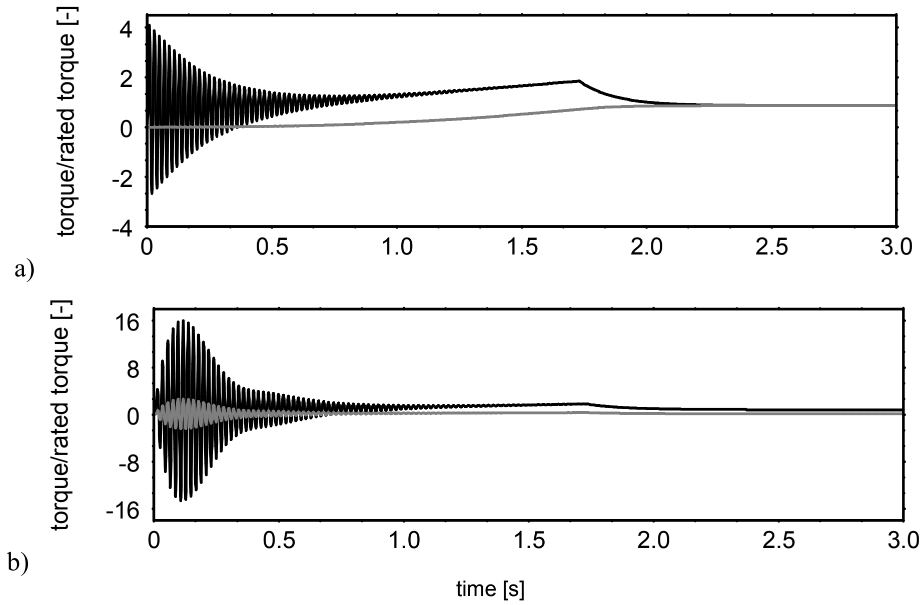

Figure 3: Time histories of the driving and retarding torques (a) and of the elastic torques (b).

The passive system transient dynamic response caused by this start-up is presented in Fig. 3b in the form of the time histories of the normalized elastic torques transmitted by the shafts in the vicinity of the low-speed coupling (black line) and the high-speed coupling (grey line). From these plots it follows that the transient component of the asynchronous motor torque in the form of an attenuated sinusoid of the network frequency 50 Hz and the initial amplitude ca. 3.5 times greater than the rated torque value, Fig. 3a, induces very severe resonance with the first system eigenvibration mode of frequency 47.4 Hz. The maximum amplitudes of the most heavily affected shaft close to the low-speed coupling are almost 16 times greater than the rated torque, Fig. 3b, which is very dangerous for its fatigue durability. Thus, control of transient torsional vibrations occurring in this drive system during start-ups is very required.

4.1.1 Control based on the brakes

Numerical optimization with respect to the objective function (8) yields a simple hold-and-release strategy: the damping should be set to the maximum value at the beginning of the run-up process and then decreased to the minimum at the time instant which depends on the assumed value of the weighting coefficient α in (8) that balances the objective functions F_1 and F_2 and limits the amount of the dissipated energy for safety reasons.

In the same way as in Fig. 3b, in Fig. 4 there are depicted plots of the analogous time histories of the elastic torques transmitted by the mentioned shafts and excited due to run-up of the controlled system. From these plots it follows that the corresponding extreme values of the elastic torques have been minimised more than 5 times in comparison with these in Fig. 3b. Here, these reduced amplitudes do not exceed dangerously the transmitted nominal torsional moment, where the exact reduction ratio depends on the technological bounds imposed on the control damping coefficient $c_j\,k(t)$ in (5). Nevertheless, it should be remarked that the computed optimum control

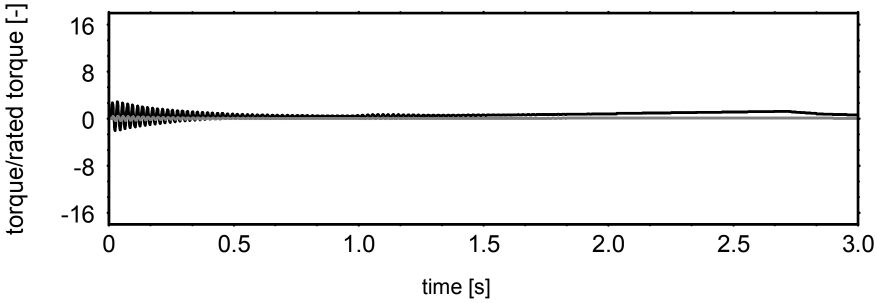

Figure 4: Time histories of the elastic torques for the brake-controlled system.

is a simple hold-and-release strategy and it results in some breaking of the rigid body motion of the system, which leads to a slight run-up retardation in time.

4.1.2 Control based on the clutches

In the considered here case of the asynchronous motor, the external excitation frequency is constant and equals 50 Hz, [5]. Therefore, only the passive control mode defined by equation (13) has been considered. The constant value of the optimum damping coefficient vector $\mathbf{k_0}$ has been chosen basing on the system frequency response function which is depicted in Fig. 5.

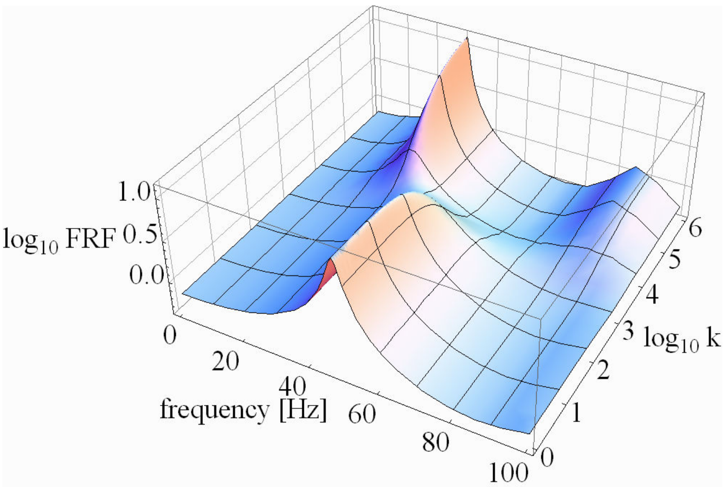

Figure 5: Frequency response function of the drive system.

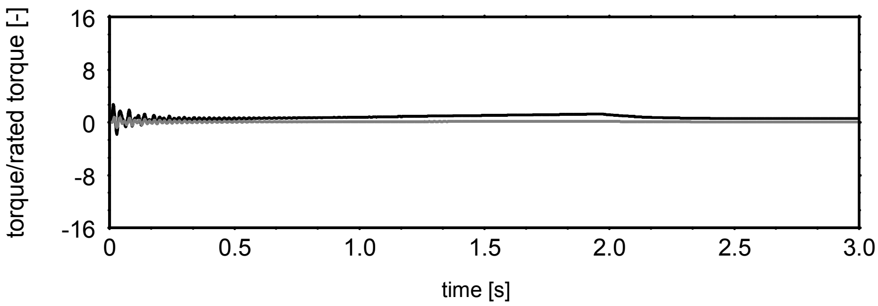

Figure 6: Time histories of the elastic torques for the clutch-controlled system.

In Fig. 6 there are presented analogous plots of time histories of the elastic torques transmitted by the above mentioned shafts, i.e. in the vicinity of the low speed coupling (black line) and in the vicinity of the high speed coupling (grey line), and excited due to run-up of the system controlled by the clutches with a constant viscosity coefficients. From these plots it follows that the corresponding extreme values of the elastic torques have been also minimised more than 5 times in comparison with these in Fig. 3b and they are similar to these in Fig. 4.

4.2 Run-up by means of the synchronous motor

The electromagnetic torque generated by the synchronous motor has been also *a priori* assumed according to the respective relations contained in [5]. Since, as it follows e.g. from [1,5], during drive train run-ups by means of the synchronous motor, the forcing electro-magnetic torques can be characterized by various mutual proportions of the fluctuating component amplitudes to the current instant values of the average torque component. Thus, in this paper two realistic and extreme cases have been investigated, called respectively "large-" and "small dynamic component" of the electromagnetic torque.

4.2.1 Run-up due to the "large dynamic component"

The time history plots of the driving and retarding torques during run-up due to the "large dynamic component" are illustrated in Fig. 7a by the black and grey lines, respectively. In this case, the amplitude envelope of the driving torque fluctuating component is assumed the same as the average torque characteristic, Fig. 7a. The corresponding passive system transient dynamic response caused by this start-up is presented in Fig. 7b in the form of time histories of elastic torques transmitted by the shafts in the vicinity of the low-speed coupling (black line) and the high-speed coupling (grey line). From these plots it follows that the transient component of the synchronous motor torque in the form of sinusoid of decreasing frequency from the double network frequency 100 Hz till zero and maximal amplitude ca. 2 times greater than the rated torque value, Fig. 7a, induces very severe resonance with the first system eigenvibration mode of frequency 47.4 Hz. The maximum amplitudes of the torque in the most heavily affected shaft close to the low-speed coupling are almost 10 times greater than the rated torque, Fig. 7b, which is very dangerous for material fatigue durability. Thus, control of transient torsional vibrations occurring in this drive system during start-ups is also very required.

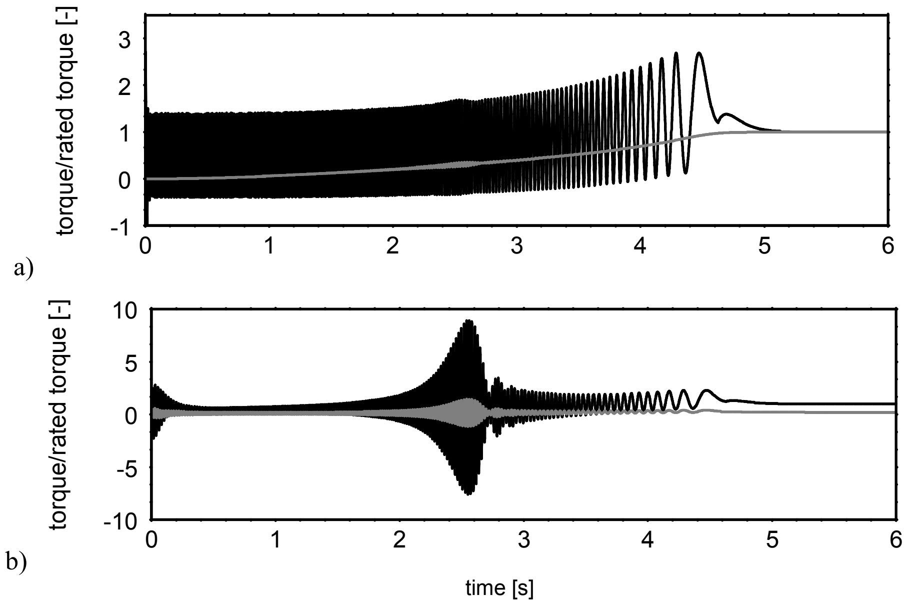

Figure 7: Time histories of the driving and retarding torques (a) and elastic torques (b), in the case of "large dynamic component".

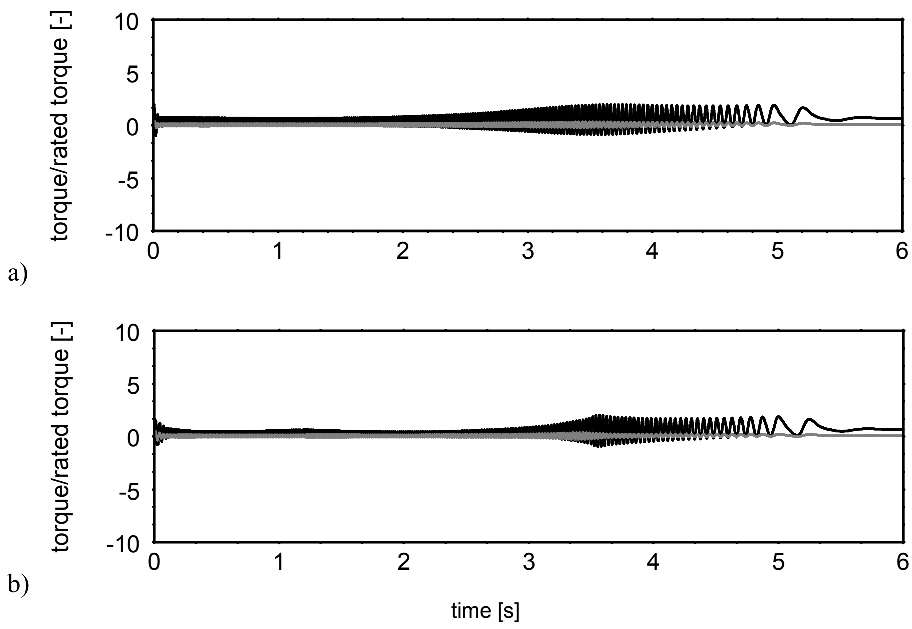

Figure 8: Time histories of the elastic torques for the passive (a) and active (b) control modes.

It has been temporarily assumed that all three inertial rings are controlled collectively:

$$k_1(t) = k_2(t) = k_3(t) = k(t). \tag{14}$$

Both passive and active control modes, defined respectively by equations (11) and (13), are based on the frequency response function, which has been demonstrated in Fig. 5. It has been assumed that the damping coefficient $k(t)$ is technologically constrained to the interval $[10^{2.5}, 10^{4.5}]$, which includes minimum of the FRF at the resonant frequency of 47.4 Hz. The numerical value of the control retardation constant d in equation (12) has been assumed equal to 25.

In the same way as in Fig. 7b, in Fig. 8 there are depicted plots of the time histories of the elastic torques transmitted by the mentioned shafts and excited due to run-up of the controlled system, using both considered control modes. From these plots it follows that the corresponding extreme values of the elastic torques have been minimised more than 5 times in comparison with these in Fig. 7b. Here, these reduced amplitudes do not exceed dangerously the transmitted nominal torsional moment. Note that both active and passive control modes result in a similar peak value of the transmitted elastic torque. However, the active mode also suppresses slightly the mean square value of the transmitted torque, which can be important for a minimization of the general fatigue effort of the shaft. It is interesting, as well, that assumption (14) of the collective control is inessential, i.e. further

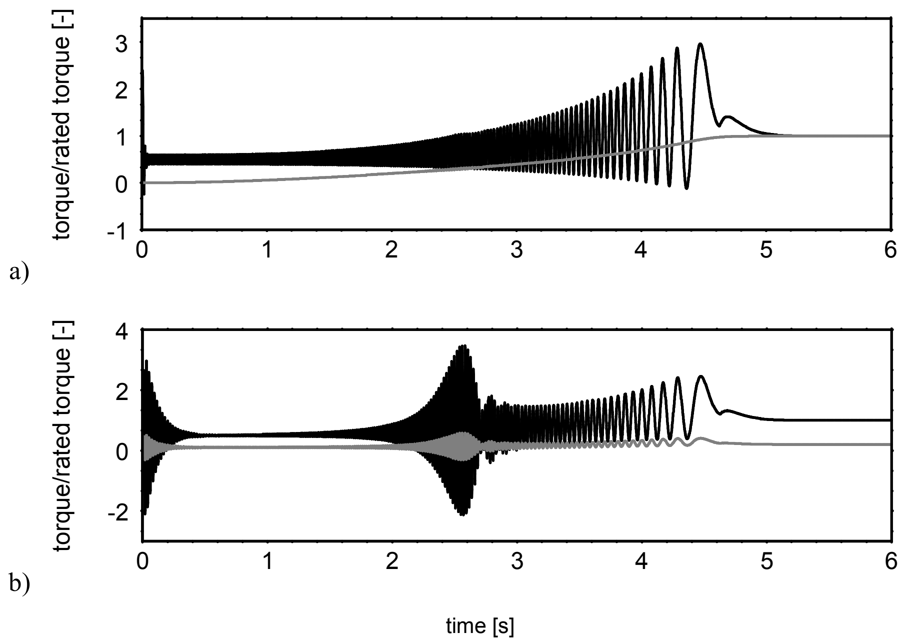

Figure 9: Time histories of the driving and retarding torques (a) and elastic torques (b), in the case of "small dynamic component".

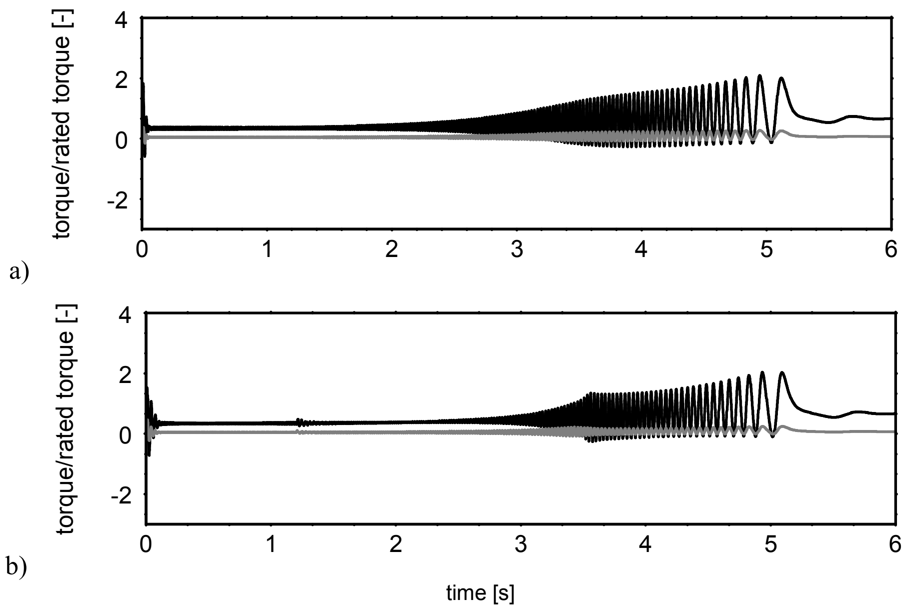

Figure 10: Time histories of the elastic torques for the passive (a) and active (b) control modes.

numerical tests have shown that the separate control does not improve considerably the results shown in Fig. 8.

4.2.2 Run-up due to the "small dynamic component"

The assumed time history plots of the driving and retarding torques during the run-up due to the "small dynamic component" are illustrated in Fig. 9a by the black and grey lines, respectively. In this case, the amplitude envelope of the of the driving torque fluctuating component increases fluently from the 0.1 at the beginning of the start-up to the maximum value of the average torque characteristics after ca. 4.5 s, as shown in Fig. 9a. The corresponding passive system transient dynamic responses in the form of elastic torques transmitted by the abovementioned two most heavily loaded shafts, i.e. in the vicinity of the low speed coupling (black line) and in the vicinity of the high speed coupling (grey line), are presented in Fig. 9b. Here, the maximum amplitudes are considerably lower than in the previously considered case of the "large dynamic component", Fig. 7b, Nevertheless, they are still over 3 times greater than the rated torque.

Both active and passive modes of the control, defined respectively by equations (11) and (13), have been tested. The corresponding results are demonstrated in Fig. 10a and 10b in the same way as above, i.e. in Fig. 9b. The extreme values of the elastic torques have been minimized approximately 2 times, i.e. below the level of the quasi-static elastic torques transmitted by these shafts for the low frequency excitation that occurs at the end of the run-up process.

7 CONCLUSIONS

In the paper control of transient torsional vibrations of the drive system driven by the asynchronous and synchronous motors has been performed by means of actuators with the magneto-rheological fluid (MRF). Depending on the driving torques generated by these electric motors and dynamic behaviours of the drive system during run-ups, two types of actuators are proposed: in the form of control brakes and control clutches. Moreover, for each type of the actuator proper control strategies have been developed. In the case of the asynchronous motor and the drive system with control brakes the computed optimum control is a simple hold-and-release strategy. However, in the case of the synchronous motor equipped with control clutches the active and passive control strategies are proposed. Both are similarly effective in terms of the maximum transmitted torque, but the passive control is easier and cheaper in application. As it follows from the numerical examples, in all considered cases optimum control by means of the applied actuators with the MRF can effectively reduce the transient torsional vibrations 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. Nevertheless, it should be remarked that all control strategies proposed in this paper result in some braking of the rigid body motion of the drive system, which leads to a slight run-up retardation in time.

In the next step of research in this field, by means of the actuators with the magneto-rheological fluid, a semi-active strategy, based on the closed-loop control for drive systems in steady-state operating conditions under randomly excited vibrations, is going to be developed.

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