MECHANIKA TEORETYCZNA
I STOSOWANA
Journal of Theoretical
and Applied Mechanics
2, 31, 1993

OFF-AXIAL SYNCHRONOUS ELIMINATION OF VIBRATIONS AND FORCES IN UNBALANCED ROTARY MACHINES

JERZY MICHALCZYK

Technical University of Mining and Metallurgy, Cracow

The possibility of synchronous elimination of vibrations and forces from unbalanced rotary machines using the set of vibrators, axes of which not coincide with the rotor axis has been demonstrated. The analytical conditions of the existence and the stability of synchronous operational cycles, respectively have been derived. The possibilitry of appearance of stable undesired solutions, as well as the preventive methods have been shown. The analytical considerations have been complemented by numerical simulations.

1. Introduction

The passive and semi-active vibration control methods of avoiding the results of the unbalance appearing in the rotary machines rotors allow for the significant reduction in the forces transmitted to the base at the cost of the unwanted loss of the system static stiffness. This often leads to inadmissible machine displacements under the influence of slowly changing external forces. The latter are, for example: the changes of the working load, variations of the belt tension, changes in the static load, etc. In that case the vibrations of the machine diminished – on the contrary, their amplitude increases. Moreover, the passive vibration control systems cause the necessity of passing through resonance frequencies during the starting and coasting of the machine, accompanied by the rise in the vibrations and forces amplitudes to high values. The semi-active systems, i.e. system of controlled elastic or damping constants, have similar properties, however they make the reduction of the rise of the vibrations and forces in the near-resonance range possible. The simultaneous reduction in the machine vibrations, and forces transmitted to the base, may be achieved using the dynamic dampers and active systems with an additional mass (cf Gosiewski, 1989).

The application of the dynamic dampers is limited in practice by the narrow frequency range of their functioning (it pertains especially to the non-damped eliminators) as well as the necessity of the application of a suspension, elastic in the directions of the vibrations elimination.

These disadvantages may be avoided by application of the active systems, in which the forces compensating the forces of unbalancing, arise due to the acceleration of the additional (so called reactive) masses by means of controlled servomotors. A particularly convenient solution for rotary machines is the application of the electromechanically driven inertial vibrators, as active elements, which is the merit of vibrocompensation (cf Genkin, 1977). The aim of the control system in this case is to retain the appropriate vibrator angular velocities and phase angles in relation to the machine rotor. Thus, the rotor rotation angle must be measured and the appropriate control system must be applied, causeing several difficulties in practical applications, rises costs and decreases operational reliability. These disadvantages may be eliminated by the construction of the self-synchronous system adjusting phase angle and angular velocity to the required values (cf Thearle, 1950; Majewski, 1978; Blekhman, 1981; Hogfors, 1984).

These systems, known as *synchronous eliminators*, function properly when the axis of vibrator rotation is in line with the axis of the rotor, what, for technical reasons, is frequently not possible. An example of the design of the synchronous eliminator is that patented by Lipka and Majewski[1]. In that design, the rolling elements, placed freely in a drum rotating at the rotor angular velocity, are automatically positioned in a way causing the damping of the vibrations of masses connected to the drum.

2. Synchronous off-axial elimination

The question arises, whether it is possible to attain the automatic damping of the vibrations caused by the rotor unbalancing, by using the elements of rotation axis distant from the rotor axis.

To answer that question let us consider the system[2]: an unbalanced rotor – a set of inertial vibrators of the axes parallel to the rotor axis and meeting the following requirements

$$\begin{aligned} &(a) \quad \sum m_i e_i = m_0 e_0 \qquad &&(c) \quad \sum m_i e_i y_i = 0 \\ &(b) \quad \sum m_i e_i x_i = 0 \qquad &&(d) \quad \sum m_i e_i z_i = 0 \end{aligned} \tag{2.1}$$

where

[1]Lipka J., Majewski T., *The Method of Automatic Damping of Vibrations and the Assembly for the Automatic Damping.* Patent no. 103855 PL

[2]Michalczyk J., *Assembly for the Elimination of Vibrations and Forces of the Rotary Machines Unbalance,* Patent Appl.OWP,/I/P/268/92

m_0e_0, m_ie_i	–	static moments of the rotor and vibrators unbalance, respectively,
x_i, y_i, z_i	–	coordinates of the vibrators pivoting points, in the system of coordinates of the origin at the nominal (i.e. without correction for the unbalance) rotor center of inertia.

The point of vibrators rotation is defined as the intersection of the rotation plane of the unbalanced mass center with the axis of a given vibrator rotation.

Such a system leads to the compensation of the forces from unbalance and elimination of the vibrations of the rotor body, providing that vibrators are out of the phase rotor unbalance.

On condition 1a the principal vectors of dynamic forces from rotor unbalance are equal to forces activated by the system of eliminators, whereas on conditions (2.1b) ÷ (2.1d) the principal moments, related to any pole, are equal for both masses, for the proper set of the phase angles.

Let us consider the possibility of automatic setting the vibrators in accordance with the requirements (2.1b) ÷ (2.1d). The integral criterion of the automatic synchronization of vibrators (cf Blekhman, 1981) will be used. It states that the existence and stability of the solution corresponding to a definite system of phase angles of the unbalances masses, driven by motors of soft characteristics and of an equal rated rotational speed, requires meeting the conditions of the existence of the following functional minimum

$$D = \frac{1}{T}\int\limits_0^T (E - V)dt \tag{2.2}$$

where

T	–	period of vibrations,
E, V	–	kinetic and potential energy of the system, respectively, without the correction for the energy of the vibrators rotation,

which leads to conditions imposed on the system of phase angles α_i

$$\frac{\partial D}{\partial \alpha_i} = 0 \tag{2.3}$$

$$\frac{\partial^2 D}{\partial \alpha_i^2} > 0 \tag{2.4}$$

The condition (2.2) may be additionally simplified considering only the systems: rotary machine – system of compensations "softly" mounted (i.e. overresonant) – as in this case the potential energy, V, accumulated in the suspending system of the rotor and the compensation system assembly, may be neglected as being many times lower than the kinetic energy of this assembly. In this case the conditions (2.2), (2.3), (2.4) may be replaced by the requirement that the kinetic energy of the system, averaged over the period of vibrations has a local minimum. As the kinetic energy is never negative, it is enough to show the existence of isolated points in the phase angles α_i space, of zero kinetic energy of the basic system (with rotating masses fixed on their rotation axes), which corresponds to the compensation of displacements. We shall investigate the problems connected with the existence of such solutions, on the example of the simple, two-body synchronous eliminator, presented in Fig.1. The unbalanced rotor of a static moment of unbalance, m_0e_0, is mounted on the frame 1, which can rotate around the point 0. The stability of the position of the system static equilibrium is assured by a torsion spring of the rotational spring condstant k, providing that $\sqrt{k/I_0} \ll \omega$, where I_0 denotes the moment of inertia of the basic system.

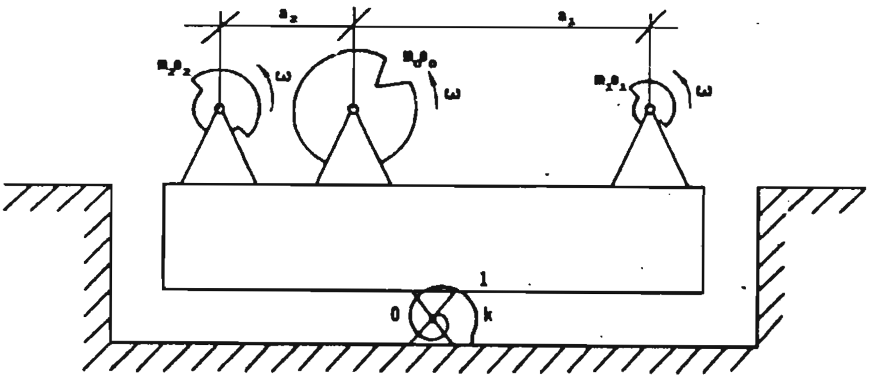

Fig. 1.

When the following conditions are fulfilled

$$m_0e_0 = m_1e_1 + m_2e_2$$
$$m_1e_1a_1 = m_2e_2a_2 \tag{2.5}$$

there exists a system of phase angles for these three rotors, corresponding to the modulation of the vibrators in the phase opposite to the unbalance of rotors. It assures the mutual compensation of forces and then the decrease of the basic system kinetic energy to zero. According to earlier considerations this solution will be stable since any change of the phase angle distribution leads to the unbalance of the system and thus to the appearance of vibrations and to the increase of the system kinetic energy. So, for example, the deviation of one of the vibrators in the phase angle by $d\alpha$ from the position maintaining the balance of forces may be treated as the application of an additional vector dF of the value $Fd\alpha$, perpendicular to the rotating vector F, which disturbs the equilibrium of the system and causes the appearance of the basic system vibrations – Fig.2.

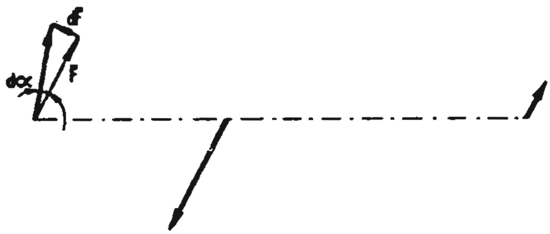

Fig. 2.

It should be noticed, that the indicated solution is not a unique one. To prove it let us observe, that the balance of moments about to the point 0, which is the condition for the absence of vibrations, may be achieved also assuming the phase angles as shown in Fig.3 and the unbalance fulfilling the conditions: $F_0 r_0 = F_1 r_1 + F_2 r_2$, which corresponds to the conditions

$$m_0 e_0 r_0 = m_1 e_1 r_1 + m_2 e_2 r_2 \tag{2.6}$$

The latter solution provides however only the elimination of the vibration due to vanishing of the principal moment about the point 0 and does not provide the elimination of the forces transmitted to the base through the rotation axis of the frame 0, as in this case the principal vector of forces does not reduce to zero.

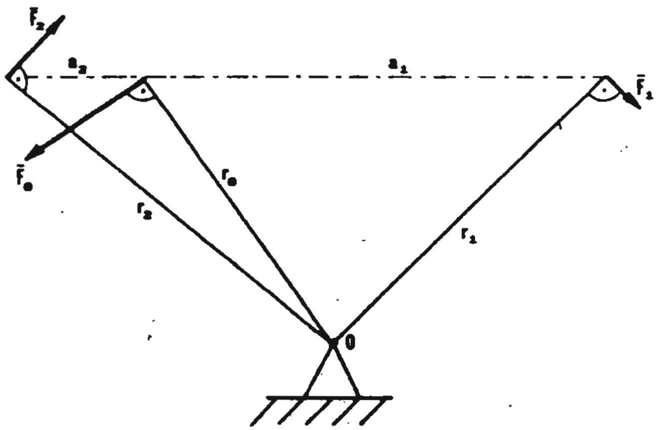

Fig. 3.

Let us assume the system of phase angles, corresponding to the solution of the second type, Fig.3, and calculate the ratio of the maximal moment about the axis of rotation $M_{1,2}$, for the case when the moments of unbalance are chosen according to the condition (2.5) and for the case when these moments fulfill the condition (2.6)

$$\frac{M_{1,2}(2.5)}{M_{1,2}(2.6)} = \frac{F_1 r_1 + F_2 r_2}{F_0 r_0} = \frac{\frac{F_0 a_2}{a_1+a_2} r_1 + \frac{F_0 a_1}{a_1+a_2} r_2}{F_0 r_0} = \frac{\frac{r_1}{r_0} + \frac{r_2 a_1}{r_0 a_2}}{1 + \frac{a_1}{a_2}} \tag{2.7}$$

The analysis of this expression shows that it's value is greater than unity for the non-zero and finite values of parameters assuring the linear configuration of the axes of these three rotors (it may be seen directly from the form of the expression (2.7) for a typical case $|r_0| < |r_1|$ and $|r_0| < |r_2|$). It proves that if the moments of vibrators unbalance are assumed to fulfill the condition (2.5) of the full elimination of vibration and forces, then such vibrators do not provide the elimination of the second type vibrations, because the generated moments exceed the moments of unbalance.

In such a case there exists however another system of the phase angles, providing the zero value of the principal moment and thus the vanishing of the assembly vibrations at the non-zero value of the principal vector of forces. This solution corresponds to the system of phase angles shown in Fig.4, where the vectors of forces, denoted by the superscript $''$, are positioned symmetrically to the vectors of the solution, corresponding to the total elimination (superscript $'$) relative to the normals to radius vectors $\boldsymbol{r}$.

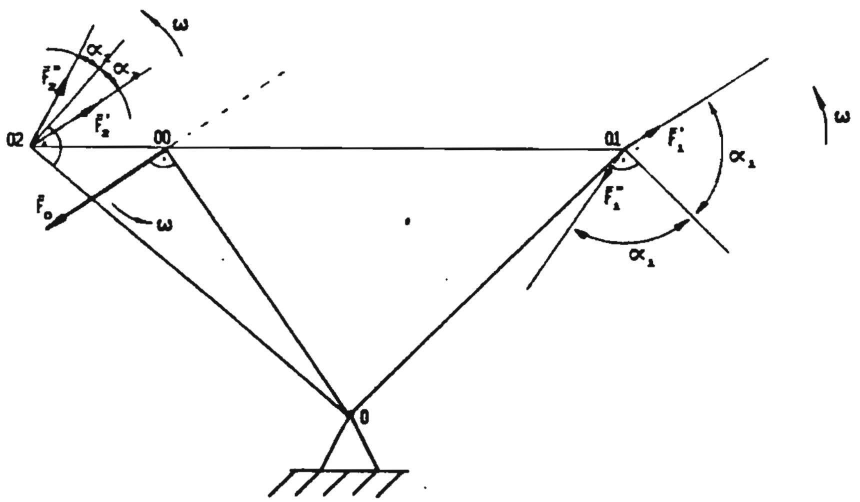

Fig. 4.

It may be proved as follows

– each of the rotating vectors $\boldsymbol{F}_1, \boldsymbol{F}_2$ is a source of the moment $M_{1,2}$, harmonically varying with time, about the point 0;

– each of these moments reaches the extreme value $M_{01,2}$ when the appropriate vector is perpendicular to the respective radius vectors $r_{1,2}$, Fig.3.

Thus, the sum of these moments equals to

$$\begin{aligned} M_{1,2} &= M_{01}\cos(\omega t+\alpha_1)+M_{02}\cos(\omega t+\alpha_2)= \\ &= \sqrt{(M_{01}\cos\alpha_1+M_{02}\cos\alpha_2)^2+(M_{01}\sin\alpha_1+M_{02}\sin\alpha_2)^2}\,\cos(\omega t+\gamma) \end{aligned} \tag{2.8}$$

where $\alpha_{1,2}$ denotes the angles between the respective vectors and half-lines $n_{1,2}$ perpendicular to $r_{1,2}$ for $t=0$

$$\tan\gamma=\frac{M_{01}\sin\alpha_1+M_{02}\sin\alpha_2}{M_{01}\cos\alpha_1+M_{02}\cos\alpha_2}$$

From the relationship (2.8) it results that if one set of phase angles α_1, α_2 causes that a summary moment $M_{1,2}$ has the angle of the phase displacement γ (calculated relative to the normal to the radius r_0) equal to 0, that means that the moments from vibrators balance the moment of rotor unbalance, then the set of angles $-\alpha_1$, $-\alpha_2$ gives also the phase shift angle for the resulting moments equal 0, which is equivalent to balancing the moment from rotor unbalance. All this proves the existence of the solution F_i'', $i=1,2$ (see Fig.4) if the vibrators are chosen according to the condition (2.5), i.e. if there exists a solution F_i', $i=1,2$.

Thus, in the case of choosing the vibrators according to the conditions (2.5) to obtain the complete elimination of vibrations and forces, there exists a second set of phase angles, providing exclusively the elimination of vibrations, which is according to the condition (2.2), stable too. There is then the danger that the system will reach the second undesirable stable state, depending on the initial conditions. Such phenomenon was observed during the computer simulation of the system vibrations corresponding to the diagram from Fig.1, also at the non-zero value of the rotational springing factor and the non-zero damping, which will be discussed further.

It should be noticed that there is no such danger, in the case of the system moving translatory. In this case the radius vectors tend to infinity, which (for the finite difference in their lengths, resulting from the finite dimensions a_1, a_2) causes that the ratio r_1/r_0 and r_2/r_0 tend to unity and also the expression (2.7) tends to unity. It means, that in the case of the system in the translational motion both solutions coincide.

The way of avoiding "indeterminacy" of the stable state for the rotating system by introducing additional bounds between vibrators, for instance by coupling their driving motors by means of the power selsyn system, was given by Michalczyk[3]. Such coupling excludes the second type synchronous elimination and enables the

[3]see footnote no.2 on p.2

system to reach the desired type of synchronization, irrespective on the initial conditions.

Let us consider the problem of the appearance and stability of the first type of solution for the case of synchronous, two-body eliminator, with the rigid bound between vibrators, in the application to the rotary-machine mounted in the way enabling a flat movement of the system – Fig.5.

Moreover, the following designations were assumed

$m_i e_i$	–	static moments of unbalanced masses, $i = 0, 1, 2$;
μ_i, ν_i	–	coordinates of the axes of rotating masses in the central movable system of coordinates $C\mu\nu$;
M, I	–	mass and the central moment of inertia of the system with the unbalanced masses related to their rotation axes;
$F = m_0 e_0 \omega^2$	–	rotating force from the rotor unbalancing, the remaining notations do not change.

The equations of the basic system motion have the following form for the case, when the vibrators fulfill the conditions (2.1)

$$M\ddot{x} + k_x x = F[\cos\omega t + \cos(\omega t + \alpha)] \tag{2.9}$$

$$M\ddot{y} + k_y y = F[\sin\omega t + \sin(\omega t + \alpha)] \tag{2.10}$$

$$I\ddot{\beta} + k_\beta \beta = F[-\nu_0 \cos\omega t + \mu_0 \sin\omega t - \nu_0 \cos(\omega t + \alpha) + \mu_0 \sin(\omega t + \alpha)] \tag{2.11}$$

The solutions of the above written equations have the following form

$$x(t) = \frac{F[\cos\omega t + \cos(\omega t + \alpha)]}{k_x - M\omega^2} \tag{2.12}$$

$$y(t) = \frac{F[\sin\omega t + \sin(\omega t + \alpha)]}{k_y - M\omega^2} \tag{2.13}$$

$$\beta(t) = F[-\nu_0 \cos\omega t + \mu_0 \sin\omega t - \nu_0 \cos(\omega t + \alpha) + \mu_0 \sin(\omega t + \alpha)] \tag{2.14}$$

From the calculations the following form of the functional (2.2) will be obtained

$$\begin{aligned} D &= \frac{\omega}{2\pi} \int_0^{\frac{2\pi}{\omega}} \frac{1}{2} \Big[M(\dot{x}^2 + \dot{y}^2) + I\dot{\beta}^2 - (k_x x^2 + k_y y^2 + k_\beta \beta^2)\Big] dt = \\ &= \frac{1}{2} F^2 (1 + \cos\alpha) \Big(\frac{1}{M\omega^2 - k_x} + \frac{1}{M\omega^2 - k_y} + \frac{\mu_0 + \nu_0}{I\omega^2 - k_\beta} \Big) \end{aligned} \tag{2.15}$$

The condition (2.3) leads to the equation

$$-\frac{1}{2} \sin\alpha F^2 \Big(\frac{1}{M\omega^2 - k_x} + \frac{1}{M\omega^2 - k_y} + \frac{\mu_0 + \nu_0}{I\omega^2 - k_\beta} \Big) = 0 \tag{2.16}$$

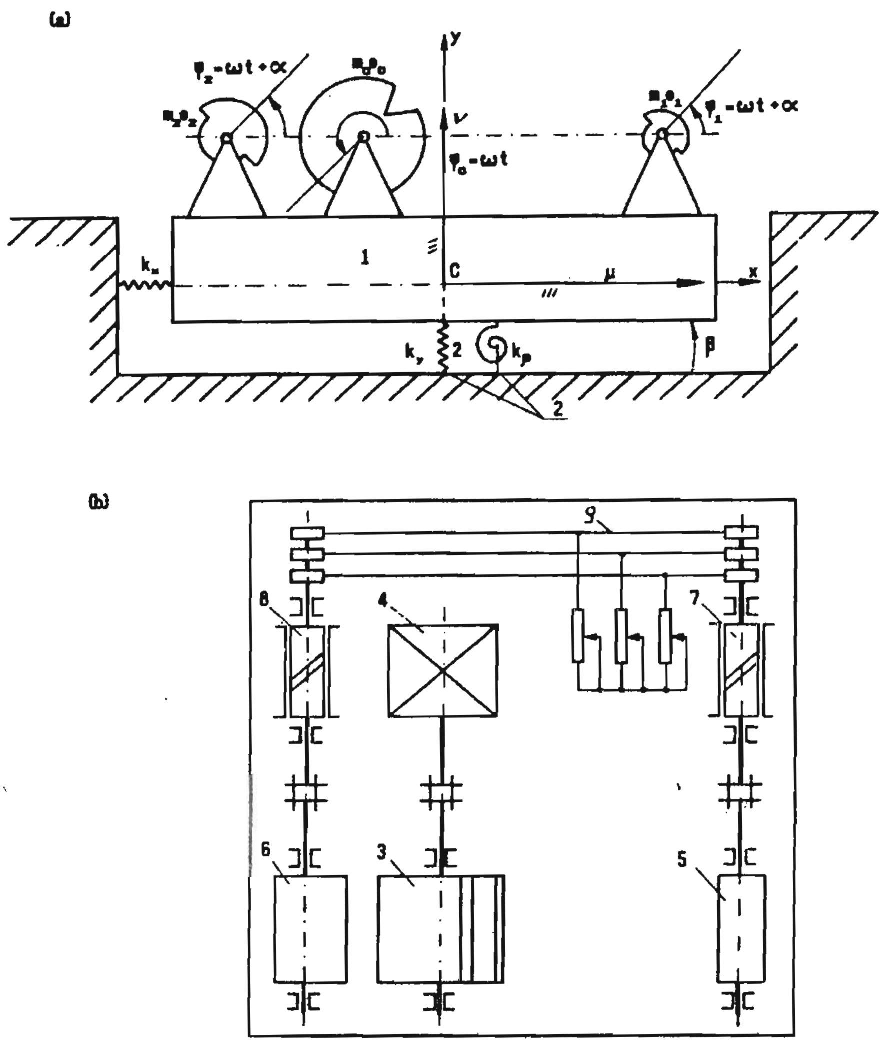

Fig. 5. Denotations: 1 – frame; 2 – suspension; 3 – unbalanced rotor; 4 – rotor driving motor; 5,6 – inertial vibrators; 7,8 – asynchronous drive of vibrators; 9 – power selsyn system

with the root of the equation reading as follows: $\alpha = n\pi$, where $n = 0, \pm 1, \pm 2, ...$

From among these roots the following phase angles fulfill the condition of the vibrations and forces elimination

$$\alpha = \pm\pi(1 + 2n) \qquad n = 0, 1, 2, ... \tag{2.17}$$

which correspond to setting the vibrators out of the phase with the rotor unbalance.

The condition (2.4) corresponds to the inequality

$$\frac{1}{2}F^2(-\cos\alpha)\Big(\frac{1}{M\omega^2 - k_x} + \frac{1}{M\omega^2 - k_y} + \frac{\mu_0 + \nu_0}{I\omega^2 - k_\beta}\Big) > 0 \tag{2.18}$$

For the angles α, given by the expression (2.17) the fulfillment of the stability condition requires the following inequality

$$\frac{1}{M\omega^2 - k_x} + \frac{1}{M\omega^2 - k_y} + \frac{\mu_0 + \nu_0}{I\omega^2 - k_\beta} > 0 \tag{2.19}$$

The more important particular solutions to this inequality are as follows

(a) $M\omega^2 > k_x, k_y \qquad I\omega^2 > k_\beta$

which corresponds to the over-resonant character of the function of the system

(b) $k_x \to \infty \qquad k_y \to \infty \qquad \sqrt{k_\beta/I} < \omega$

which corresponds to the rotational support

(c) k_x or $k_y \to \infty$ and $k_\beta \to \infty$

while the remaining spring constant assures the over-resonant character of one of the translatory movements.

3. Simulation studies

The case (b), previously analyzed theoretically was studied using the numerical simulation, taking into account the existence of the elastic bound and the damping of the rotation, the mechanical characteristics of the inductive driving motors of the rotating units and the full type power selsyn system coupling between vibrators.

The equations of motion taking into account the nonlinear coupling between the motion of the basic system and the motion of the rotors take the form

$$\mathbf{M}\ddot{\mathbf{q}} = Q \tag{3.1}$$

where the coordinates vector $\mathbf{q}$ is given by

$$\mathbf{q} = \mathrm{col}\{\beta, \varphi_0, \varphi_1, \varphi_2\} \tag{3.2}$$

while the matrix $\mathbf{M}$ elements are given by the following relations

$$
\begin{aligned}
M_{11} &= I_c + \sum_{i=0}^{2} m_i(\mu_i^2 + \nu_i^2) & M_{12} &= m_0 e_0(\mu_0 \cos\varphi_0 + \nu_0 \sin\varphi_0)\\
M_{13} &= m_1 e_1(\mu_1 \cos\varphi_1 + \nu_1 \sin\varphi_1) & M_{14} &= m_2 e_2(\mu_2 \cos\varphi_2 + \nu_2 \sin\varphi_2)\\
M_{21} &= m_0 e_0(\nu_0 \sin\varphi_0 + \nu_0 \cos\varphi_0) & M_{22} &= I_0 + m_0 e_0^2\\
M_{23} &= 0 & M_{24} &= 0\\
M_{31} &= m_1 e_1(\nu_1 \sin\varphi_1 + \mu_1 \cos\varphi_1) & M_{32} &= 0 \qquad (3.3)\\
M_{33} &= I_1 + m_1 e_1 & M_{34} &= 0\\
M_{41} &= m_2 e_2(\nu_2 \sin\varphi_2 + \mu_2 \cos\varphi_2) & M_{42} &= 0\\
M_{43} &= 0 & M_{44} &= I_2 + m_2 e_2^2
\end{aligned}
$$

The elements of the forcing vectors are described with the following relationships

$$
\begin{aligned}
Q_1 &= -b_\beta\dot{\beta} - k_\beta\beta + \sum_{i=0}^{2} m_i e_i \dot{\varphi}_i^2[\mu_i \sin\varphi_i - \nu_i \cos\varphi_i]\\
Q_i &= 2M_{uk}\frac{(\omega_s - \dot{\varphi}_k)(\omega_s - \omega_{uk})}{(\omega_s - \omega_{uk})^2 + (\omega_s - \dot{\varphi}_k)^2 + \kappa_i M_u(\varphi_2 - \varphi_1)} \qquad (3.4)\\
i &= 2,3,4 \qquad k = i-2 \qquad \kappa_2 = 0 \qquad \kappa_3 = 1 \qquad \kappa_4 = -1
\end{aligned}
$$

where k_β, b_β are the spring constant and the damping coefficient, respectively, of the supporting system

- M_{uk}, ω_{uk} – the out of the step falling moment and ω of the ith induction motor, respectively,
- M_u – the out of the step falling moment of the correction motors of the electric shaft,
- ω_s – the synchronous rotational velocity of the induction motors.

The interaction of the power selsyn system (cf Michalczyk, 1991) was described with the relationship

$$
\begin{aligned}
M_{el} &= \frac{M_u}{\frac{s}{s_u} + \frac{s_u}{s}}\Big[1 - \cos(\varphi_2 - \varphi_1) \pm \frac{s}{s_u}\sin(\varphi_2 - \varphi_1)\Big] \cong\\
&\cong M_u(\varphi_2 - \varphi_1)
\end{aligned}
\qquad (3.5)
$$

where s, s_u – are the slide and the slide of the falling out of the correction motors.

For the simulation the following values of the parameters were assumed

$I_c = 300\text{kgm}^2$	$m_0 = 200\text{kg}$	$e_0 = 10^{-3}\text{m}$	$\mu_0 = \nu_0 = 0.5\text{m}$
$I_0 = 20\text{kg m}^2$	$I_1 = I_2 = 0.02\text{kg m}^2$	$m_1 = m_2 = 2\text{kg}$	$e_1 = e_2 = 0.05\text{m}$
$\mu_1 = 0$	$\nu_2 = 1\text{m}$	$\gamma_1 = \gamma_2 = 0.5\text{m}$	$\omega_s = 157\text{s}^{-1}$
$k_\beta = 7.5 \cdot 10^5\text{Nm/rad}$	$b_\beta = 250\text{Nms/rad}$		

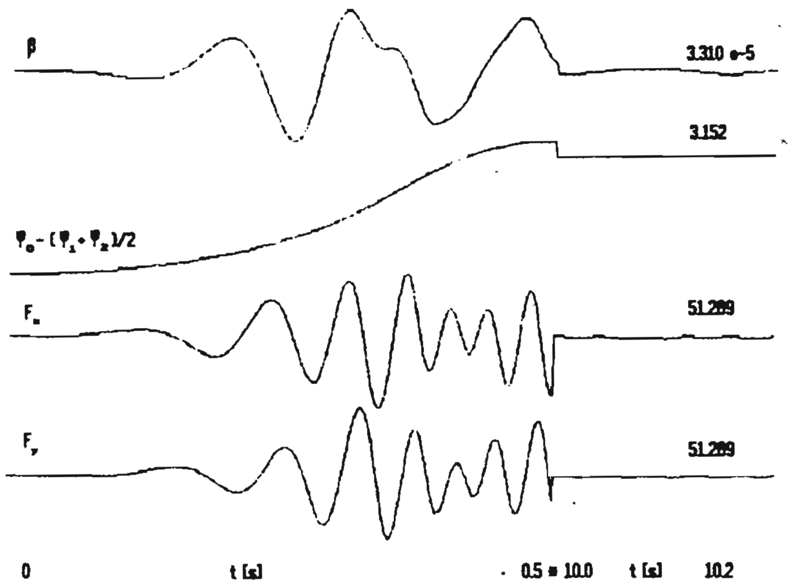

Fig. 6.

The form of the time courses of the chosen parameters, obtained as the result of the simulation are illustrated in Fig.6, where the numerical value of the mean deviation of the vibrators phase angle from the phase angle of the rotor, $\varphi_0 - (\varphi_1 + \varphi_2) = 3.152$, corresponds to the assumed state, while the values given for the angle of frame swinging β and the forces F_x, F_y, transmitted in the horizontal and vertical directions to the base are described by the amplitudes of these values, changing around zero. The diagrams illustrate the run of these values from the position corresponding to the desired solution (of the first type) to $t = 0.5$ s, while the quasi stationary state establishes at the time of $10.0 \div 10.2$ s. One may observe that the vibrators, after the intermediate time during the machine starting fix with a great accuracy out of the phase with the rotor. The error equals $(3.152 - \pi) \times 57.3^{\circ} = 0.57^{\circ}$. The amplitudes of the forces transmitted to the base in the fixed stationary state, equal 51.289 N and make about 1.04% of the value which would load the foundation without the eliminator.

In the case of disconnecting the power selsyn system $(M_u = 0)$ and starting from the initial conditions close to the first type solution, the solution obtained for the stationary state was close to the first type of the solution and the deviations of the vibrators 1 and 2 phase angles from their theoretical positions were equal to 18.3° and -2.6°, respectively. The values of the horizontal and vertical forces, transmitted to the base are 12.0% and 14.6% of the forcing forces, respectively.

In the case of disconnecting the power selsyn system ($M_u = 0$) and starting from the initial conditions close to the second type solution, the solution obtained for the stationary state was close to the second type of the solution and the deviations of the vibrators phase angles from their theoretical positions were equal to $-4.2°$ and $-7.4°$, respectively. The values of the horizontal and vertical forces transmitted to the base are in this case 36.1% and 29.6%, respectively.

The system described above may be applied, after the appropriate multiplication of the pairs of the vibrators, in the case of the unknown or varying value of the unbalance[4], too. Moreover it may be used in the case of the dynamic unbalance of the rotor. In the first two cases the application of two sets of vibrators, instead of the single set of vibrators meeting the requirements (2.1) is sufficient. From these two sets of vibrators each must fulfill the conditions (2.1b) ÷ (2.1d) separately, and all of them together must fulfill one condition 1a ($m_0\ e_0$ means in this case the limiting value of the rotor unbalance, which may appear).

The system chosen in this way, displayes such an automatic adaptability, that the activating forces, acting in both sets of vibrators add or subtract geometrically, so that their sum balances the actual unbalance of the rotor. In the case of the dynamic unbalancing of the rotor there is a need of constructing the system acting in two planes perpendicular to the axis of the rotor. In contrary to the majority of other solutions the described system does not need to cause any deterioration of the time course of the transient resonance during the starting the machine, because it allows formation of the intermediate states of the eliminator independently of the intermediate state of the machine, for example by using the eliminator only for the time of stationary movement of the machine.

In consideration of the point 2 the influence of the energy dissipation, which takes place in real objects, for instance in the supporting elements of the machine, was neglected. This begs some comments. In the typical cases of supporting using the steel springs or the rubber elements of the moderate the contribution of the damping forces (beyond the near-resonance region), is negligible and is limited to generation of the small deviations of the phase angles and incomplete elimination, which results from the low value of the dissipation factors for these elements (cf Lapunov, 1988). This thesis is justified both by theoretical premises – the synchronous elimination is based on the phenomenon of self-synchronization and the effect of damping of the movement of the system must be in both phenomena similar, and by results of the simulation and experimental studies (cf Majewski, 1978), respectively. The extension of the analysis on the case of the considerable damping or the studies of the system movement in the near resonance range will require the application of another analytical methods, for instance the analysis of the vibration moments (cf Blekhman, 1981).

[4]see footnote no.2 on p.2

4. Conclusions

- There exists the possibility of obtaining the synchronous elimination of vibrations and forces of the unbalanced rotary machines using the set of inertial vibrators, (in the simplest case there may be only two vibrators), of axes not overlaping the axis of the rotor. This system must fulfill the conditions (2.1) and the support of the assembly consisting of the machine and the eliminator must fulfill the condition (2.19).
- In the case of the supporting system allowing rotation of the assembly there may appear also a type of eliminator synchronization, different from the desired one and leading to the elimination of vibrations, but not assuring the elimination of forces transmitted to the base. The initial conditions determine the resulting stable state.
- The desired type of synchronous elimination may be achieved by virtue of coupling the eliminator rotors with an additional coupling.

References

1. BLEKHMAN I.I., 1981, *Sinkhronizaciya v prirode i tekhnike,* Nauka, Moskva
2. BLEKHMAN I.I., 1988, *Shto mozhet vibraciya,* Nauka, Moskva
3. GENKIN M.D., ed., 1977, *Vibroizolushchye systemy v mashinakh i mekhanizmakh,* Nauka, Moskva
4. GOSIEWSKI Z., 1989, *Active Control of Rotors Vibrations,* WSI Publishers, Koszalin, (Polish)
5. HOGFORS C., 1984, *Autobalancing,* XVIth Congress of Theoretical and Applied Mechanics, Lyngby, 78-84
6. LAPUNOV V.T. ET ALL., 1988, *Rezinovyje Vibroizolatory,* Sudostroenie, Leningrad
7. MAJEWSKI T., 1978, *Automatic Balancing of the Rotor Elastically Supported in two Directions,* Mechanika Teoret. i Stosowana, 16, 25-39, (Polish)
8. MICHALCZYK J., 1991, *Analysis of Synchronisation Conditions for Inertial Vibrators Using the Power Selsyn System,* Archiwum Budowy Maszyn, XXXVIII, 3, 201-209 (Polish)
9. THEARLE E., 1959, *Automatic Dynamic Balancers,* Machine Design, 9,10,11

Pozaosiowa eliminacja synchroniczna drgań i sił w niewyważonych maszynach wirnikowych

Streszczenie

W pracy wskazano na możliwość uzyskania synchronicznej eliminacji drgań i sił od niewyważenia maszyn wirnikowych za pomocą zespołu wibratorów, których osie nie pokrywają się z osią wirnika. Wyprowadzono analityczne warunki istnienia i stabilności synchronicznych cykli roboczych i wskazano na możliwość wystąpienia stabilnych rozwiązań niepożądanych oraz sposób ich uniknięcia. Rozważania analityczne uzupełniono badaniami metodą symulacji cyfrowej.

Manuscript received June 9, 1992; accepted for print November 6, 1992