Applied and Computational Mechanics 7 (2013) 223–234

Determination of the transient vibrations of a rigid rotor attenuated by a semiactive magnetorheological damping device by means of computational modelling

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Received 10 June 2013; received in revised form 23 December 2013

Abstract

Unbalance is the principal source of increase of time varying forces transmitted between the rotor and its stationary part. Their magnitudes can be considerably reduced if the rotor is flexibly suspended and if the damping devices are added to the support elements. Their damping effect must be high for low rotor velocities and small for velocities approximately higher than the critical one to minimize the transmitted forces and the vibrations amplitude. This implies to achieve maximum efficiency of the damping elements, their damping effect has to be adaptable to the current operating conditions. Such technological solution is offered by application of a squeeze film magnetorheological damper. Its hybrid variant consisting of two damping units (one controllable) in a serial arrangement is investigated in this paper. The damping takes place in two concentric lubricating films formed by normal and magnetorheological oils. The damper is equipped with an electric coil generating magnetic flux passing through the layer of the magnetorheological fluid. As resistance against its flow depends on magnetic induction, changing magnitude of the applied current enables to control the damping force. In the computational model, the rotor is considered to be absolutely rigid, unbalanced and the damping elements are represented by force couplings. The goal of the analysis is to study influence of the investigated magnetorheological damper on behaviour of a rigid rotor during different transient regimes. A special attention is focused on passing the rotor through the critical speed and on planning the dependence of the applied current on speed of the rotor rotation to achieve the optimum compromise between minimizing the transmitted forces and maximum attenuation of the rotor vibrations.

Keywords: rigid rotors, controllable damping, hybrid magnetorheological dampers, transient response

1. Introduction

Unbalance is the main source of time varying forces transmitted between the rotor and its frame. They cause fatigue loading of mechanical parts, which can arrive at occurrence of cracks and consequently at fractures. A frequently used technological solution for reducing these undesirable effects consists in application of a rotor flexible suspension.

Basic information on the influence of damping devices added to the constraint elements on rotor vibrations is provided by a simplified dynamical analysis assuming absolutely rigid rotor and linear properties of its flexible suspension (Fig. 1).

The dependences of amplitude of the force transmitted between the rotor and its casing on angular speed of the rotor rotation and on the coefficient of linear damping b_D of its suspension are drawn in Fig. 2. Fig. 3 shows the corresponding frequency responses.

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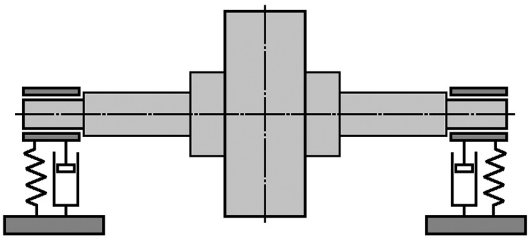

Fig. 1. Scheme of the simplified rotor system

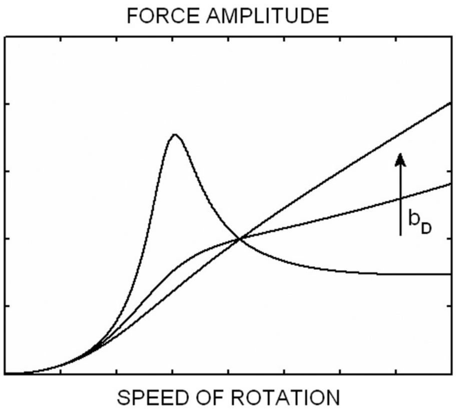

Fig. 2. Transmitted force – speed of rotation relationship

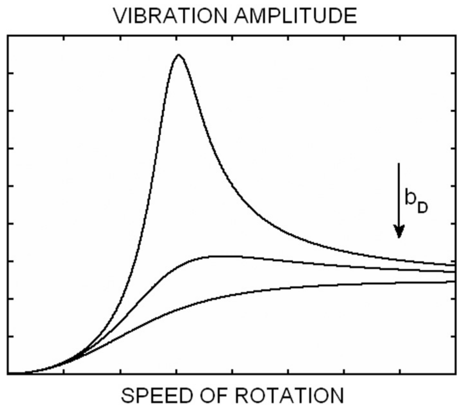

Fig. 3. Rotor frequency responses

It is evident from the analysis of Figs. 2 and 3 that the increasing damping reduces both the transmitted force and the vibration amplitude in the case of lower rotor angular velocities (approximately lower than the critical speed). On the contrary, the rising damping leads to significant increase of the transmitted force but has only small influence (or even negligible) on the attenuation of the rotor vibrations for higher speeds (approximately higher than the critical velocity). This implies to achieve efficient performance of the damping devices in the rotor supports in a wide range of running speeds the damping effect of which must be adaptable to the current operating conditions.

The squeeze film dampers lubricated by Newtonian oils are frequently used in practical rotor dynamic applications. One of the first reported works dealing with their control was published by Burrows et al. [1]. The authors examined the effect of the controlled oil-supply pressure on the change of the system damping coefficients and showed that the proposed approach reduced both the rotor vibrations and the forces transmitted to the machine frame. A different design of a controllable squeeze film damper was developed by Mu et al. [6]. Between the inner and outer rings of the damper there is a gap of a conical form filled with normal oil. The position of the outer ring in the axial direction is adjustable, which enables to change both the radial clearance and the land length of the damper and thus to control the damping force by the ring shifting.

A new concept of controlling the damping effects is represented by magnetorheological devices. As resistance against the flow of magnetorheological liquids depends on magnetic induction, the damping force can be controlled by changing the magnitude of electric current generating magnetic flux passing through the lubricating film. Wang et al. [7] studied the vibration characteristics and the control method of a flexible rotor equipped with a magnetorheological squeeze film damper by means of experiments. Forte et al. [4] presented results of the theoretical and experimental investigations of a long magnetorheological damping device. Wang et al. [8] developed a mathematical model of a long squeeze film magnetorheological damper based on the modified Reynolds equation. The results of experiments performed with a squeeze film magnetorheological damping element on a small test rotor rig were reported by Carmignani et al. [2, 3]. Zapoměl et al. [10] worked out a mathematical model of a short squeeze film magnetorheological damper, which can be applied for analysis of both the steady state and transient rotor vibrations.

A new hybrid variant of a magnetorheological damper intended for minimizing the force transmission between the rotor and its casing during unsteady operating regimes is investigated in this article. A squirrel spring of the damper supporting the rotor forms a flexible suspension. The rotor is considered as absolutely rigid. The damper is of a squeeze film kind and the dissipation of mechanical energy takes place in two concentric fluid layers of normal and magnetorheological oils arranged in a serial way. The damping effect produced by squeezing the layer of the magnetorheological oil is controlled by the change of magnetic flux generated in electric coils. The computational simulations showed that a suitable current control in dependence on speed of the rotor rotation enabled to achieve the optimum compromise between the reduction of the force transmitted through the constraint elements to the stationary part and the attenuation of the rotor vibrations during its acceleration and passing the critical velocity.

Proposal of a new design solution of the damping element, development of its mathematical model together with learning more on its effect on a rigid rotor behaviour represent the principal contributions of this article.

2. The hybrid squeeze film magnetorheological damping element with serial arrangement of the oil layers

The carried out simplified analysis shows that to achieve maximum efficiency of the damping elements placed between the rotor and its casing their damping effect must be as large as possible for lower velocities and minimum for angular speeds approximately higher than the critical one. This has arrived at the idea of investigating a concept of a hybrid damping element consisting of two independent damping units, from which one is controllable, in a serial arrangement. In this case, the resulting damping coefficient is always lower than the lower one from both individual damping units. This makes it possible to reach lower damping in the range of high rotor velocities than those produced by a compact device containing only one lubricating layer.

The principal parts of the proposed damping element (Fig. 4) are three rings, from which two are moveable. The clearances between the rings are filled with lubricating oils. The inner ring is coupled with the rotor journal by a rolling element bearing and with the damper's body by a squirrel spring. The outer ring is stationary and fixed to the damper housing. The lubricating films are concentric formed by normal (inner) and magnetorheological (outer) oils and are mutually separated by a thin ring flexibly coupled with the damper's housing. From the physical point of view, the lubricating films are arranged in a serial way. Because of the lateral vibrations of the rotor, the oil films between the rings are being squeezed, which produces the damping effect. A substantial part of the studied damper is an electric coil generating magnetic flux passing through the layer of the magnetorheological liquid. As resistance against its flow depends on magnetic induction, the change of the applied electric current can be used to control the damping force.

The developed mathematical model of the studied damping element is based on utilization of the classical theory of lubrication with some modifications. The magnetorheological oil is

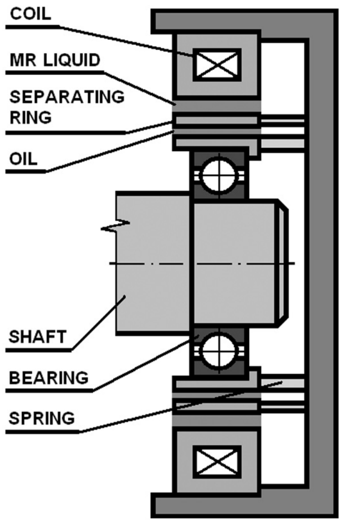

Fig. 4. Scheme of the damping device

represented by Bingham material with the yielding shear stress depending on magnetic induction. If no magnetic field is applied, it behaves as Newtonian liquid, as well as the normal oil. Further, it is assumed that the geometric and design parameters enable to consider the damping device as short (the length to diameter ratio of the rings is small, no or soft sealings are applied at the damper's ends).

Based on a simple geometric analysis, the relations for thicknesses of the thin films of normal and magnetorheological oils read (Krämer [5])

$$h_{NO} = c_{NO} - e_{NO}\cos(\varphi - \gamma_{NO}), \tag{1}$$

$$h_{MR} = c_{MR} - e_{MR}\cos(\varphi - \gamma_{MR}). \tag{2}$$

h_{NO}, h_{MR} are the thicknesses of the classical and magnetorheological oil films, c_{NO}, c_{MR} are the widths of the gaps between the rings filled with normal and magnetorheological oils, e_{NO}, e_{MR} denote the rotor journal and the separating ring eccentricities, φ is the circumferential coordinate and γ_{NO}, γ_{MR} denote the position angles of the lines of centres of the rotor journal and the separating ring respectively (Fig. 5).

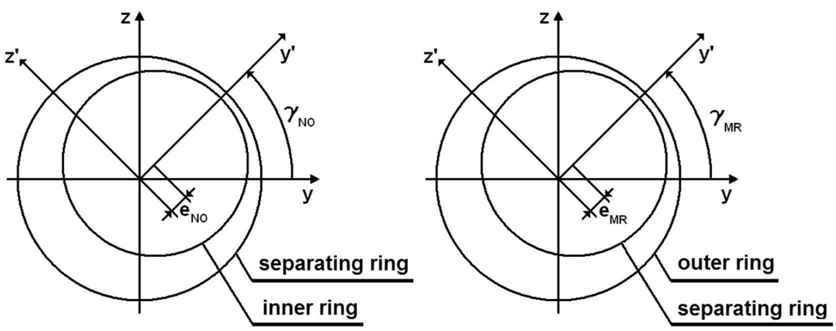

Fig. 5. The damper's coordinate systems

The Reynolds equations governing the pressure distribution in the layers of normal and magnetorheological oils adapted for short squeeze film dampers take the form (Krämer [5], Zapoměl et al. [10]),

$$\frac{\partial^2 p_{NO}}{\partial Z^2} = \frac{12\eta}{h_{NO}^3}\dot{h}_{NO}, \tag{3}$$

$$h_{MR}^3 p'^3_{MR} + 3\left(h_{MR}^2\tau_y - 4\eta_B\dot{h}_{MR}Z\right)p'^2_{MR} - 4\tau_y^3 = 0 \quad \text{for } p'_{MR} < 0, \tag{4}$$

$$h_{MR}^3 p'^3_{MR} - 3\left(h_{MR}^2\tau_y + 4\eta_B\dot{h}_{MR}Z\right)p'^2_{MR} + 4\tau_y^3 = 0 \quad \text{for } p'_{MR} > 0. \tag{5}$$

p_{NO}, p_{MR} denote the pressures in the layers of normal and magnetorheological oils respectively, η, η_B are the dynamical and Bingham viscosities of the normal and magnetorheological oils, τ_y represents the yield shear stress, Z is the axial coordinate and $(\dot{\ })$, $(')$ denote the first derivative with respect to time and coordinate Z. The Reynolds equations (4) and (5) are valid for $Z > 0$.

The governing equations (3)–(5) are solved for the boundary conditions expressing that the pressure at the damper's ends is equal to the pressure in the ambient space. Relationships (4) and (5) represent the polynomial algebraic equations of the third order for the pressure gradient. The sought root must fulfil the conditions: p'_{MR} is real (not complex), is negative for equation (4) or positive for equation (5) and it satisfies the relation

$$|p'_{MR}| > \frac{2\tau_y}{h_{MR}}. \tag{6}$$

After determining the pressure gradient from equations (4) or (5), the pressure distribution in the axial direction is calculated by the integration

$$p_{MR} = \int p'_{MR}\,\mathrm{d}Z. \tag{7}$$

To solve equation (3) and integral (7), the constants of integration are determined for the boundary conditions which express the pressure at the damper's ends is equal to the pressure in the ambient space.

For the damper's simplest design arrangement, the outer and separating rings can be considered as a divided core of an electromagnet having the gap filled with the magnetorheological oil. Then dependence of yielding shear stress on magnetic induction can be approximately expressed

$$\tau_y = k_d\left(\frac{I}{h_{MR}}\right)^{n_y}, \tag{8}$$

where n_y is the magnetorheological liquid material constant, k_d is the design parameter depending on the number of the coil turns and material properties of the magnetorheological liquid and I is the applied electric current. More detailed information on determination of the yielding shear stress in the layer of the magnetorheological film can be found in [9, 10].

In the areas where the thickness of the lubricating films increases with time ($\dot{h}_{NO} > 0$, $\dot{h}_{MR} > 0$), a cavitation is assumed. In these regions, the pressure of the medium remains constant and equal to the pressure in the ambient space. In noncavitated areas, the pressure is governed by solutions of the Reynolds equation (3) and integral (7).

Consequently, components of the damping forces are calculated by integrating the pressure distributions around the circumference and along the length of the damping element taking into account the cavitation in the oil films. Then it holds

$$F_{MRy} = -2R_{MR} \int_0^{2\pi} \int_0^{\frac{L}{2}} p_{DMR} \cos\varphi \, \mathrm{d}Z \, \mathrm{d}\varphi, \tag{9}$$

$$F_{MRz} = -2R_{MR} \int_0^{2\pi} \int_0^{\frac{L}{2}} p_{DMR} \sin\varphi \, \mathrm{d}Z \, \mathrm{d}\varphi, \tag{10}$$

$$F_{NOy} = -2R_{NO} \int_0^{2\pi} \int_0^{\frac{L}{2}} p_{DNO} \cos\varphi \, \mathrm{d}Z \, \mathrm{d}\varphi, \tag{11}$$

$$F_{NOz} = -2R_{NO} \int_0^{2\pi} \int_0^{\frac{L}{2}} p_{DNO} \sin\varphi \, \mathrm{d}Z \, \mathrm{d}\varphi. \tag{12}$$

F_{NOy}, F_{NOz}, F_{MRy}, F_{MRz} are the y and z components of the hydraulic forces produced by the pressure in the films of normal and magnetorheological oils respectively, R_{NO}, R_{MR} are the mean radii of the layers of normal and magnetorheological oils, L is the axial length of the damping element and p_{DNO}, p_{DMR} denote the pressure distributions in the layers of normal and magnetorheological oils (taking into account different pressures in cavitated and noncavitated regions).

3. The investigated rotor system

The investigated rotor consists of a shaft and of one disc and it is coupled with the frame by the studied constraint elements at both its ends (Fig. 6). The rotor operates at variable angular speed and it is loaded by its weight and excited by the disc unbalance. The squirrel springs of the damping elements are prestressed to eliminate their deflection caused by the rotor weight. The whole system is symmetric relative to the disc middle plane.

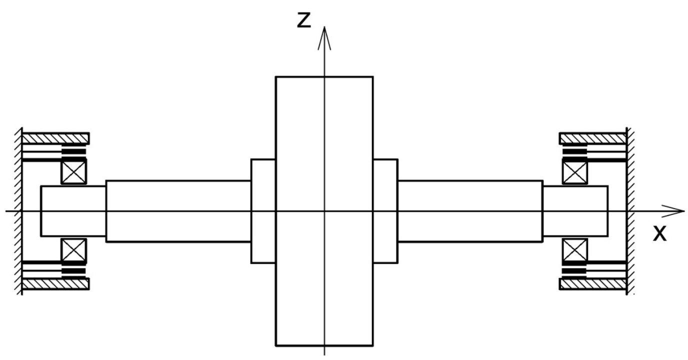

Fig. 6. Scheme of the investigated system

The technological parameters of the rotor system are: mass of the rotor 450 kg, stiffness of one squirrel spring 5 MN/m, the length of each constraint element 60 mm, widths of the clearances filled by the normal and magnetorheological oils 0.2 mm, 1.0 mm, middle radii of the layers of the normal and magnetorheological oils 55 mm, 75 mm, dynamical and Bingham viscosities of the normal and magnetorheological oils 0.004 $\mathrm{Pa \cdot s}$, 0.3 $\mathrm{Pa \cdot s}$, eccentricity of the rotor centre of gravity 0.1 mm, exponential material constant of the magnetorheological oil 2 and the value of the design parameter 0.001 N/A^2.

In the computational model, the rotor is considered as absolutely rigid and the constraint devices are represented by springs and force couplings. The task was to find an effective rule for controlling the damping force to achieve the optimum compromise between the reduction of the force transmitted to the rotor frame and minimization of the rotor vibration during its acceleration from angular velocity of 150 rad/s to 350 rad/s during the time period of 2 s (Fig. 7).

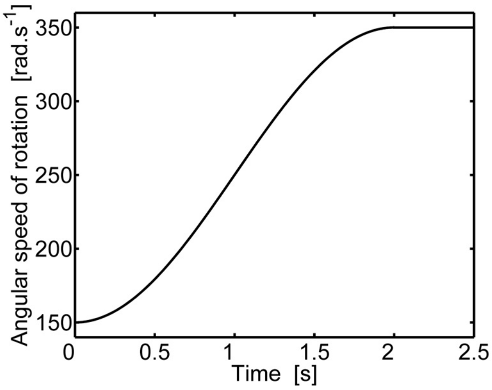

Fig. 7. Time history of required angular speed of the rotor rotation

The lateral vibration of the rotor system is governed by a set of four nonlinear differential equations

$$m_R \ddot{y}_R + b_P \dot{y}_R + 2k_R y_R = m_R e_T \left(\dot{\vartheta}^2 \cos\vartheta + \ddot{\vartheta} \sin\vartheta \right) + 2F_{NOy}, \tag{13}$$

$$m_R \ddot{z}_R + b_P \dot{z}_R + 2k_R z_R = m_R e_T \left(\dot{\vartheta}^2 \sin\vartheta - \ddot{\vartheta} \cos\vartheta \right) - m_R g + 2F_{PSR} + 2F_{NOz}, \tag{14}$$

$$m_{SR} \ddot{y}_{SR} + k_{SR} y_{SR} = -F_{NOy} + F_{MRy}, \tag{15}$$

$$m_{SR} \ddot{z}_{SR} + k_{SR} z_{SR} = -F_{NOz} + F_{MRz} - m_{SR} g + F_{PSSR}. \tag{16}$$

m_R is the rotor mass, b_P is the coefficient of the rotor external damping, k_R is the stiffness of the squirrel spring supporting the rotor, m_{SR}, k_{SR} are the mass of the ring separating the lubricating layers and the stiffness of its support respectively, e_T is the eccentricity of the rotor unbalance, θ is the angle of the rotor rotation about its axis, t is the time, y_R, z_R, y_{SR}, z_{SR} are the displacements of the rotor centre (centre of the rotor journal) and of the centre of the ring separating the lubricating layers in the horizontal and vertical directions, g is the gravity acceleration, F_{PSR}, F_{PSSR} are the forces prestressing the dampers' springs and $(\ddot{\ })$ denotes the second derivative with respect to time.

4. Results of the computational simulations

The time histories of the force transmitted between the rotor and its casing for two constant magnitudes of the applied current (0 A, 1.27 A) are depicted in Figs. 8 and 9. The corresponding time courses of displacements of the rotor and of the separating ring centres in the horizontal direction are drawn in Figs. 10–13. The results show that the applied current reduces the transmitted force and amplitude of the rotor vibration for lower velocities (approximately lower than the critical speed) but for higher angular speeds it leads to their increase. The vibration amplitude of the separating ring is attenuated in the whole extent of the rotor operating velocities (Fig. 13).

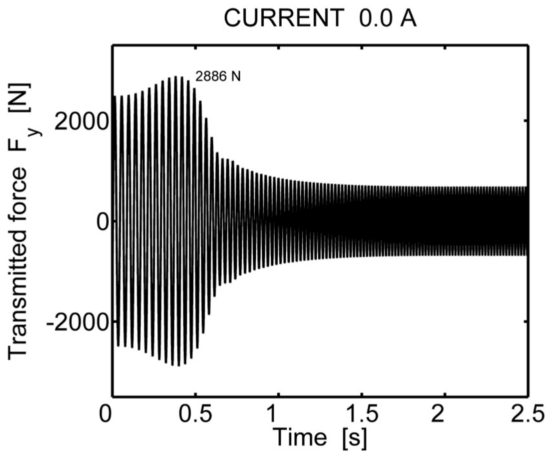

Fig. 8. Time history of the transmitted force

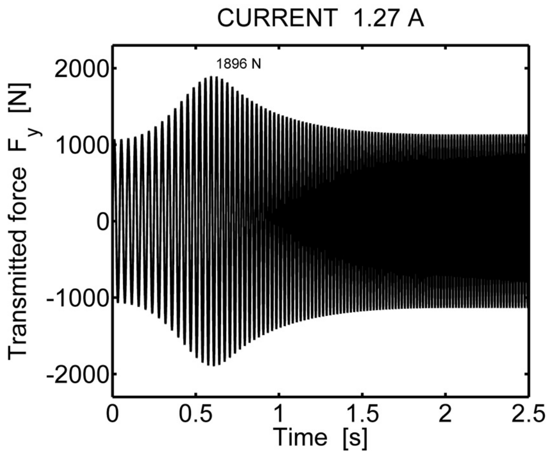

Fig. 9. Time history of the transmitted force

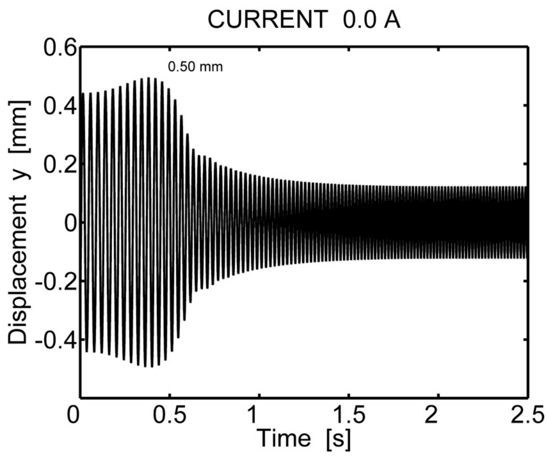

Fig. 10. Time history of the rotor displacement

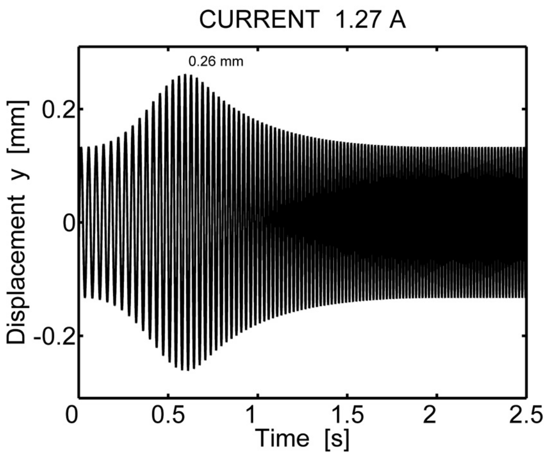

Fig. 11. Time history of the rotor displacement

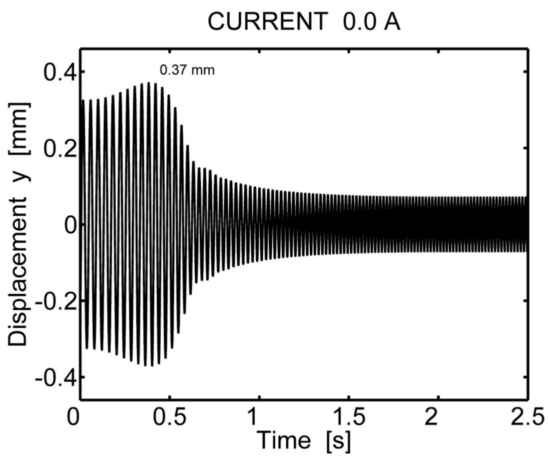

Fig. 12. Separating ring displacement

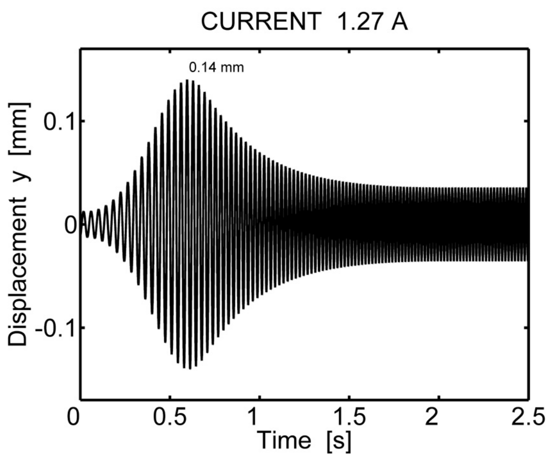

Fig. 13. Separating ring displacement

The rotor mass and the supporting squirrel springs stiffness determine the system undamped resonance frequency to 149 rad/s. Comparison of Figs. 8 with 9, 10 with 11 and 12 with 13 gives the evidence that increase of the hydraulic (damping) force shifts the system resonance frequency to a higher value, by employing diagram in Fig. 7, approximately to 169 rad/s if no current is applied and to 193 rad/s if the current is switched on.

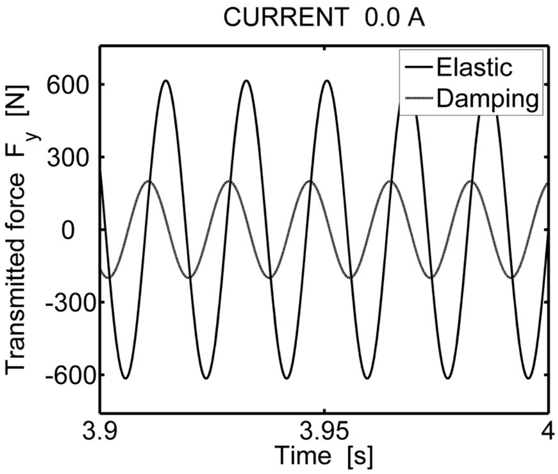

Fig. 14. Transmitted force components

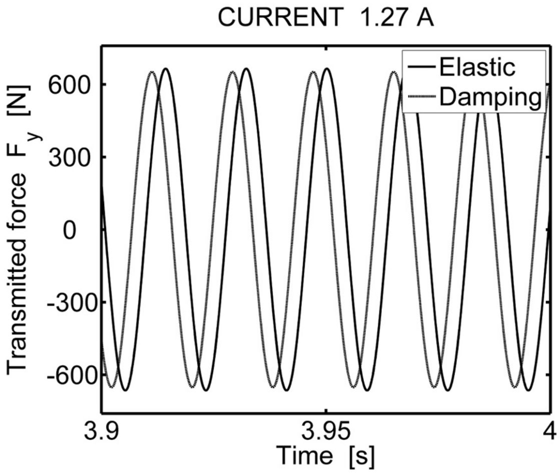

Fig. 15. Transmitted force components

The force between the rotor and its casing is transmitted through the squirrel spring (the elastic force) and the lubricating layers (the damping force) of the damping element. The time histories of components of both these forces referred to the rotor steady state vibration after finishing its acceleration are drawn for two magnitudes of the applied current in Fig. 14 and 15. It is evident that the applied current influences their magnitudes but has almost no effect on their mutual phase shift which is caused by their different physical substance (elastic and damping forces).

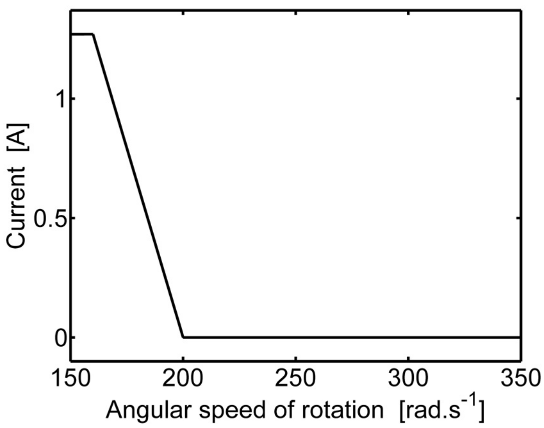

Fig. 16. Current angular speed of rotation history – variant 1

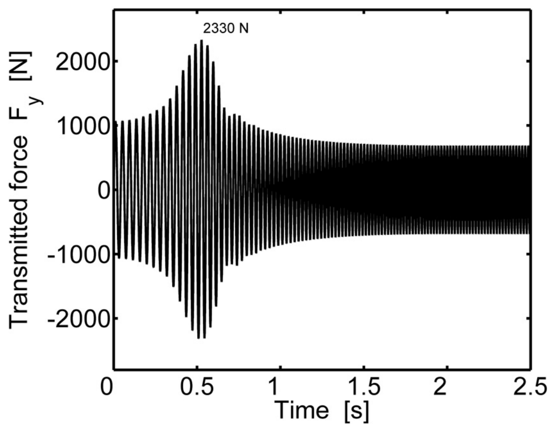

Fig. 17. Force time history – variant 1

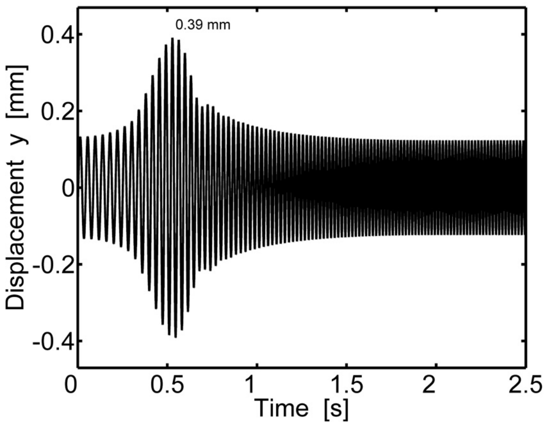

Fig. 18. Rotor displacement time history – variant 1

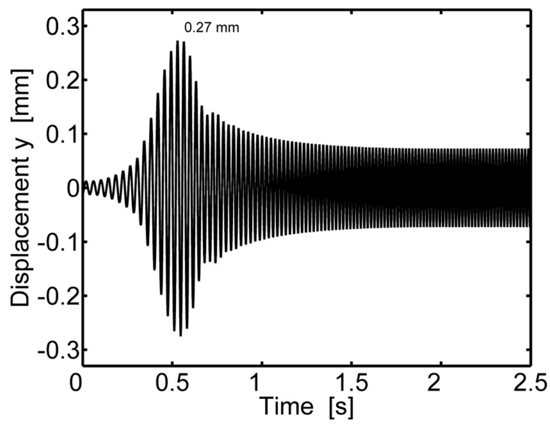

Fig. 19. Separating ring displacement – variant 1

Analysis of Figs. 8–11 shows that the maximum suppression of the transmitted force and of the rotor vibration requires to apply high current in the area of the rotor low velocities but to switch it off after crossing the critical speed.

Therefore, to achieve the optimum system response, several strategies differing in the moment of beginning, course and duration of switching off the current have been studied. The corresponding time histories of the transmitted force and the rotor oscillations, both in the horizontal direction, are drawn for three variants of dependence of the current on the rotor rotational speed in Figs. 16–27. It is evident that variant 3 gives the best results as it enables maximum reduction of the transmitted force and minimum elevation of the rotor vibration amplitude. Early starting this manipulation (variant 1) arrives at large increase of the transmitted force accompanied by large increase of vibration of the rotor and of the separating ring. Similarly, shifting the beginning of this manipulation to higher angular velocities and extending time of its duration (variant 2) does not lead to the rotor satisfactory behaviour.

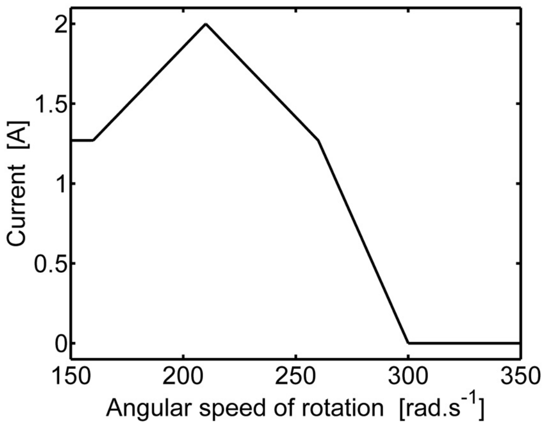

Fig. 20. Current angular speed of rotation history – variant 2

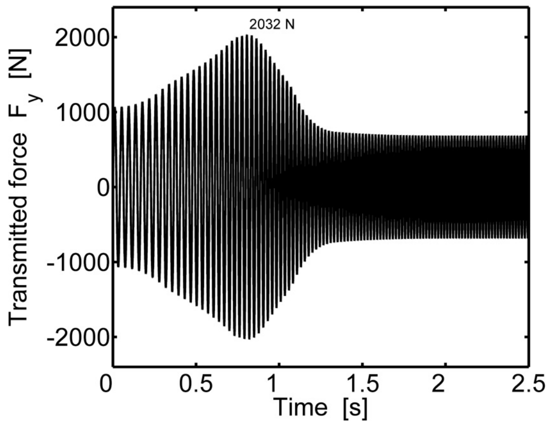

Fig. 21. Force time history – variant 2

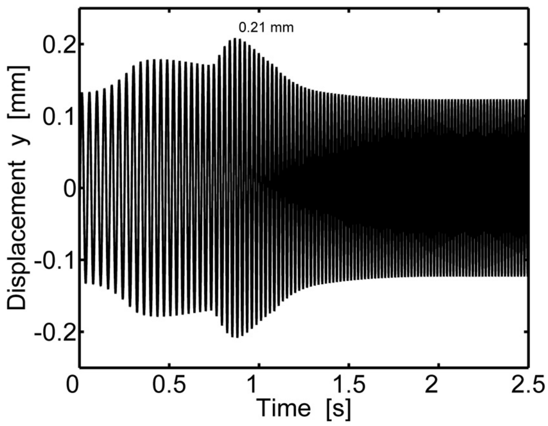

Fig. 22. Rotor displacement time history – variant 2

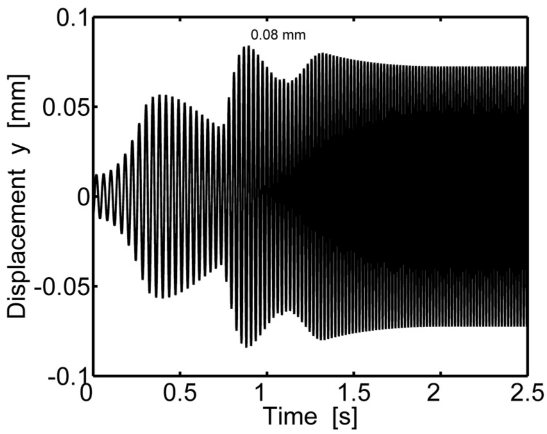

Fig. 23. Separating ring displacement – variant 2

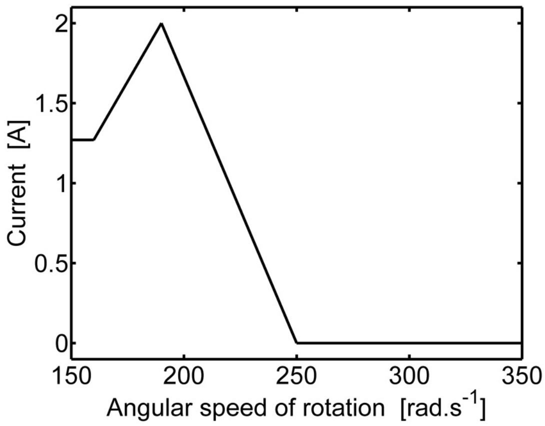

Fig. 24. Current angular speed of rotation history – variant 3

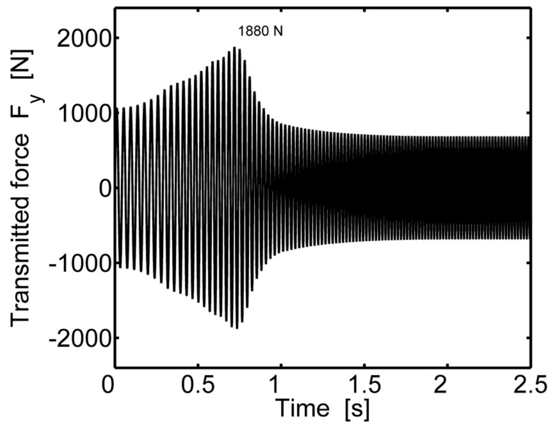

Fig. 25. Force time history – variant 3

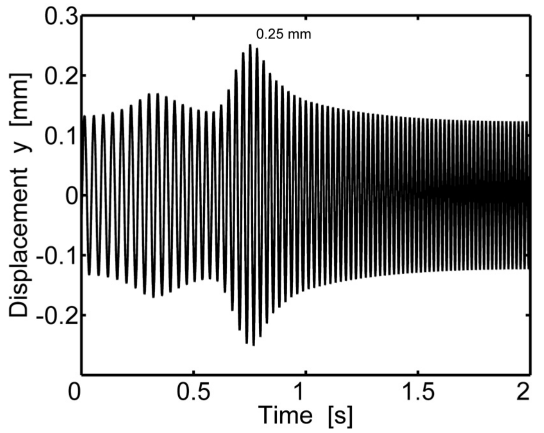

Fig. 26. Rotor displacement time history – variant 3

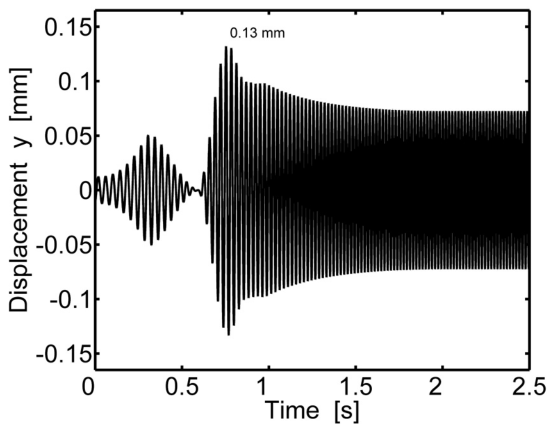

Fig. 27. Separating ring displacement – variant 3

5. Conclusions

A flexible suspension of rigid rotors is an often used technological solution that enables reduction of the forces transmitted between the rotor and its casing. The carried out computational simulations confirmed that the dampers added to the rotor supports had significant influence on both the magnitude of the transmitted force and the amplitude of the system vibrations.

The advantage of the studied damping device is that it enables to change the damping effect by controlling electric current and thus to adapt its performance to the current running conditions. The damper works on a semiactive principle and thus it does not require a complicated and expensive control system for its operation.

The theory confirmed by results of computational simulations shows that the damping effect of the damping devices added to the rotor supports should be high for velocities approximately lower than the resonance frequency and minimum for higher angular speeds. The studied design represents a hybrid variant of a magnetorheological damping element consisting of two lubricating layers in a serial arrangement. It enables to achieve larger reduction of the damping force in the range of higher rotor angular velocities than a compact damping device that has only one lubricating film. The results of performed computational simulations give the evidence that the change of the current (the current switching off) must not be sudden but that it has to be distributed over a certain speed interval so that the optimum performance of the damping element could be achieved. The efficiency of this manipulation depends on the moment of its starting and on the time of its duration.

Acknowledgements

The work reported in this article has been done with the institutional support RVO:61388998.

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