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DISTURBANCES IN SELF-SYNCHRONISATION OF VIBRATORS IN VIBRATORY MACHINES

ZABURZENIA SAMOSYNCHRONIZACJI WIBRATORÓW W MASZYNACH WIBRACYJNYCH

An influence of elastic support elements arrangements and ratios of elasticity and damping constants in vertical and horizontal direction on self-synchronisation accuracy was investigated in the paper. The obtained results of the other factors influence on disturbances in self-synchronisation of vibrators in vibratory machines are also presented. Especially, influence of a diversification of driving and anti-torque moments, not central direction of the resulting force of the vibrators set , the local flexibility of mounting of vibrators to the machine body and influence of collisions with a feed were described.

Keywords: vibratory machines, self-synchronisation, phase incompatibility of vibrators motion

W pracy zbadano wpływ rozmieszczenia elementów sprężystych zawieszenia i proporcji pomiędzy stałymi sprężystości i tłumienia tych elementów na kierunkach pionowym i poziomym na dokładność samosynchronizacji wibratorów. Przytoczono też rezultaty badań autorów nad wpływem innych przyczyn na zaburzenia samosynchronizacji wibratorów maszyn wibracyjnych. W szczególności określono wpływ na zjawisko samosynchronizacji takich zjawisk jak: zróżnicowanie momentów napędowych i oporowych, niecentralne ukierunkowanie wypadkowej siły wymuszającej zespołu wibratorów, lokalna podatność zamocowania wibratorów do korpusu maszyny i oddziaływanie zderzeń z nadawą.

Słowa kluczowe: maszyny wibracyjne, samosynchronizacja, niewspółfazowość ruchu wibratorów

1. Self-synchronisation effect

Achieving a synchronous, cophasal vibrators running is the main problem of vibratory machines applied in the mining, metal-forming, foundry and building industry. The effect of self-synchronisation of vibrators, being a spontaneous equalisation of their rotational speed and phases has been known and applied in machines of two or more drives since half of the century

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(Blechman, 1971). Its essence is based on a creation of additional moments, called *vibratory moments*, in elastically supported machines excited for vibrations by inertial vibrators. These moments adding themselves to driving moments of motors of soft characteristics cause mutual readjusting of rotational speeds and phase angles of driving units. An example of such machine is shown in Fig. 1, presenting schematically the vibratory conveyor supported on a set of parallel leaf springs.

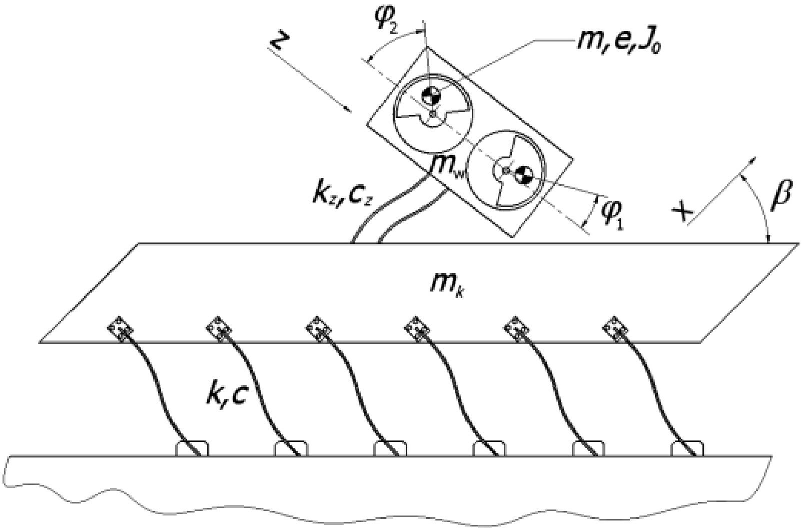


Fig. 1. Calculation model of over-resonance vibratory conveyor;

k, c, k_z, c_z — elastic and damping constants of the main supporting system and the supporting system of the vibrator in their motion direction x and z – respectively

m, e — mass and eccentric of the vibrator,

J_o — moment of inertia of a rotor of the electrovibrator calculated in relation to its axis of rotation

The desirable character of counter running vibrators means obtaining equal angles: $\varphi_1 = \varphi_2$. This allows to achieve the resultant excitation force in the perpendicular direction to the conveyor springs and limits a dynamic influence of the machine on its foundation.

Information whether a system has a self-synchronisation tendency and what type of phase dependence exists in between angles $\varphi_1, \varphi_2, \varphi_3 \ldots \varphi_n$, can be obtained, relatively simply, by means of the *I.I.Blechman's integral criterion* (Blechman, 1994), in which the phase angle system is considered stable when it minimises the function:

$$D(\varphi_1-\varphi_2,\varphi_1-\varphi_3,\ldots,\varphi_1-\varphi_n) = \frac{1}{T}\left[\int_0^T (E-V)dt - \int_0^T (E_w - V_w)dt\right] = \min \tag{1}$$

where E, V and E_w, V_w are the kinetic and potential energy of the linearised basic system and constrains between vibrators – respectively. These values are calculated for the steady motion state. The application of the above criterion to the system shown in Fig. 1 leads to the conclusion that achieving the needed synchronisation: $\varphi_1 - \varphi_2 = 0$ for the over-resonance tuning of the conveyor requires the degree of freedom z and an over-resonance mass m_w tuning on elasticity k_z. More precisely it means, that the positive sign of the expression under the module in dependence (2) is required, (Blechman, 1971).

However, in actual systems several factors – mainly related to not full symmetry of vibratory drives or their placement – occur, which cause that their self-synchronisation is not accurate or even impossible to be obtained.

Practical experiments (Lawendel, 1981) indicate ranges of permissible angular differences of vibrators running, thus e.g.:

– for conveyors the disphasing angle $\Delta\varphi$ should not exceed 12-16°
– for feeders this angle should be within 5-12°,
– for vibrating screens it should not exceed 3-5°.

2. Influence of a diversification of driving and anti-torque moments

Criterion (1) does not answer the question, how accurate will be a self-synchronisation of vibrators in case of differences in e.g. vibrators resistance to motion or inaccuracy of their preparation or assembling.

A qualitative answer for the first question can be obtained by means of the *vibratory moments method* (Lawendel, 1981). Thus, e.g. for the system shown in Fig. 1 – to prevent inaccuracy of cophasing exceeding 12 to 16°, which is required for this type of machines, it is enough to fulfil the condition:

$$\frac{m^2 e^2 \omega^4}{2(m_k + m_w)} \cdot \left| \frac{1}{k/(m_k + m_w) - \omega^2} - \frac{m_k + m_w}{m_w} \cdot \frac{1}{k_z / m_w - \omega^2} \right| \geq (0.5 \div 1.0)\ \mathcal{M}_{zn} \tag{2}$$

where $\mathcal{M}_{zn}$ means the rated value of the induction motor driving moment.

The problem of the influence of diversification of driving-anti-torque moments on the self-synchronisation occurs also in systems supported in a way allowing the machine **plane motion** – Fig. 2

By means of *the averaging method* (Hayasi, 1964) it was shown, in (Michalczyk, 2010), that in case of diversification of driving and anti-torque moments in the system presented in Fig. 2, the phase angle $\Delta\varphi(t)$, can be determined on the bases of the differential equation:

$$J_{zr}\Delta\ddot{\varphi} + (a_{el} + a_o)\Delta\dot{\varphi} + K\sin\Delta\varphi = (\omega_{ust1} - \omega_{ust2})(a_{el} + a_o) \tag{3}$$

where:

$a_{el} = 2p\mathcal{M}_{zn}/(\omega_s - \omega_u)$,

p — motor over-load capacity,
$\mathcal{M}_{zn}$ — motor rated moment,
ω_s — synchronous angular velocity,
ω_u — stall angular velocity,
ω_{ust1}, ω_{ust2} — natural, generally slightly different, angular velocity of drives 1 and 2 — at the motionless machine body.

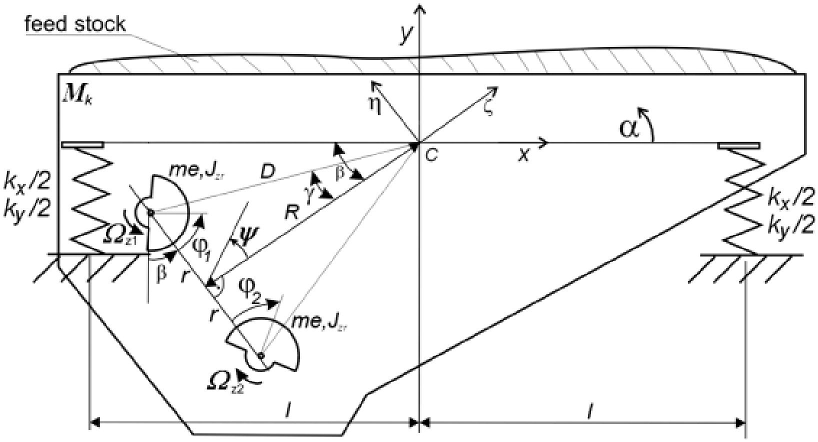


Fig. 2. Two-vibrator over-resonance vibratory machine in plane motion

m	—	unbalanced mass [kg],
e	—	eccentric of a rotor unbalance [m],
M_k	—	machine body mass [kg],
$M = M_k + 2m$	—	mass of the vibrating part of the machine (body and vibrator) [kg],
J	—	central moment of the body inertia with unbalanced masses brought to the vibrator axis of rotation [kgm^2],
J_{zr}	—	moment of inertia of the vibrator together with engine, calculated versus the axis of rotation [kgm^2],
ω	—	angular velocity [1/s],
k_x, k_y, k_ξ, k_η	—	coefficients of elasticity of the body supporting system, along axes: x, y and ξ, η [N/m] – respectively,
$\overline{\Omega}_{z1}, \overline{\Omega}_{z2}$	—	moments exerted to the rotating masses originated from the drive and resistance to motion [Nm],
Ψ	—	see item 4

$$a_o \approx \mu m e \omega_{ust}\, d$$

where:

μ — substitutive coefficient of friction for rolling bearings,

d — bearing journal diameter,

Remaining notations the same as before.

A physical interpretation of equation (3) as a pendulum equation, leads to the conclusion that **obtaining of a synchronous motion is not possible when the following condition is met**:

$$\left|\omega_{ust1} - \omega_{ust2}\right| (a_{el} + a_o) > K \tag{4}$$

where:

$$K = m^2 e^2 \omega^4 \left[\frac{D^2}{J\omega^2 - k_y l^2} + \frac{1}{M\omega^2 - k_\eta} - \frac{1}{M\omega^2 - k_\xi} \right] \tag{5}$$

When the condition (4) is not met, the synchronisation depends on initial conditions and can be investigated by analysing equation (3).

Especially for $K > 0$:

$$\Delta\varphi = \arcsin[\frac{(\omega_{ust1} - \omega_{ust2})(a_{el} + a_o)}{K}] \tag{6}$$

3. Influence of not central direction of the resulting force of the vibrators set

The influence of not central position of the axis of symmetry of the two-vibrator driving system on the accuracy of vibrators self-synchronisation and undesired angular oscillations of the machine body (Fig. 3) was considered in paper (Michalczyk, 2012a).

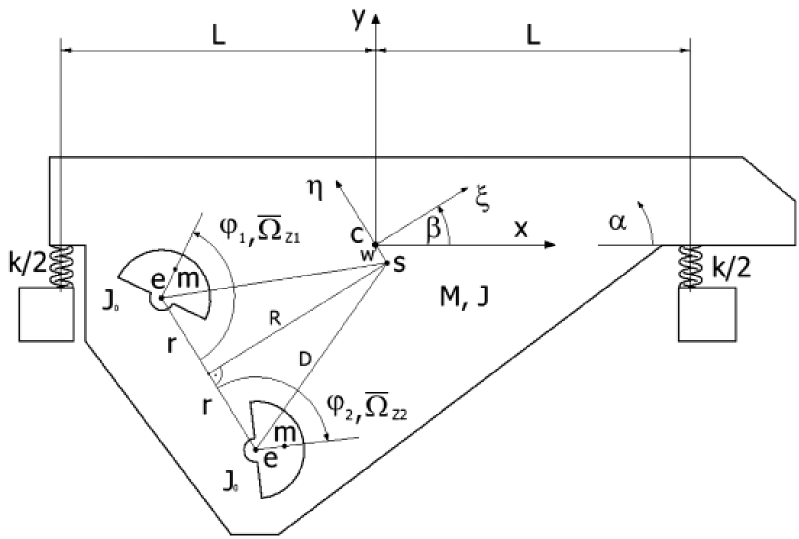


Fig. 3. Two-vibrator over-resonance vibratory machine calculation model.

C — mass centre of the body with unbalanced masses brought to the vibrator axis of rotation,
S — centre of symmetry of the driving system,
w — eccentricity of the force direction,
$M = M_k + 2m$ — mass of the vibrating part of the machine (body and vibrator) [kg],
M_k — machine body mass [kg],
m — unbalanced mass of the vibrator,
e — eccentric of the rotor unbalance [m],
J — central moment of the body inertia with unbalanced masses brought to the vibrator axis of rotation [kgm^2],
J_o — moment of inertia of the vibrator, calculated versus the axis of rotation [kgm^2],
J_w — moment of inertia of the engine rotor, reduced versus the vibrator axis of rotation [kgm^2],
$J_{zr} = J_o + J_w$ [kgm^2],
ω — angular velocity of vibrators [1/s],
k_x, k_y, k_ξ, k_η — coefficients of elasticity of the body supporting system k – along axes: x, y and ξ, η [N/m] – respectively,
$\overline{\Omega}_{z1}, \overline{\Omega}_{z2}$ — moments exerted to the rotating masses originated from the drive and resistance to motion [Nm].

Remaining notations the same as before

It was found, that the deviation of the nominal direction of the excitation force from the system mass centre by w causes an occurrence of the disphasing angle of vibrators, $\Delta\varphi$, determined by the equation:

$$\tan\left(\frac{\Delta\varphi}{2}\right) = \frac{\dfrac{wR}{J\omega^2 - k_y l^2}}{\dfrac{D^2}{J\omega^2 - k_y l^2} + \dfrac{1}{M\omega^2 - k_\eta} - \dfrac{1}{M\omega^2 - k_\xi}} \tag{7}$$

Non-zero value of the disphasing angle leads – in the system shown in Fig. 3 – to angular oscillations of the body in the machine symmetry plane. The amplitude of these oscillations is as follows:

$$A_\alpha \cong \frac{2me\omega^2 w}{J\omega^2 - k_y l^2} \cdot \frac{\sqrt{r^4 + R^2 (r-w)^2}}{D^2} \tag{8}$$

The way of determining the trajectory of an arbitrary body point performing motion combined of the desired translatory motion and of the harmful angular body oscillations was also indicated in the quoted paper.

4. Influence of the local flexibility of mounting vibrators to the machine body.

The scheme of the two-vibrator vibratory machine shown in Fig. 2, can be also used for analysing systems in which vibrators are not sufficiently stiffly mounted to the machine body. Such cases can be found in industrial structures (Banaszewski, 1990), where they are often reasons of an improper synchronisation of vibrators. Ψ – in Fig. 2 – marks the direction, along which the stiffness of the drive mounting to the machine body is not sufficient.

Such cases were theoretically analysed (Michalczyk, 2012b) and it was found that:

1° A self-synchronisation character is not changing in case of an *individual* flexibility of vibrators mounting to the machine body.

2° In case when the *mutually stiff system of two vibrators* is not sufficiently mounted to the machine body in direction ψ (Fig. 2), the disphasing angle, $\Delta\varphi$, of vibrators can be determined from the equation:

$$\left[\frac{D^2}{J\omega^2 - k_y l^2} + \frac{1}{M\omega^2 - k_\eta} - \frac{1}{M\omega^2 - k_\xi}\right]\sin\Delta\varphi + \frac{1}{k_\psi}\cdot\sin(\Delta\varphi - 2\psi) = 0 \tag{9}$$

where k_ψ denotes the coefficient of elasticity of mounting this set in the direction ψ.

3° When the *vibrators set* – apart from the elasticity of mounting in the direction ψ – has the identical elasticity in the direction $\psi + \pi/2$, equation (9) obtains the identical form as for the stiff mounting, which means that the set is not sensitive to this type of elasticity.

4° When the *vibrators set* is mounted to the machine body with rotational flexibility k_o [Nm/rad], in such a way that it can perform angular oscillations versus its centre of symmetry, the equation for the vibrators disphasing angle, $\Delta\varphi$, obtains the form:

$$\left[\frac{D^2}{J\omega^2 - k_y l^2} + \frac{1}{M\omega^2 - k_\eta} - \frac{1}{M\omega^2 - k_\xi} - \frac{r^2}{k_o}\right] \sin \Delta\varphi = 0 \tag{10}$$

Since the sign of the coefficient in square bracket in equation (10) decides on the synchronisation character, as it is seen from this dependence form, too small rotational stiffness of the vibrators set placement k_o can lead to the change of the stable value of the vibrators phase angle $\Delta\varphi$ from 0 (desired value) into $\pm\pi$. This would cause the vibrations decay in the working direction and undesirable movement in the transverse direction.

Thus, the condition for the proper synchronisation has in this case the form:

$$\frac{r^2}{k_o} + \frac{1}{M\omega^2 - k_\xi} < \frac{D^2}{J\omega^2 - k_y l^2} + \frac{1}{M\omega^2 - k_\eta} \tag{11}$$

5. Influence of collisions with a feed

It was shown in paper (Michalczyk & Czubak, 2010), that the collision of the vibratory machine with the transported material (feed), can also lead to dissynchronising of vibrators. For the system corresponding with the one shown in Fig. 2, the disphasing angle $\Delta\varphi$, can be determined from the equation:

$$\frac{2m_n \cdot g}{M} \cos\beta \sin\varphi_0 = m\ e\ \omega^4 \left[\frac{D^2}{J\omega^2 - k_y l^2} + \frac{1}{M\omega^2 - k_\eta} - \frac{1}{M\omega^2 - k_\xi}\right] \Delta\varphi \tag{12}$$

where: g — acceleration of gravity, $\varphi_o = \omega t_3 + \pi$, while t_3 — time of the feed collision with the machine body, calculated in relation to the initial moment $t = 0$ assumed in the instant of obtaining the maximum speed by the machine body (Michalczyk, 1995), m_n — mass of a material feed.

The form of equation (12) indicates the possibility of avoiding the vibrators dissynchronisation and angular oscillations of the machine body. Thus, it is enough to ensure that $\sin\varphi_o = 0$ in the moment when the feed is colliding with the machine body. For the single-stroke motion it occurs when $t_3 = \dfrac{\pi}{\omega}$ or $t_3 = \dfrac{2\pi}{\omega}$, it means when the collision occurs in the moment of the machine body passing through the balance point. On the bases of the analysis of the feed motion (Michalczyk, 1995) it can be stated, that this case occurs for the coefficients of throw: $k_p = 1.14$ and $k_p = 2.97$. Since the value $k_p = 1.14$ is the most often insufficient for the efficient technological process, assuming $k_p = 2.97$ can be recommended for avoiding the machine body oscillations causing uneven amplitudes distribution along the body.

However, the simulation investigations (Michalczyk & Czubak 2010) indicate, that for the **loose material feed** the minimum of the influence of the collisions with the feed material on disphasing of vibrators occurs for slightly lower value $k_p = 2.7$. The maximum of this influence occurs for $k_p = 1.75$.

6. Influence of not central position of the elasticity centre of the supporting system

An influence on drives self-synchronisations the way of arrangement of elements of elastic machine supports with respect to the machine body mass centre, as well as an influence of the elasticity ratio on vertical and horizontal directions of these elements, constitute problems not investigated until now. Usually, due to the necessity of ensuring equal values of static deflections of elastic elements under the machine load and due to benefits resulting from uncoupling of equations of motion, the symmetrical arrangement of supporting elements distributed in the horizontal plane versus the central vertical axis, is applied. However, in consideration of constructional reasons, this plane not always crosses the machine body mass centre – Fig. 4.

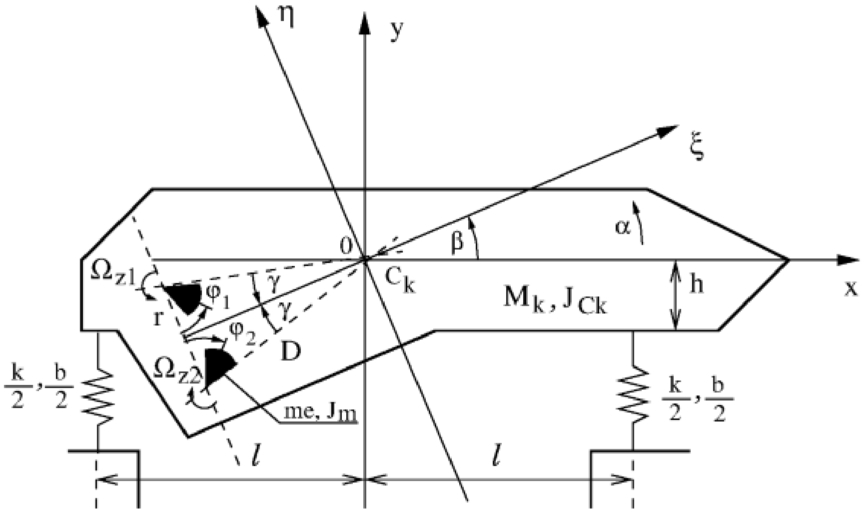


Fig. 4. Dynamic scheme of the vibratory machine.

M_k — machine body mass, without rotating elements of vibrators,
C_k — mass centre of M_k,
J_{Ck} — moment of inertia of the machine body versus the central axis,
me — static moment of vibrator unbalance,
J_m — central inertial moment of rotating parts of a vibrator,
$\overline{\Omega}_{z1}, \overline{\Omega}_{z2}$ — sum of driving and resistance moments,
k, b — elasticity and damping coefficients of the machine body suspension

Due to the necessity of taking into account the damping influence in the suspension system, the digital simulation was applied.

The mathematical model applied for the system description, after utilising the fact that the vibration amplitude is significantly smaller than the linear dimensions of the machine, is of the form:

$$M_k \ddot{x} = P_{x1} + P_{x2} - k_x(x + \alpha h) - b_x(\dot{x} + \dot{\alpha} h) \tag{13}$$

$$M_k \ddot{y} = P_{y1} + P_{y2} - k_y y - b_y \dot{y} \tag{14}$$

$$\begin{gathered} J_{Ck}\ddot{\alpha} = P_{x1} D\sin(\beta-\gamma) + P_{x2} D\sin(\beta+\lambda) - P_{y1} D\cos(\beta-\gamma) - \\ P_{y2} D\cos(\beta+\gamma) - k_x(x+\alpha h)h - b_x(\dot{x}+\dot{\alpha}h)h - k_y l^2\alpha - b_y l^2\dot{\alpha} \end{gathered} \tag{15}$$

$$J_{zr}\ddot{\varphi}_1 = \Omega_{z1} + P_{x1} e\cos(\varphi_1+\beta) + P_{y1} e\sin(\varphi_1+\beta) \tag{16}$$

$$J_{zr}\ddot{\varphi}_2 = \Omega_{z2} + P_{x2} e\cos(\varphi_2-\beta) - P_{y2} e\sin(\varphi_2-\beta) \tag{17}$$

$$P_{x1} = -m\ddot{x} - mD\ddot{\alpha}\sin(\beta-\gamma) + me\dot{\varphi}_1^2\sin(\varphi_1+\beta) - me\ddot{\varphi}_1\cos(\varphi_1+\beta) \tag{18}$$

$$P_{x2} = -m\ddot{x} - mD\ddot{\alpha}\sin(\beta+\gamma) + me\dot{\varphi}_2^2\sin(\varphi_2-\beta) - me\ddot{\varphi}_2\cos(\varphi_2-\beta) \tag{19}$$

$$P_{y1} = -m\ddot{y} + mD\ddot{\alpha}\cos(\beta-\gamma) - me\dot{\varphi}_1^2\cos(\varphi_1+\beta) - me\ddot{\varphi}_1\sin(\varphi_1+\beta) \tag{20}$$

$$P_{y2} = -m\ddot{y} + mD\ddot{\alpha}\cos(\beta+\gamma) + me\dot{\varphi}_2^2\cos(\varphi_2-\beta) + me\ddot{\varphi}_2\sin(\varphi_2-\beta) \tag{21}$$

$$\Omega_{z1} = \frac{2pM_{zn}(\omega_s-\dot{\varphi}_1)(\omega_s-\omega_u)}{(\omega_s-\omega_u)^2+(\omega_s-\dot{\varphi}_1)^2} - \rho me\dot{\varphi}_1^2\,\mathrm{sgn}(\dot{\varphi}_1-\dot{\alpha}) \tag{22}$$

$$\Omega_{z2} = \frac{2pM_{zn}(\omega_s-\dot{\varphi}_2)(\omega_s-\omega_u)}{(\omega_s-\omega_u)^2+(\omega_s-\dot{\varphi}_2)^2} - \rho me\dot{\varphi}_2^2\,\mathrm{sgn}(\dot{\varphi}_2+\dot{\alpha}) \tag{23}$$

The equations, shown above, allow to determine time-histories of coordinates: $x, y, \alpha, \varphi_1, \varphi_2$ and components of forces: P_{x1}, P_{y1}, P_{x2}, P_{y2} in vibrators bearings and moments: Ω_{z1}, Ω_{z2}.
where:

$J_{zr} = J_m + J_w$

J_w — moment of inertia of the motor rotor reduced to a vibrator shaft,

$\mathcal{M}_{zn}$ — rated moment of the induction driving motor,

p — over-load capacity of the motor,

ω_s — synchronous angular velocity,

ω_u — stall angular velocity,

ρ — constant of vibrator bearings resistance,

$k_{x,y}$, $b_{x,y}$ — elasticity and viscous damping constants of elastic elements along corresponding axes.

Damping constants were assumed as:

$$b = 2\xi_{1,2}\sqrt{(M_k+2m)k}$$

where $\xi_{1,2}$ are coefficients of a relative damping for suspensions on springs (1) or on rubbers (2).

Simulations were performed for the following parameters values:

$M_k = 2000$ [kg],

$J_{Ck} = 2667$ [kgm^2],

$me = 25$ [kgm],

$J_{zr} = 0.5$ [kgm^2],

$l = 1$ [m],

$\beta = 30$ [°],

$\gamma = 14$ [°],

$D = 0.8$ [m],

$h = variable$,

$k_{x,y} = variable$,

$\xi_1 = 0.01$,

$\xi_2 = 0.07$,

$\mathcal{M}_{zn} = 40$ [Nm],

$p = 2.5$,

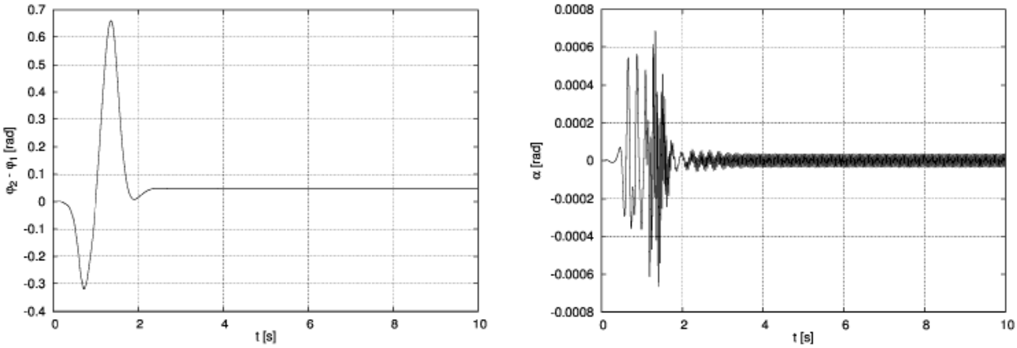
$\omega_s = 157.1$ [1/s],

$\omega_u = 126$[1/s].

When the system obtained the steady state, the disphasing angle $\Delta\varphi$ and amplitude of angular oscillations of the machine body A_α, were estimated. The results obtained for various combinations of parameters are listed in Table 1, while the time-history of steadying of $\Delta\varphi$ and A_α for various parameters of the machine body suspension system are shown in Figures: 5, 6, 7 and 8.

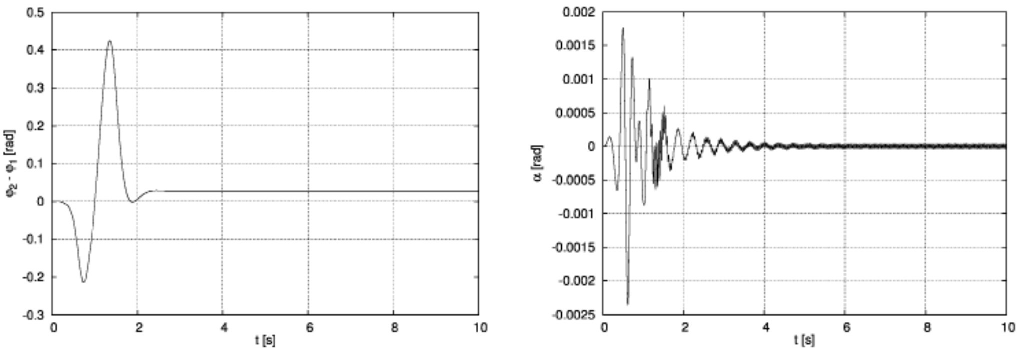
TABLE 1

Number	k_x [N/m]	k_y [N/m]	b_x [Ns/m]	b_y [Ns/m]	h [m]	α [rad]	$\Delta\varphi$ [rad]
				$\xi = 0.07$			
1	7.46E+05	2.07E+06	782.2	1303.7	0.0	3.5E-05	0.047
2	7.46E+05	2.07E+06	782.2	1303.7	0.5	3.71E-05	0.027
3	2.07E+06	2.07E+06	1303.7	1303.7	0.0	0.00	0.000
4	2.07E+06	2.07E+06	1303.7	1303.7	0.5	1.52E-05	-0.052
				$\xi = 0.01$			
5	7.46E+05	2.07E+06	782.2	1303.7	0.0	3.55E-05	0.048
6	7.46E+05	2.07E+06	782.2	1303.7	0.5	3.58E-05	0.031
7	2.07E+06	2.07E+06	1303.7	1303.7	0.0	0.0	0.0
8	2.07E+06	2.07E+06	1303.7	1303.7	0.5	0.51E-05	0.046



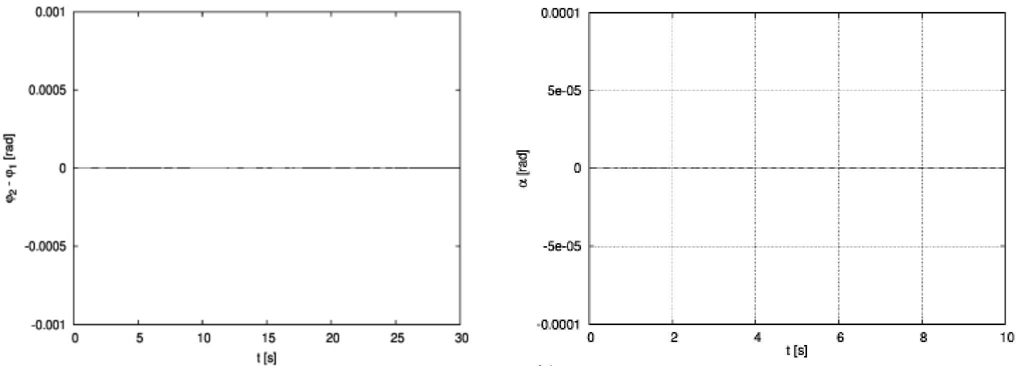
a) b)

Fig. 5. Time-history of disphasing angle of vibrators $\Delta\varphi$ (a) and angular coordinate α (b) for suspension parameters presented in Table 1 – 1



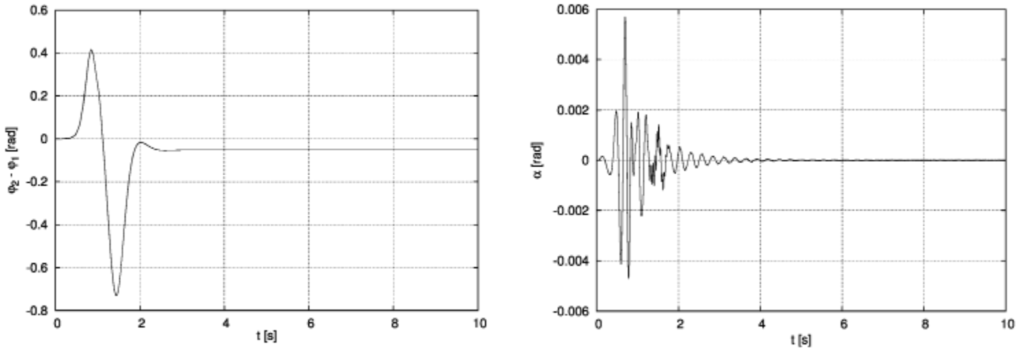
a) b)

Fig. 6. Time-history of disphasing angle of vibrators $\Delta\varphi$ (a) and angular coordinate α (b) for suspension parameters presented in Table 1 – 2



a) b)

Fig. 7. Time-history of disphasing angle of vibrators $\Delta\varphi$ (a) and angular coordinate α (b) for suspension parameters presented in Table 1 – 3



a) b)

Fig. 8. Time-history of disphasing angle of vibrators $\Delta\varphi$ (a) and angular coordinate α (b) for suspension parameters presented in Table 1 – 4

7. Conclusions

Classic and new results concerning disturbances of self-synchronisations, achieved in the last years, among others by the authors, are presented in the paper. Especially the way of determining disphasing angles of vibrators for the cases of: asymmetry of driving and resistance moments, geometrical asymmetry of driving systems, flexible mounting of vibrators to the machine body and influence of collisions between the feed material and the machine body – are given.

In the original part of the work the influence of the elastic machine support arrangement and the elasticity ratio in vertical and horizontal directions of these elements on the drives self-synchronisation were investigated.

It was found, that **in order to obtain the accurate self-synchronisation** ($\Delta\varphi = 0$) of **vibrators the elastic elements should be symmetrically arranged versus the machine mass centre both in the vertical and horizontal direction ,maintaining the same values of elasticity and damping constants in both directions,** e.g. this way as in (Cieplok 2009).

Frequent, in practice, shifting of the supporting plane down or up – versus the mass centre – in a similar way as a diversification of elasticity and damping constants in a vertical and horizontal direction leads to a certain unconformity of phases of both vibrators and – in consequence – to angular oscillations of the machine body, which are disturbing the amplitude distribution along the working surface.

The phase incompatibility angle recorded in simulations, for the typical deviation values from the symmetric state, was of the order of $\Delta\varphi \approx 0.027 \div 0.052$ [rad], **which corresponds to values from** 1.55 **to** 2.98°**.**

Angular oscillations of the machine body – caused by disphasing – were in simulation investigations of the order of $(1.5 \div 3.7) \cdot 10^{-5}$ [rad], **which can be considered permissible for the typical machines.**

It should be noticed that, in some cases, machine body oscillations related to disphasing of vibrators can be significantly compensated by means of the proper dynamic scheme of the machine, e.g. by increasing its degree of freedom number (Czubak, 2013).

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