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HIGH-EFFECTIVE VIBRATION INSULATION SYSTEM OF ROTOR MACHINES USED ON SHIPS AND TRANSPORT VEHICLES

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ABSTRACT
The possibility of obtaining highly effective vibroinsulation of unbalanced rotor machines and at the same time maintaining rather high stiffness of their support system has been discussed in the paper. It could be essential in vibroinsulating equipment installed on ships or other means of transport. The method of synchronous elimination in its off-axial variant has been proposed. An example of its application together with the simulation results have been also presented.

1 - INTRODUCTION
Providing highly effective vibroinsulation of unbalanced rotor machines installed on means of transport (ships, planes, road and rail vehicles etc) in case of low and medium rotational speed machines is very difficult to achieve by a classic vibroinsulation and very expensive and risky when applying typical active elements [2]. This is caused by the necessity of providing a relatively stiff attachment between machine and a vehicle to avoid excessive relative motions generated by accelerations of the vehicle body.
Synchronous elimination method is able to give the solution for this problem [3]. The set of inertial vibrators of axes parallel to the axis of rotation of an unbalanced rotor and individually power-driven by motors of soft mechanical characteristics is being connected to the body of an unbalanced rotor machine. If the rotational direction and the angular nominal velocity of those motors and the rotor are compatible and the machine body is attached in a way permitting its vibration against foundation (the required flexibility is much smaller than for the classic vibroinsulation), then the solution of the system motion equation in which the resultant force generated by correcting vibrators reduces the influence of an unbalanced motor force, can be found. It holds only in the instance when static moments of unbalance and the spatial arrangement of correcting vibrators satisfy conditions (1) and (2).

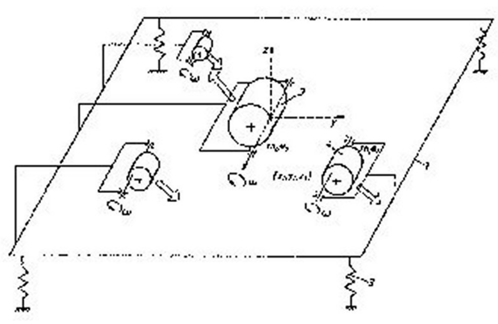

Figure 1: Schematic presentation of a synchronous elimination system: 1 − Machine body, 2 − Unbalanced rotor, 3 − Suspension, 4 − Correcting vibrator.

$$\sum_{i=1}^{n} \chi_i m_i e_i = m_0 e_0 \tag{1}$$

$$\sum_{i=1}^{n} \chi_i m_i x_i = 0 \tag{2a}$$

$$\sum_{i=1}^{n} \chi_i m_i y_i = 0 \tag{2b}$$

$$\sum_{i=1}^{n} \chi_i m_i z_i = 0 \tag{2c}$$

where:

- m_0 − static moment of the rotor unbalance,
- $m_i e_i$ − static moment of the i^{th} correcting vibrator unbalance,
- $x_i,\ y_i,\ z_i$ − coordinates of the i^{th} vibrator in the reference system attached to the rotor − Fig. 1.

Physical realization of the above solution requires it to be stable. That can be achieved by utilization of a natural tendency of elastically supported systems with unbalanced rotors attached, to such arrangement of motion in which rotors move synchronously irrespective of the existing disturbances. Their mutual setting (phase angles) cause either increase or decrease of system vibrations.
Occurrence of a synchronization and its kind (increase or decrease of vibrations) depends on parameters of the system. An analytical criterion of the stability of the periodic solution around phase angle value: $\alpha_{10}, \alpha_{20}, \cdots, \alpha_{n0}$, is the integral condition (3):

$$D\left(\alpha_{10}, \alpha_{20}, \ldots, \alpha_{n0}\right) = \frac{1}{T}\left[\int_0^T (E - V)\,dt - \int_0^T (E_w - V_w)\,dt\right] = \min \tag{3}$$

where:

- α_{i0} − difference between angle of rotation of the i $^{\mathrm{th}}$ vibrator and the rotor at the steady state,
- $E,\ V,$ − kinetic and potential energy of the machine body,
- $E_w,\ V_w$ − kinetic and potential energy of vibrators constrains,
- $T = 2\pi/\omega$ − period of forced vibrations.

2 - FOUR-VIBRATORY ELIMINATION SYSTEM

The application of the synchronous elimination method will be shown on an example of an asymmetrical rotor machine with a statically unbalanced rotor. Taking into account a time variable value of the rotor unbalance the double, four-vibratory, synchronous elimination system was applied, Fig. 2.

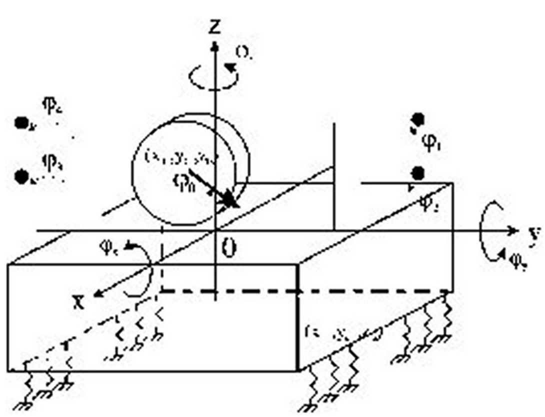

Figure 2: Calculation chart.

In Euler's reference system rearranged as shown in Fig. 3 − at the assumption that Euler's angles are small − the motion equations of the system can be presented as:

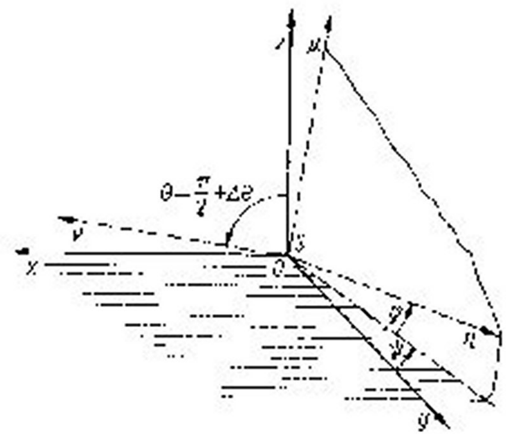

Figure 3: Reference system.

$$[M]\,\frac{d}{dt}\,\{X\} = \{Q\} \tag{4}$$

where:

$$M = \begin{bmatrix} [M_{11}]_{12x12} & [M_{12}]_{12x10} \\ [M_{21}]_{10x12} & [M_{22}]_{10x10} \end{bmatrix}_{22x22}$$

$$[W_{LIN}] = \begin{bmatrix} -[B]_{\text{6x6}} & -[K]_{\text{6x6}} & -[0]_{\text{6x10}} \\ [I]_{\text{6x6}} & [0]_{\text{6 x6}} & [0]_{\text{6x10}} \\ [0]_{\text{5x6}} & [0]_{\text{5 x6}} & [0]_{\text{5x10}} \\ [0]_{\text{6x12}} & [I]_{\text{5x5}} & [0]_{\text{5x5}} \end{bmatrix}_{\text{22x22}}$$

$$Mel_i = \frac{2M_{\max}\left(\omega_0 - \omega_{\max}\right)\left(\omega_0 - \dot{\varphi}_i\right)}{\left(\omega_0 - \omega_{\max}\right)^2 + \left(\omega_0 - \dot{\varphi}_i\right)^2}$$

$$\{Q\} = \{Q_{EULER}\} + [W_{LIN}]\,\{X\} + \{Q_{NLIN}\} + \{Q_{GYR}\}$$

$$\begin{aligned} Q_{EULER}\ \ (4,1) &= (J_\eta - J_\mu)\left(\dot{\Psi}\varphi + \Delta\dot{\theta}\right)\left(\dot{\Psi} - \Delta\dot{\theta}\varphi\right) + J_\nu \dot{\Psi}\Delta\dot{\theta} \\ Q_{EULER}\ \ (5,1) &= (J_\nu - J_\eta)\left(\dot{\Psi}\varphi + \Delta\dot{\theta}\right)\left(-\dot{\Psi}\Delta\theta + \dot{\varphi}\right) + J_\mu\left(\dot{\Psi}\varphi\dot{\varphi} + \dot{\Psi}\Delta\dot{\theta}\Delta\theta + \Delta\dot{\theta}\dot{\varphi}\right) \\ Q_{EULER}\ \ (6,1) &= (J_\mu - J_\nu)\left(\dot{\Psi} - \Delta\dot{\theta}\varphi\right)\left(\dot{\varphi} - \Delta\theta\dot{\Psi}\right) - J_\eta\left(\dot{\Psi}\dot{\varphi} - \dot{\Psi}\varphi\Delta\dot{\theta} - \Delta\dot{\theta}\dot{\varphi}\varphi\right) \end{aligned}$$

$$\{X\} = col\left(\dot{x}, \dot{y}, \dot{z}, \dot{\varphi}, \dot{\Psi}, \Delta\dot{\theta}, x, y, z, \varphi, \Psi, \Delta\theta, \dot{\varphi}_0, \dot{\varphi}_1, \dot{\varphi}_2, \dot{\varphi}_3, \dot{\varphi}_4, \varphi_0, \varphi_1, \varphi_2, \varphi_3, \varphi_4\right)$$

$$\begin{aligned} Q_{GYR}\,(5,1) &= -\sum_{i=0}^{4} J_{wi}\dot{\varphi}_i\dot{\Psi} \\ Q_{GYR}\,(6,1) &= \sum_{i=0}^{4} J_{wi}\dot{\varphi}_i\Delta\dot{\theta} \end{aligned}$$

$$Q_{NLIN}(1) = 0$$
$$Q_{NLIN}(2) = \sum_{i=1}^{4} m_i e_i \dot{\varphi}_i^{\,2} \cos(\varphi_i)$$
$$Q_{NLIN}(3) = \sum_{i=1}^{4} m_i e_i \dot{\varphi}_i^{\,2} \sin(\varphi_i)$$
$$Q_{NLIN}(4) = \sum_{i=1}^{4} m_i e_i \dot{\varphi}_i^{\,2} \left[\eta_i \sin(\varphi_i) - \mu_i \cos(\varphi_i)\right]$$
$$Q_{NLIN}(5) = \sum_{i=1}^{4} m_i e_i \mu_i \dot{\varphi}_i^{\,2} \cos(\varphi_i)$$
$$Q_{NLIN}(6) = -\sum_{i=1}^{4} m_i e_i \eta_i \dot{\varphi}_i^{\,2} \sin(\varphi_i)$$

$$\begin{gathered} Q_{NLIN}(7) = Mel_0(\dot{\varphi}_0) \\ Q_{NLIN}(8) = Mel_1(\dot{\varphi}_1) \\ Q_{NLIN}(9) = Mel_2(\dot{\varphi}_2) \\ Q_{NLIN}(10) = Mel_3(\dot{\varphi}_3) \\ Q_{NLIN}(11) = Mel_4(\dot{\varphi}_4) \end{gathered}$$

where matrixes $[M]$, $[B]$, $[K]$ as in [2].
Euler's angles in the rearranged reference system have the meaning of angular displacements about x, y, and z axes.
The above system has 11 degrees of freedom, in the instance when an analysis of driving units is limited to their static characteristics only. When parameter values of the large rotor machine (fan type) are introduced into the equations of motion the course of separate coordinates showing the machine body position can be received due to the digital simulation, Fig. 4.

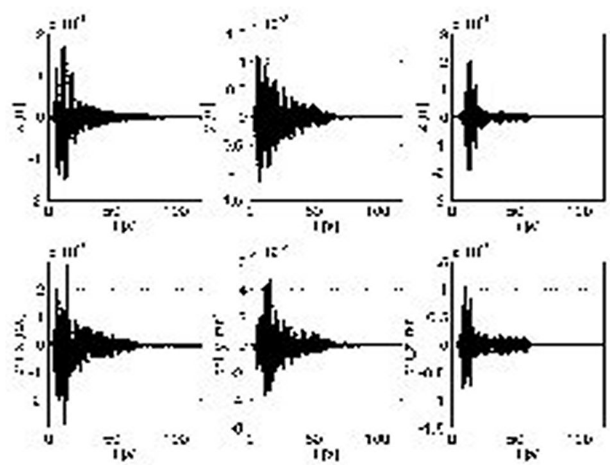

Figure 4: The transient resonance phenomenon as well as the process of the synchronization of correcting vibrators, being in an antiphase towards the main rotor in the super resonance region (t=60s), can be traced in that figure.

The analysis of the amplitude of forces transferred to the foundation shows that the reduction of a dynamic load in a steady state is 5 to 8 times larger in comparison with the case when only the vibroinsulating system was used. It means there is the possibility of several times decrease of force transferred to the foundation and simultaneously retaining the relatively stiff supporting elastic system of the rotor machine. The applied solution does not require any measuring and active devices. Experiments show that there is a farther possibility of improvement of the efficiency of the method in the case when the signal from the vibrating machine body will be utilized for an additional control of the voltage supplying motors of the vibrators.

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