



LIMITATION OF DYNAMIC LOADS IN MACHINE DRIVES

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Summary

Couplings incorporating rubber-cord shells, and their viscoelastic characteristics have been studied. The effectiveness of the couplings to limit the dynamic load is shown by an example of aballmill drive. It has been found that the use of the couplings equipped with rubber-cord shells as the flexible elements allows limiting the dynamic shock loads in machinedrives. It has been shown that the elimination of resonance modes and restriction of forced torque fluctuations in the drive can be provided by selecting a limit torque for the coupling actuation, which is determined by an initial air pressure in the rubber-cord shells of the coupling.

Keywords: pneumatic flexible shaft coupling, bellows flexible shaft couplings, rubber-cord shells, ball mill drive, shock loads, forced oscillations.

OGRANICZENIE OBCIĄŻEŃ DYNAMICZNYCH W NAPĘDACH UKŁADÓW MECHANICZNYCH

Streszczenie

Tematem artykułu naukowego są sprzęgła elastyczne z członami pneumatyczno-elastycznymi łączące wały. Zalety oraz efektywność zastosowania wspomnianych sprzęgieł w celu ograniczenia obciążeń dynamicznych przedstawiono na przykładzie napędu młyna bębnowego. Udowodniono, że poprzez zastosowanie sprzęgieł z członami pneumatyczno-elastycznymi możliwe jest ograniczenie obciążeń napędów młynów bębnowych charakteryzujących się przede wszystkim dynamicznymi obciążeniami udarowymi. Równocześnie stwierdzono, że wykluczenie trybów pracy, w których występuje rezonans i ograniczenie w ten sposób drgań wymuszonych, jest możliwe poprzez odpowiedni wybór nadciśnienia medium gazowego w członach pneumatyczno-elastycznych danych sprzęgieł.

Słowa kluczowe: sprzęgło elastyczne, pneumatyczne sprzęgło elastyczne łączące wały, człony pneumatyczno-elastyczne, napęd młyna bębnowego, dynamiczne obciążenia udarowe, drgania wzbudzone.

1. INTRODUCTION

In the start-up period and the steady state operation of drives in various-type production machinery [1], [2], [3], there occur dynamic loads that may several times exceed the nominal load and may decrease the rated life and reliability of mechanical systems [4], [5], [6].

Flexible couplings have long been used to limit the dynamic loads in machine drives [7], [8], [9], [10], [11]. Recently, the couplings that incorporate rubber-cord shells as flexible elements have become more and more widespread [6], [7], [12], [13], [14].

Rubber-cord flexible pneumatic elements offer several advantages as compared to metal elements [15], [16], [17], [18], [19], [20]:

- higher durability;
- the possibility to automatically control the stiffness and dissipative characteristics;

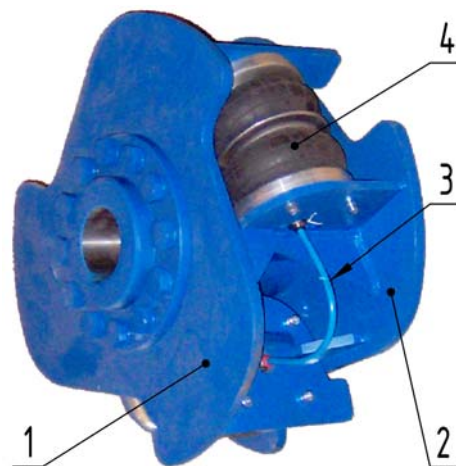


Fig. 1. Bellows flexible shaft coupling:
1 – driving part, 2 – driven part, 3 – conduit,
4 – pneumatic flexible elements (rubber-cord shells)

- increased compensation of the axes misalignment angles

Fig.1 shows the flexible shaft coupling (bellows flexible shaft coupling), developed by the Polish company FENA at the Technical University in Kosice (Slovakia) under the guidance of Prof.

Jaroslav Homisin [13], [14], [21], [22], [23], [24]. It uses rubber-cord shells filled with air as flexible elements.

Bellows flexible shaft couplings can be used in rotary mills, crushers, pumps, fans, compressors, conveyors, mixers and other machines and apparatus [6], [7], [18], [24], [26], [27], [28], [29], [30].

The aim of the study is to prove the effectiveness of flexible shaft couplings by an example of the ball mill drive.

2. A DYNAMIC MODEL OF A SYNCHRONOUS DRIVE OF A TUMBLING BALL MILL

An analysis of dynamic processes in the mill drive is done using the side drive as an example [5], [18], the dynamic diagram being shown in Fig. 2, where I_1 , I_2 are equivalent moments of inertia of the rotor and the drum under load, respectively; c_{12} is the equivalent stiffness of the mechanical transmission; $M_{EM}(t)$, $M_R(\varphi_2, \omega_2)$ are the electromagnetic engine torque and the drum resistance torque caused by the charge material gravity torque Gl/u (G is the charge material gravity torque with respect to the rotational axis, u is gear ratio) and the frictional torque M_{fr} .

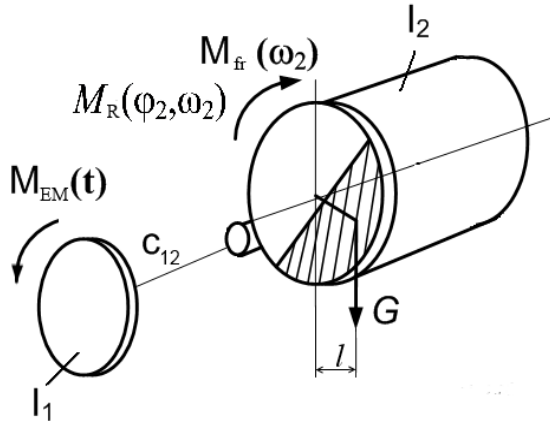


Fig.2. A dynamic model of the mill drive:

I_1 , I_2 are equivalent moments of rotor and drum inertia;
 c_{12} is equivalent stiffness of mechanical transmission;
 $M_{EM}(t)$, $M_R(\varphi_2, \omega_2)$ are electromagnetic engine torque and drum resistance moments

Differential dynamic equations for the ball mill power unit, which in relative terms describe the electromagnetic processes in the synchronous motors by the Park-Gore equations, will have the following form [12]:

$$\left. \begin{aligned} \frac{d\omega_1}{dt} &= \frac{1}{H_1} [M_{EM} - M(\alpha_{12}) - M(\dot{\alpha}_{12})] \\ \frac{d\omega_2}{dt} &= \frac{1}{H_2} [M(\alpha_{12}) + M(\dot{\alpha}_{12}) - M_C(\varphi_2, \omega_2)] \\ \frac{d\varphi_1}{dt} &= \omega_1 \\ \frac{d\varphi_2}{dt} &= \omega_2 \\ \alpha_{12} &= \varphi_1 - \varphi_2 \end{aligned} \right\} (1)$$

$$\left. \begin{aligned} M_{EM} &= \psi_q \left(\frac{\psi_d}{x_d} - \frac{\psi_f}{x_{df}} - \frac{\psi_{ed}}{x_{dee}} \right) - \psi_d \left(\frac{\psi_q}{x_q} - \frac{\psi_{eq}}{x_{qee}} \right) \\ \frac{d\psi_d}{dt} &= U_m \sin \theta - r_a \left(\frac{\psi_d}{x_d} - \frac{\psi_f}{x_{df}} - \frac{\psi_{ed}}{x_{dee}} \right) - \psi_q \omega_1 \\ \frac{d\psi_q}{dt} &= U_m \cos \theta - r_a \left(\frac{\psi_q}{x_q} - \frac{\psi_{eq}}{x_{qee}} \right) + \psi_d \omega_1 \\ \frac{d\psi_f}{dt} &= U_f - r_f \left(\frac{\psi_f}{x_f} - \frac{\psi_d}{x_{df}} - \frac{\psi_{ed}}{x_{fed}} \right) \\ \frac{d\psi_{ed}}{dt} &= -r_{ed} \left(\frac{\psi_{ed}}{x_{ed}} - \frac{\psi_d}{x_{dee}} - \frac{\psi_f}{x_{fee}} \right) \\ \frac{d\psi_{eq}}{dt} &= r_{eq} \left(\frac{\psi_{eq}}{x_{eq}} - \frac{\psi_q}{x_{qee}} \right) \\ \frac{d\theta}{dt} &= 1 - |\omega_1| \end{aligned} \right\} (2)$$

where Ψ_d , Ψ_q are magnetic-flux linkage of the stator winding along a longitudinal and a transverse axis; Ψ_f is field winding flux linkage; Ψ_{ed} , Ψ_{eq} are damper winding flux linkage along a longitudinal and a transverse axis; U_m is the amplitude of the phase voltage; U_f is field winding voltage; ω_1 is the angular speed of the motor; H_1 , H_2 are moments of inertia of the rotor and drum in relative units; M_{EM} is electromagnetic torque of the motor; $M_R(\varphi_2, \omega_2)$ is moment of resistance; θ is peripheral displacement between stator and rotor space vectors; r_a , r_f , r_{ed} , r_{eq} are resistances of the stator phase, field winding, and damper winding along the longitudinal and transverse axes; x_d , x_f , x_{ed} , x_{df} , x_{ded} , x_{fed} , x_q , x_{eq} , x_{qee} are super transition resistances; φ_1 , φ_2 are angles of the engine and drum rotation.

Equivalent to the rotor shaft and presented in relative terms, the moment of resistance M_R takes the following form

$$M_R(\varphi_2, \omega_2) = M_{fr}(\omega_2) + M_G(\varphi_2), \quad (3)$$

where

$$M_{fr}(\omega_2) = \begin{cases} M_{fr.0} + \omega_2 \frac{M_{fr.st} - M_{fr.0}}{\omega_2^*} & \text{if } 0 \leq \omega_2 \leq \omega_2^* \\ M_{fr.st} & \text{if } \omega_2 > \omega_2^* \end{cases} \quad (4)$$

$$M_G(\varphi_2) = \begin{cases} \frac{Gl}{u} \sin\left(\beta_0 + \frac{\varphi^2}{p_e u}\right) & \text{if } 0 \leq \varphi_2 \leq \frac{\pi}{2} u p_e \\ 2M_{Gmax} - M_{Gst} + \varphi_2 \frac{M_{Gst} - M_{Gmax}}{0,5\pi u p_e} & \text{if } \frac{\pi}{2} u p_e \leq \varphi_2 \leq \pi u p_e \\ M_{Gst} & \text{if } \pi u p_e < \varphi_2 \end{cases} \quad (5)$$

where $M_{fr.0}$ is the friction torque at the initial time; $M_{fr.st}$ is steady-state friction torque; ω_2^* is angular rotational speed of the drum, which produces the smallest friction torque; β_0 is the initial angle of the charge material center of gravity in respect to the vertical axis of the drum; M_{Gmax} is maximum moment of charge material resistance; M_{Gst} is moment of resistance of the charge material weight in the steady-state operation; p_e is the number of motor pole pairs.

The start occurs asynchronously at $U_f = 0$, and the field winding is closed on a discharge resistor $r_{f\Sigma}$. When sub-synchronous speed ω' has been reached, the discharge resistor is switched off and the field current is supplied. Thus, the following condition should be met:

$$r_f = r_{f\Sigma}, U_f = 0 \text{ at } 0 \leq \omega_2 \leq \omega';$$

$$r_f = r_{f\Sigma}, U_f \neq 0, \text{ at } \omega_2 \geq \omega'.$$

The adequacy of the proposed mathematical model was verified experimentally using the MMC 90x30 ball mill. The mill drive includes: a synchronous motor SDMZ 2-24-59-80U4 with rated power $P=4,000$ kW and rotational speed $n_1 = 75$ rpm, and a flexible coupling (Fig.3) that uses rubber plates as flexible elements (equivalent coupling

stiffness is $50 \cdot 10^3$ kN·m·rad⁻¹). The torque is transmitted to the drum by an open gear with a gear ratio of $u = 6.76$.

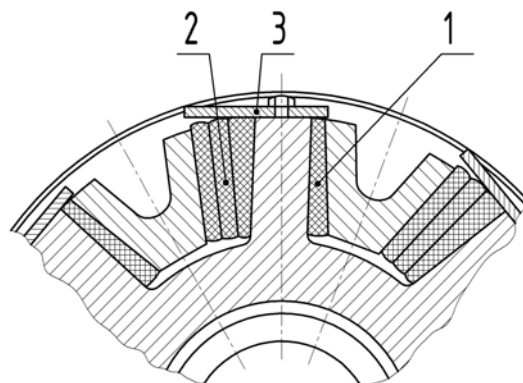


Fig.3. Schematic of a flexible coupling: 1,2 – rubber plates; 3 – mounting bracket

Fig.4a shows an experimental oscillogram of the torque in the mill shafting [5], [12] Fig.4b shows a calculated torque- time dependence.

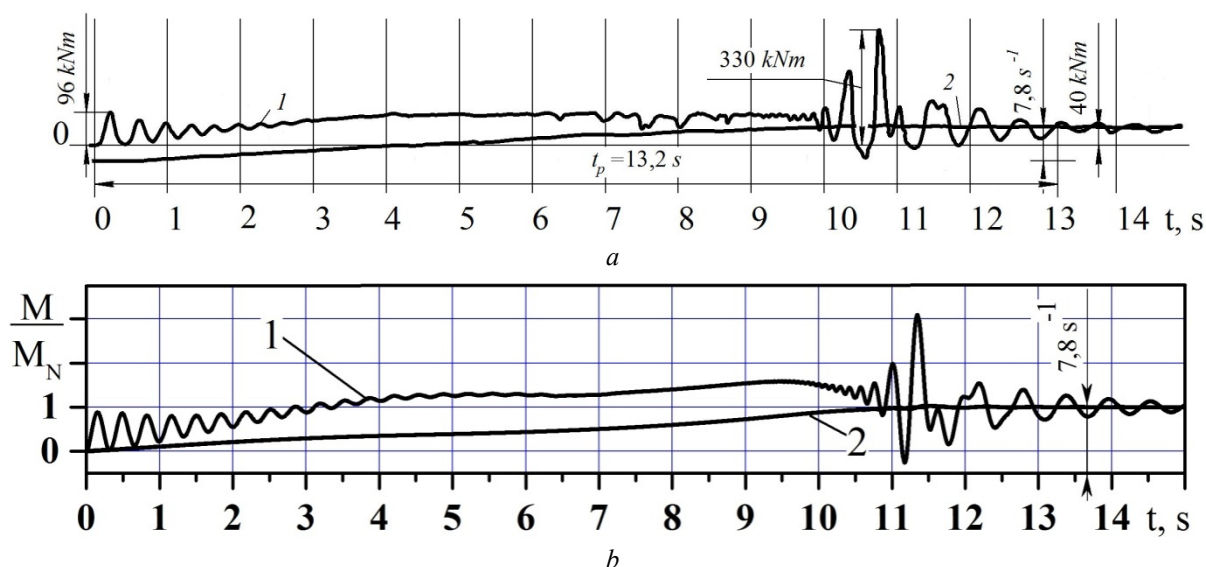


Fig.4. Torque oscillogram for the MMC 90 × 30 mill shafting :
a, bare the experimental and calculated relationships between the torque and the rotational speed;
1 – torque; 2 – rotor rotational speed

Comparison of the experimental and calculated data leads to a conclusion that the proposed mathematical model adequately describes the

dynamic processes in the drive during the start-up period and in a steady state operation.

3. THE ELASTIC CHARACTERISTICS OF BELLOWS FLEXIBLE SHAFT COUPLINGS

The effectiveness of bellows shaft couplings may be shown by an example of the above ball-mill drive. To suit this drive, a bellows flexible shaft coupling should use the model Ya-300 rubber-cord shells as pneumatic elements [6], [7], [12], [17]. With the internal pressure of $p = 0.51$ MPa, load-carrying capacity of the pneumatic element is $Q = 147\,100$ N, and with the initial height of $H = 165$ mm, a compression and back travel is $Z = \pm 40$ mm.

Five pneumatic elements installed in the bellows shaft coupling (Fig.5.a) can transmit a torque of $M_{torque} = 509$ kNm, while eight pneumatic elements will transmit a torque of $M_{torque} = 814$ kNm (Fig.5.c). The largest diameter will be $D_2 = 2200$ mm and the diameter of the pneumoelements installation will be $D_1 = 1384$ mm. To decrease the bellows shaft coupling stiffness and increase its travel, the pneumatic elements may be installed in series (Fig.5b). A further decrease in the stiffness of the bellows shaft couplings is provided by connecting the pneumatic elements to an additional volume.

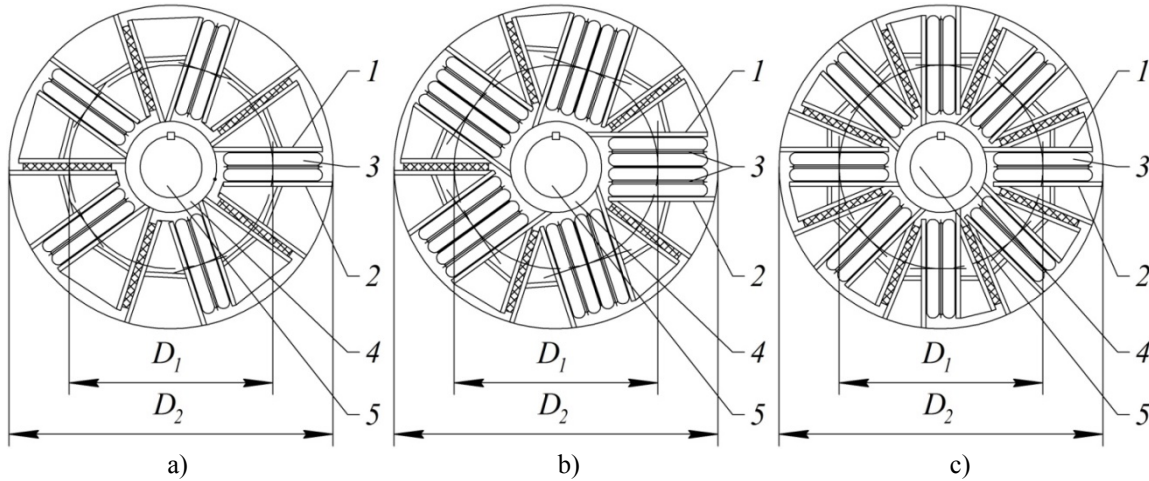


Fig. 5. Flexible couplings with Ya-300 rubber-cord shells:

a, b- torque $M_{torq} = 509$ kN·m, torsional stiffness $c = 4.7 \cdot 10^6$ N·m·rad, and $c = 2.35 \cdot 10^6$ N·m·rad⁻¹ respectively; c- torque $M_{torq} = 814$ kN·m, $c = 7.52 \cdot 10^6$ N·m·rad⁻¹;

1, 2 -half-coupling cheeks; 3 – pneumatic element; 4 – hub; 5 – shaft

To improve the dissipative properties of the coupling, the rubber-cord shells may be filled with a fluid and connected by means of a throttle valve to a hydropneumatic accumulator, which can be installed directly on the shaft [3].

The working portion of viscoelastic characteristic of the coupling is described as follows [8]:

$$M(\alpha) = n_{BELL} R_M S(\alpha) \left[\frac{\rho (S(\alpha))^2 R_M^2}{2 \mu_1^2 f^2} \left(\frac{d\alpha}{dt} \right)^2 + (p_a + p_{uo}) \left(\frac{V_g}{V_g - R_M \int_0^\alpha S(\alpha) d\alpha} \right)^n - p_a \right] \quad (6)$$

$$S(\alpha) = S_0 \cdot (1 + k_a \alpha R_M)$$

where α is the coupling twist angle; $M(\alpha)$ is torque received by the pneumatic flexible coupling; $S(\alpha)$ is an effective area of the rubber-cord shell at an arbitrary time; S_0 is the initial effective area of the rubber-cord shell at rated load; k_a is approximation factor; n_{BELL} V_g is the gas volume in the coupling bellows or in the pneumatic chamber of a hydropneumatic accumulator; n_{BELL} is the number of rubber-cord shells (bellows); p_a , p_{uo} are atmospheric and surplus pressure in the fluid and gas, respectively; ρ is the fluid density; μ_1 is fluid flow rate through the throttling orifice; f is the throttle flow

area; n is polytropic coefficient; $R_M = D_1/2$ is radius of pneumatic elements installation.

The first term in the expression (6) represents the damping forces, and the second term describes the static characteristic of the coupling at $n = 1$:

$$M(\alpha) = n_{BELL} R_M S(\alpha) \left[\frac{(p_a + p_{uo}) \frac{V_g}{R_M}}{\frac{V_g}{R_M} - \int_0^\alpha S(\alpha) d\alpha} - p_a \right] \quad (7)$$

Fig. 6 shows the static characteristic of the flexible couplings under consideration.

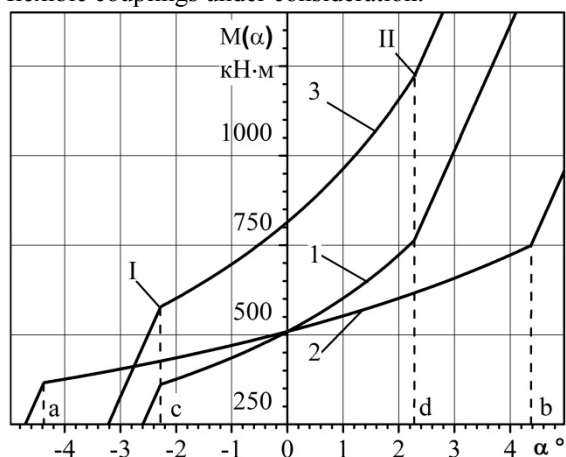


Fig.6. Static characteristic of a flexible coupling with rubber-cord shells

1, 2, 3 – elastic properties of the couplings

Fig.5 a, b, respectively; ab, cd – the coupling travel; the coupling start and closing time

4. ANALYSIS OF THE CALCULATED DATA

Fig. 7 shows the calculated torque dependences in the MMC 90x30 mill drive equipped with a flexible bellows shaft coupling.

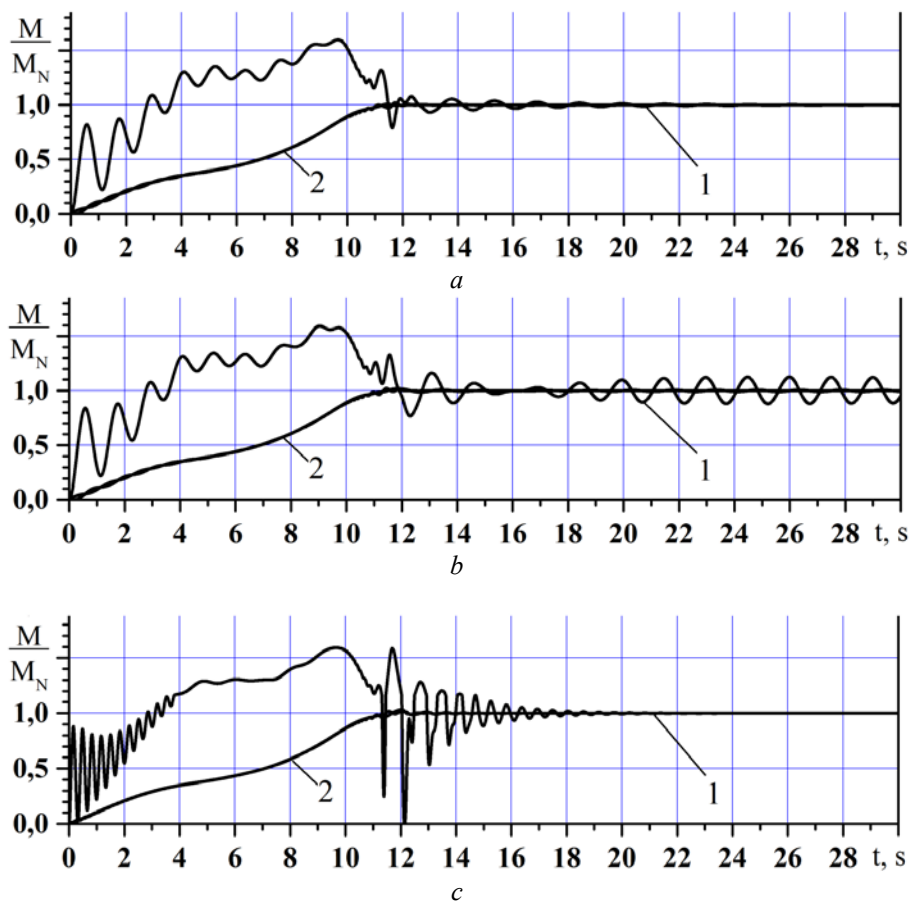


Fig. 7. The calculated torque - time dependences in the MMC 90x30 mill drive:
a, b– bellowsflexibleshaftcoupling;c – bellows flexible coupling with precompression

The calculated data (Fig. 4b and Fig. 7a) show that the use of the bellows flexible coupling has almost halfreduced the peak dynamic loads in the drive power system at the moment of motor coming into synchronism.

On the other hand, a bellows flexible coupling, when used in the drive system, significantly reduces

the stiffness of the mechanical transmission. This, in turn, reduces the frequencies of the system natural oscillations, which may cause resonance modes at low frequencies. In ball mill drives, these low-frequency disturbances may occur in the frequencies of the drum rotation, or the drum charge material rotation, or the lifters - charge material interaction.

Fig. 7b shows the case when a disturbing torque, with an amplitude equal to 2 % of the rated value ($0,02 \cdot M_N$) is acting on the mill drum.

In this case, forced resonance oscillations, with a frequency of $4,21 \text{ s}^{-1}$ and an amplitude of 12% of the rated torque are excited in the flexible system of the drive. The use of bellows flexible couplings can solve this problem. Fig. 7c shows the drive torque dependence for the case when the coupling starts with the torque increased to $1,2 \cdot M_N$, which is achieved by selecting a specific air pressure in the bellows. As a result, the resonance modes at low disturbance frequencies are eliminated, and peak dynamic loads can be limited.

5. CONCLUSION

1. Comparison of experimental and calculated data suggests a conclusion that the presented mathematical model adequately describes the dynamic processes in the synchronous drive of the drum mill [12], [18], [26], [28].

2. Upon analysing the calculated data it has been found that:

- Effective limitation of shock dynamic loads in machines can be achieved by using flexible couplings that incorporate rubber-cord shells as the flexible elements [5], [15], [16], [17];

- Prevention of resonant modes and restriction of forced torque oscillations in the drives can be provided through a proper selection of a limit torque for the coupling actuation, which is determined by an initial air pressure in the rubber-cord shells of the coupling [6], [7], [8], [27].

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His main scientific direction is "Statics and Dynamics in the Mechanical Systems of Production Equipment Drives." He published 277 scientific papers, including two monographs and two textbooks, and he is the owner of 51 patents and inventions. He was awarded the medal "Inventor of the USSR". His major works are devoted to the dynamics and strength of the mechanical systems of ball mills.



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Mr. prof. **J. Homišin** is a university professor nominated in the branch of science "Department of Construction, Automotive and Transport Engineering" with professional specialisation oriented to:

- optimisation of torsionally oscillating mechanical systems with an accent to the control of dangerous torsional oscillations by means of application of flexible shaft couplings, namely by pneumatic flexible shaft couplings,
- diagnostics of torsionally oscillating mechanical systems performed with regard to the characteristics of torsional oscillation and excited mechanical oscillation.

He is a graduate of the Faculty of Mechanical Engineering of the former Technical College (VŠT) of Košice (in 1987). He defended his dissertation thesis in 1990, his habilitation process was finished successfully in 1997 and in March 2004 he was nominated as university professor.

Results of his scientific-research activities are summarized in more than 295 original scientific works published in domestic and foreign professional journals, as well as in the form of professional articles presented in many scientific conferences and symposiums at home and abroad. He is owner of 71 technical patents. There are also another important and relevant publication outputs of his professional life: 3 domestic monographs, 3 foreign monograph, 14 university textbooks and several study textbooks.



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