A R C H I V E S O F M E T A L L U R G Y A N D M A T E R I A L S
Volume 58 2013 Issue 4
DOI: 10.2478/amm-2013-0123

P. CZUBAK*

ANALYSIS OF THE NEW SOLUTION OF THE VIBRATORY CONVEYOR

ANALIZA NOWEGO ROZWIĄZANIA PRZENOŚNIKA WIBRACYJNEGO

It was noticed, in industrial practice, that in typical vibratory conveyors suspended on the coil springs system and driven by two self-synchronising vibrators their dissynchronisation can lead to local elevations of loose feed and in extreme cases to the arrestment of a feed flow from the machine or even to the reversal of its transportation direction along the body. The new structural solution of the vibratory conveyor which, in spite of disphasing of vibrators, enables the feed material transport along the trough with a constant velocity, was presented and analysed in the paper.

Keywords: vibratory conveyor, disphasing of vibrators, feed material transport

W praktyce przemysłowej zauważono, że w typowych przenośnika wibracyjnych zawieszonych na układzie sprężyn śrubowych i wymuszonych dwoma samosynchronizującymi się wibratorami ich rozfazowanie prowadzić może do miejscowego wypiętrzania nadawy sypkiej, a w skrajnych przypadkach, do zatrzymania spływu nadawy z maszyny lub miejscowego odwrócenia kierunku jej transportu wzdłuż korpusu. W pracy zaproponowano i analizowano nowe rozwiązanie konstrukcyjne przenośnika wibracyjnego, który mimo rozfazowania wibratorów umożliwia transport nadawy wzdłuż rynny z jednakową prędkością.

1. Introduction

Several essential treatment and transport processes are realised in industry by means of vibratory machines and devices such as: vibrating screens, vibratory conveyors, shake out grids for castings knocking out or vibrating tables for producing concrete prefabricates.

Vibratory conveyors are utilised in metallurgical industry for continuous transport – usually at short distances up to 20m – of hot materials (furnace slag, small steel elements etc.), caustic substances or substances emitting gases hazardous for the environment. In addition, vibratory conveyors enable cooling of feeds, recovery of heat from the transported materials (used later e.g. for warming furnace blowers), drying, humidifying etc. Furthermore they allow transporting in closed conduits.

The often applied kind of a vibratory feeder or conveyor of a linear trajectory of vibrations is - in industrial practice – the machine, which is schematically presented in Fig. 1. Its drive constitute two independent inertial vibrators set in motion by induction motors. Conveyors of this type are in the commercial offer of the majority of companies producing transport devices [1,2,3,4].

The proper running of the working process of this type of machines depends on obtaining synchronous, cophasal angular motion of unbalanced masses, being the source of the needed dynamic forces [5].

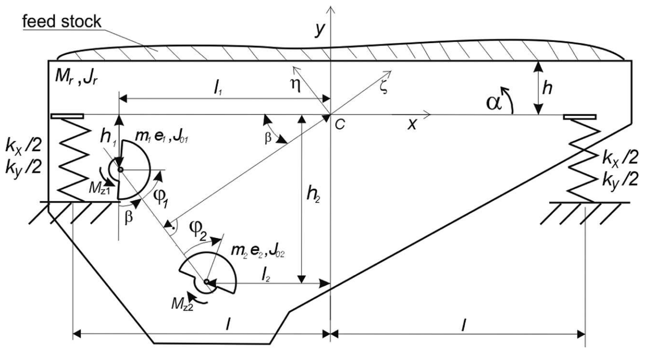

Fig. 1. Schematic presentation of the typical vibratory machine with two vibrators

The desired state is the situation when both vibrators are counter and cophasal running, generating the resultant force in the direction of working vibrations (s). This force direction should be passing via the machine mass centre, which ensures (at a symmetry of the elastic support system) the lack of excitations for angular oscillations of the machine body.

The conditions for a tendency for the desired synchronous, cophasal running of vibrators can be determined on the basis of the integral criterion formulated by I.I.Blechman [5,6].

$$D(\phi_1 - \phi_2, \phi_1 - \phi_3, ..., \phi_1 - \phi_n) = \frac{1}{T}\left[\int_0^T (E - V)dt - \int_0^T (E_w - V_w)dt\right] = \min \quad (1)$$

* AGH UNIVERSITY OF SCIENCE AND TECHNOLOGY, FACULTY OF MECHANICAL ENGINEERING AND ROBOTICS, AL. A. MICKIEWICZA 30, 30-059 KRAKÓW, POLAND

According to this criterion the phase angles system is stable around values $\Delta\phi_{12}, \Delta\phi_{13}, ..., \Delta\phi_{1n}$, if for these values function D, determined by equation (1), attains the local minimum, where:

$\phi_1, \phi_2, ..., \phi_n-$ angles of rotation of individual vibrators versus their initial positions,

$T = \frac{2\pi}{\omega}$ – forced vibrations period,

E – kinetic energy of the machine body with rotors mass concentrated in the pivoting point,

V – potential energy of the machine body supporting system,

E_w, V_w – kinetic and potential energy of constrains between vibrators – respectively.

For machines working in the far-over-resonance mode, for which an influence of elasticity forces on the suspension system can be neglected (in accordance with the scheme presented in Fig. 1), the above given condition leads to (5) the equation:

$$D>0 \tag{2}$$

However, this criterion does not decide the occurrence and accuracy of the synchronisation in case when there are counter acting factors. The range of the allowable disphasing angles of vibrators for various types of vibratory machines, given in paper [7], indicates the meaning of the dissynchronisation effect for the working process:

$\Delta\varphi \leqslant$ 3-5^o for vibrating screens,

$\Delta\varphi \leqslant$ 5-12^o for feeders,

$\Delta\varphi \leqslant$ 12-16^o for vibratory conveyors and self-feeding shaking grids.

It was noticed in the industrial practice, that disphasing of vibrators can lead to local elevations of loose feed, and in extreme cases to the arrestment of a feed flow from the machine or even to the reversal of its transportation direction. This problem was widely discussed in papers Michalczyk, Czubak [8,9].

2. Reasons of an irregular transportation velocity of feed materials

Several reasons of this type of disturbances are listed below [8,9,10,11,12,13,14].

1. Asymmetry of vibrators motion, caused by different direction of the vibrators force of gravity in relation to unbalanced masses.
2. Collisions of the machine body with the feed, which are changing vibrators running and, in effect, leading to rotational vibrations of the body.
3. Asymmetry of driving moments and resistance of bearings of vibrators caused by producing, assembling or exploitation factors.
4. Structure or assembling errors causing that the direction of operation of the resultant force of the inertial two-mass drive is not passing through the mass centre of the machine.
5. Coupling between the translatory motion along horizontal and vertical axis and the angular body motion caused by not proper system of the elastic support of the machine body.
6. Overlapping of transverse vibrations and the working ones – at whippy bodies.

Except for the case of the body transverse vibrations, the effect of irregular distribution of vibrations along the machine working surface is related to an occurrence of the body angular motion (oscillations). This motion causes the generation of additional components of velocity and displacements resulting from rotations, which disturbs and diversifies vibrations in individual areas of the working surface.

In the case of point 4 the effects of irregular distribution of vibrations along the machine working surface can be eliminated by removal of constructional errors and the accurate assembling of the conveyor. In the case of point 5 – by changing parameters of the machine body suspension, while in case of point 6 – by changing the body structure for more stiff one.

A diversification of driving-anti-torque moments of both vibrators, an influence of gravitation as well as influence of collisions of feed materials with the machine body cause their deviation from the synchronous running forced by vibratory synchronising moments. The synchronizing moments occur, at meeting certain conditions [5], self-acting in a system in which vibrators phase angles are not identical. The asymmetry of vibrators positioning causes a deviation of resultant forces from the direction passing through the system mass centre and the excitation of the machine body rotational vibrations, which – in turn – are responsible for the irregular distribution of vibrations on the body surface and, in effect, for the diversified transport velocities.

The main purpose of the invention is designing of the conveyor, which – in spite of the vibrators deviation from the synchronous running – will have a constant transport velocity along the whole length of the trough.

3. Structure of the vibratory conveyor – according to the invention

The conveyor, feeder, according to the invention [15], in a similar fashion as the classical solution, has the trough open on both ends, elastically supported on the stiff foundation and excited for vibrations by two counter-running inertial vibrators (3) providing the resultant force in the direction s of the main mass (1) vibrations. The vibrators are mounted to the main mass on the additional system of leaf springs (5) allowing for the vibrators motion in the direction n, perpendicular to the excitation forces direction τ. If the vibrators running is not symmetrical their resultant force is not acting accurately in the direction of the conveyor trough vibrations τ. This force can be geometrically divided into two directions: working τ and perpendicular n.

The principle of operation of the additional suspension of vibrators is based on the fact that a part of the force acting in the working direction acts directly on the main mass (axial direction of leaf springs (5)), while a part of the force acting in direction n acts on the main mass via leaf springs (5) decreasing its acting on the rotation of the main mass (1) thus, decreasing difference in the feed material transport (2) along the trough (1). Suspending vibrators on the additional degree of freedom eliminates the influence of the vibrators motion asymmetry caused by different direction of the vibrators force of gravity versus unbalanced masses, collisions of the machine body with feed materials and – the most important – the asymmetry of driving moments and the vibrators resistance of bearings on the distribution of the transport velocity along the trough length.

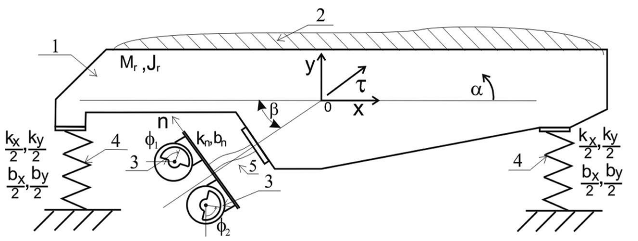

Fig. 2. Conveyor, according to the patent application No P.399532

This type of solution was not applied, up to the present, since the additional degree of freedom is not the result of the necessity of enabling the self-synchronisation of vibrators. The influence of vibrators asymmetry of motion on the transporting velocity distribution along the trough, in spite of being noticed in industrial practice some time ago, was only recently thoroughly investigated and described in papers [8,9].

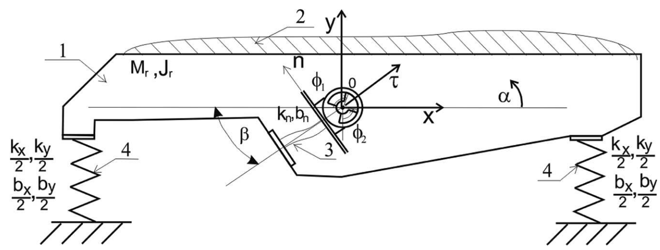

Fig. 3. Conveyor with vibrators situated in the centre of gravity of the main mass

It should be mentioned, that due to the additional degree of freedom vibrators can be suspended in such a way that their centre of gravity will coincide with the centre of gravity of the main mass (1) (Fig. 3), which is not possible in the classical conveyor on account of the lack of the self-synchronisation of vibrators. In the further part of this work it will be shown how advantageous is this solution.

4. Simulation investigations

The simulation model, corresponding to the scheme shown in Fig. 2, was designed for assessing the influence of the feed material – collisions with the trough, additional damping in the system [16,17,18,19,20], as well as the influence of a diversification of resistance of vibrators bearings on irregularity of the transportation velocity along the trough in the new vibratory conveyor.

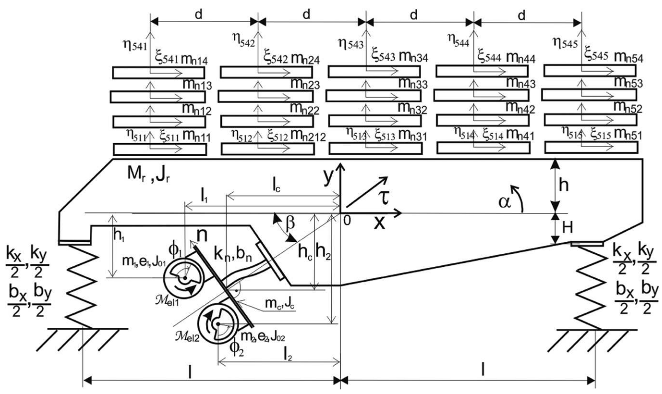

Fig. 4. Model of the conveyor together with the feed material

The model consists of: two inertial vibrators of independent induction drives described by static characteristics, suspended on the body by means of the leaf springs system allowing their motion along the dependent coordinate n, the machine body performing plane motion and supported on the vertical helical springs system, five four-layers models of the loose feed material [16,21] placed in various points of the working surface of the machine. The influence of the force of gravity on the vibrators angular motion was also taken into account in this model.

The mathematical model of such system consists of the matrix equation (3) describing the machine motion, equations (7) and (8) describing normal and tangent interactions between feed layers and between feed and the machine body. equations (9) serving for the determination motions of successive feed layers and equations (10) describing electromagnetic moments of driving motors.

$$M \cdot \ddot{q} = Q \tag{3}$$

where:

$$M = \begin{bmatrix} M_r + m_1 + m_2 + m_c & 0 & m_1h_1 + m_2h_2 + m_ch_c & m_1e_1\sin\phi_1 & m_2e_2\cos\phi_2 & -(m_1+m_2+m_c)\sin\beta \\ 0 & M_r + m_1 + m_2 + m_c & -m_1a_1 - m_2a_2 - m_ch_c & m_1e_1\cos(\phi_1) & m_2e_2\sin(\phi_2) & (m_1+m_2+m_c)\cos\beta \\ m_1h_1 + m_2h_2 + m_ch_c & -m_1a_1 - m_2a_2 - m_ch_c & \begin{matrix} m_2h_2^2 + m_2a_2^2 + m_1h_1^2 + J_c \\ m_1a_1^2 + m_ca_c^2 + m_ch_c^2 + J_r \end{matrix} & \begin{matrix} m_1H_1e_1\sin(\phi_1)- \\ m_1a_1e_1\cos(\phi_1) \end{matrix} & \begin{matrix} m_2H_2e_2\cos(\phi_2)- \\ m_2a_2e_2\sin(\phi_2) \end{matrix} & \begin{matrix} -(m_1h_1+m_2h_2+m_ch_c)\sin\beta \\ -(m_1a_1+m_2a_2+m_ca)_c\cos\beta \end{matrix} \\ m_1e_1\sin\phi_1 & m_1e_1\cos(\phi_1) & \begin{matrix} m_1H_1e_1\sin(\phi_1)- \\ m_1a_1e_1\cos(\phi_1) \end{matrix} & m_1e_1^2 + J_{01} & 0 & \begin{matrix} -m_1e_1\sin\phi_1\sin\beta \\ +m_1e_1\cos\phi_1\cos\beta \end{matrix} \\ m_2e_2\cos\phi_2 & m_2e_2\sin(\phi_2) & \begin{matrix} m_2H_2e_2\cos(\phi_2)- \\ m_2a_2e_2\sin(\phi_2) \end{matrix} & 0 & m_2e_2^2 + J_{02} & \begin{matrix} -m_2e_2\cos\phi_2\sin\beta \\ +m_2e_2\sin\phi_2\cos\beta \end{matrix} \\ -(m_1+m_2+m_c)\sin\beta & (m_1+m_2+m_c)\cos\beta & \begin{matrix} -(m_1h_1+m_2h_2+m_ch_c)\sin\beta \\ -(m_1a_1+m_2a_2+m_ca)_c\cos\beta \end{matrix} & \begin{matrix} -m_1e_1\sin\phi_1\sin\beta \\ +m_1e_1\cos\phi_1\cos\beta \end{matrix} & \begin{matrix} -m_2e_2\cos\phi_2\sin\beta \\ +m_2e_2\sin\phi_2\cos\beta \end{matrix} & m_1 + m_2 + m_c \end{bmatrix}$$

(4)

$$\ddot{q} = [\ddot{x}\ \ddot{y}\ \ddot{\alpha}\ \ddot{\phi}_1\ \ddot{\phi}_2\ \ \ddot{n}\ \ ]^T \tag{5}$$

$$Q = \begin{bmatrix} m_2e_2\dot{\phi}_2^2\sin(\phi_2) - m_1e_1\dot{\phi}_1^2\cos(\phi_1) - 2k_x(x + H\alpha) - 2b_x(\dot{x} + H\dot{\alpha}) - T_{101} - T_{102} - T_{103} - T_{104} - T_{105} \\ -m_2e_2\dot{\phi}_2^2\cos(\phi_2) + m_1e_1\dot{\phi}_1^2\sin(\phi_1) - k_y(y + l_1\alpha) - k_y(y - l_2\alpha) - b_y(\dot{y} + l_1\dot{\alpha}) - b_y(\dot{y} - l_2\dot{\alpha}) - F_{101} - F_{102} - F_{103} - F_{104} - F_{105} \\ -m_1h_1e_1\dot{\phi}_1^2\cos(\phi_1) - m_1a_1e_1\dot{\phi}_1^2\sin(\phi_1) + m_2h_2e_2\dot{\phi}_2^2\sin(\phi_2) + m_2a_2e_2\dot{\phi}_2^2\cos(\phi_2)) - 2k_xh^2\alpha - 2k_xhx - 2b_xh\dot{x} - 2b_xh^2\dot{\alpha} - \\ -k_y(y + l_1\alpha)l_1 + k_y(y - l_2\alpha)l_2 - b_y(\dot{y} + l_1\dot{\alpha})l_1 + b_y(\dot{y} - l_2\dot{\alpha})l_2 + (T_{101} + T_{102} + T_{103} + T_{104} + T_{105})h + F_{101}2d + F_{102}d - F_{104}d - F_{105}2d \\ M_{el1} - b_{s1}\dot{\phi}_1^2 sign(\dot{\phi}_1) - m_1ge_1\cos(\phi_1) \\ M_{el2} - b_{s2}\dot{\phi}_2^2 sign(\dot{\phi}_2) - m_2ge_2\cos(\phi_2) \\ \\ m_1e_1\dot{\phi}_1^2\sin(\beta)\cos(\phi_1) + m_1e_1\phi_1^2\cos(\beta)\sin(\phi_1) + m_2e_2\dot{\phi}_2^2\sin(\beta)\sin(\phi_2) + m_2e_2\dot{\phi}_2^2\cos(\beta)\cos(\phi_2)) - k_nn - b_n\dot{n} \end{bmatrix}$$

where:

$F_{j,j-1,k}$ – normal component of the j^{th} layer pressure on the j^{th}-1 in the k^{th} column,

$T_{j,j-j,k}$ – tangent component of the j^{th} layer pressure on the j^{th}-1 in the k^{th} column,

j – index of the feed layer, j=0 concerns the machine body,

k – index of the feed layer column.

When successive feed layers (in the given column) *j* and *j-1* are not in contact, then the contact force in the normal direction $F_{j,j-1,k}$ and tangent $T_{j,j-1,k}$ between these layers equals zero:

$$F_{j,j-1,k} = 0, \quad T_{j,j-1,k} = 0 \text{ for } \eta_{j,k} \geqslant \eta_{j-1,k} \tag{6}$$

Otherwise, the contact force occurs in the normal direction between feed layers j,k and *j-1,k* (or in case of the first layer between the layer and trough), which model [16] is of a form:

$$F_{j,j-1.k} = (\eta_{j-1,k} - \eta_{j,k})^p \cdot k \cdot \\ \cdot \left\{1 - \frac{1-R^2}{2}\left[1 - sgn(\eta_{j-1,k} - \eta_{j,k}) \cdot sgn(\dot{\eta}_{j-1,k} - \dot{\eta}_{j,k})\right]\right\} \tag{7}$$

and the force, originated from friction, in the tangent direction:

$$T_{j,j-1,k} = -\mu F_{j,j-1,k} sgn(\dot{\xi}_{j,k} - \dot{\xi}_{j-1,k}) \tag{8}$$

where: R – coefficient of restitution of normal impulses at collision.

Equations of motion in directions ξ and η of individual feed layers, with taking into account the conveyor influence on lower feed layers, are of a form:

$$\begin{aligned} m_{nj,k}\ddot{\xi} &= T_{j,j-1,k} - T_{j+1,j,k}, \\ m_{nj,k}\ddot{\eta} &= -m_{nj,k}g + F_{j,j-1,k} - F_{j+1,j,k} \end{aligned} \tag{9}$$

M_{eli}- electromagnetic moment developed by the i^{th} motor, assumed in the form corresponding the static characteristic of this motor [22]:

$$M_{eli} = \frac{2M_{ut}(\omega_{ss} - \dot{\phi}_{i1}) \cdot (\omega_{ss} - \omega_{ut})}{(\omega_{ss} - \omega_{ut})^2 + (\omega_{ss} - \dot{\phi}_i)^2} \quad i = 1,2 \tag{10}$$

where:

M_{ut} – breakdown torque of driving motors

ω_{ss} – synchronous frequency of driving motors

ω_{ut} – breakdown torque frequency of driving motors

The simulation was performed for the following values of parameters:

l = 1[m]

l_1 = 1[m]

l_2 = 0.5[m]

l_c = 0.6[m]

h = 0[m]

h_1 = 0.5[m]

h_2 = 1[m]

h_c = 0.6[m]

$b_x = b_y$ = 400 [Ns/m]

b_n = 100 [Ns/m]

$k_x = k_y$ = 150000[N/m]

k_n = 25000 [N/m]

$m_1 = m_2$ =5[kg]

m_c = 10[kg]

M_r = 120[kg]

$J_{01} = J_{02}$ = 0.0021[kgm^2]

J_r = 25[kgm^2]

J_c = 1[kgm^2]

$e_1 = e_2$ = 0.02[m]

M_{ut} = 50[Nm]

ω_{ss} = 50π[rad/s]

ω_{ut} = 15.9*2π[rad/s]

b_{si} – coefficient of resistance to motion of vibrators

b_{s1} = variable [Ns^2/m]

b_{s2} = 0.00009 [Ns^2/m]

The obtained simulation results for the new vibratory conveyor (Fig. 2) were compared with the results for the classical conveyor (such as shown in Fig. 1). Equations of motion of the classical conveyor or feeder were discussed in papers [8,9]. In practice, if it is assumed that $k_n \to \infty$ in equations for the conveyor presented in Figure 2 the results corresponding to the classical conveyor are obtained (Fig. 1).

5. Results of simulation investigations

5.1. Influence of the machine body collisions with the feed material on the angular vibrations amplitude

In order to verify the operation of the new conveyor, its amplitudes of angular oscillations of the machine body A_α resulting from disphasing of vibrators – and caused by collisions with the feed – were compared with amplitude values for the classical conveyor of the same parameters. It should be emphasised that – in this case – the machine body oscillations depend on the coefficient of throw. Within the range $1 \leqslant k_p \leqslant 3.3$ in theoretical considerations (performed by prof. Michalczyk) two minima of the dependence of body oscillations on the coefficient of throw were found, while the numerical simulations indicate only one minimum at $k_p \approx 2.7$. This value was recommended [8] to avoid body oscillations causing an irregular distribution of vibration amplitudes along the machine body.

Diagrams in Figure 5 present the comparison of the body oscillations angle on the coefficient of throw for the classical conveyor (theoretical curve and determined by simulation [8]) with the body oscillations angle for the new, proposed conveyor of the same parameters.

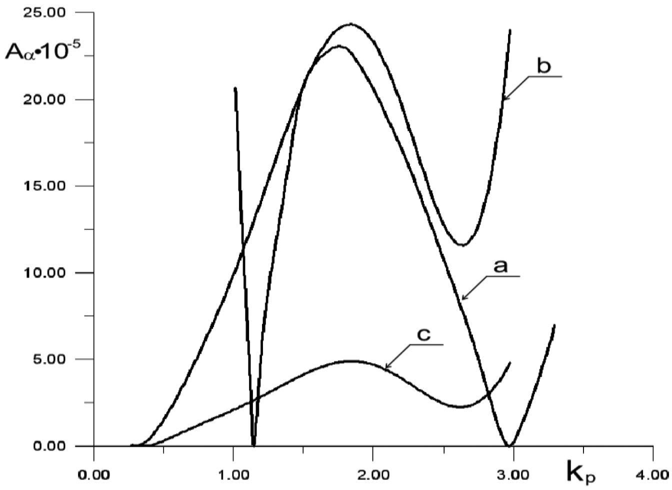

Fig. 5. Dependence of the amplitude of the body oscillations A_α on the coefficient of throw k_p, for loose feed of a mass 60 kg: a) Theoretical curve, b) Diagram obtained by the numerical simulation of the classical conveyor, c) Diagram obtained by the numerical simulation of the new conveyor

The oscillations angle of the body caused by dissynchronisation of vibrators under the influence of collisions between the body and feed in the new conveyor (for the applied parameters) constitutes 20% of values obtained for the classical conveyor. The minimum occurs at the same value of the coefficient of throw, $k_p = 2.7$.

5.2. Influence of the vibrators resistances of bearing on the amplitude of angular vibrations and transport velocities

Significantly higher influence on the conveyor operation has the difference in the resistances of bearings of the vibrators causing their disphasing, which – in turn – causes considerable body oscillations and related to them differences in the transport velocity along the trough length. In this case the operation of the new proposed conveyor was also compared with the classical one of the same parameters. During the simulation the influence of the body collisions with the feed as well as the influence of the force of gravity on dissynchronising of vibrators was taken into account.

By means of the numerical simulation the machine body oscillation (Fig. 6) as well as the time history of the transport along the trough length for individual columns of the feed (Fig. 7) were determined for the classical conveyor (as the one shown in Fig. 1) at the resistance on one vibrator determined for the efficient bearing, while on the second bearing being twice higher: $b_{s1} = 2 \cdot b_{s2}$ [9].

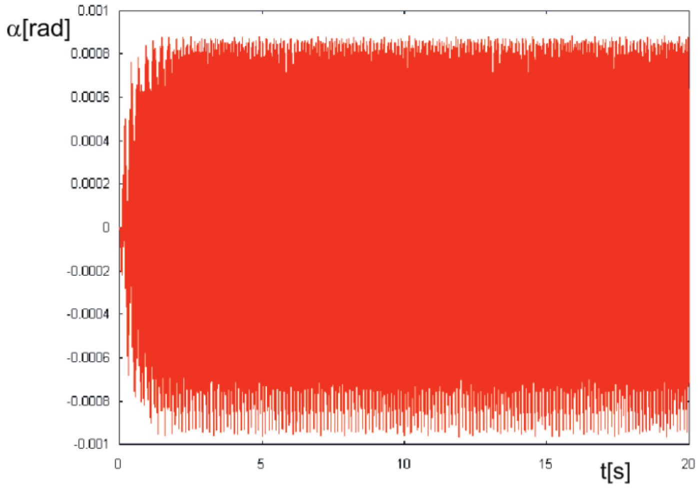

Fig. 6. Machine body oscillations – of the classical conveyor, $b_{s1} = 2 \cdot b_{s2}$

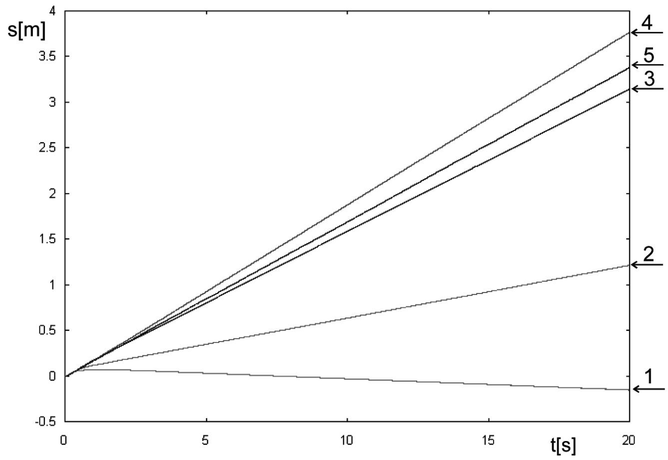

Fig. 7. Transport velocity of individual columns of the feed – on the classical conveyor, $b_{s1} = 2 \cdot b_{s2}$

Fig. 7 presents displacing versus time of the individual columns of the feed on the conveyor of the classical built. Successive columns are marked by numbers 1-5 (counted from the left side).

It is seen from these diagrams that a diversification of the resistance to motion of vibrators (or, generally, driving-anti-torque moments) leads to the body oscillations, which can significantly diversify the transport along the machine working surface.

In case of large differences of the driving-anti-torque moments even the reversal of the transport direction can occur on one end of the body. It is seen in the diagram that the left column, representing a multi-layer model of the loose feed, is moving in the opposite direction than other columns.

Figure 8 presents the time history of the body oscillations, while Figure 9 the transport velocity for the new conveyor of the same parameters, at $b_{s1} = 2 \cdot b_{s2}$. At the new structure of the vibrators suspension the 4-times decrease of the body oscillations in relation to the classical conveyor was obtained, and to this end the velocity differences are significantly lower and the most often acceptable in industry.

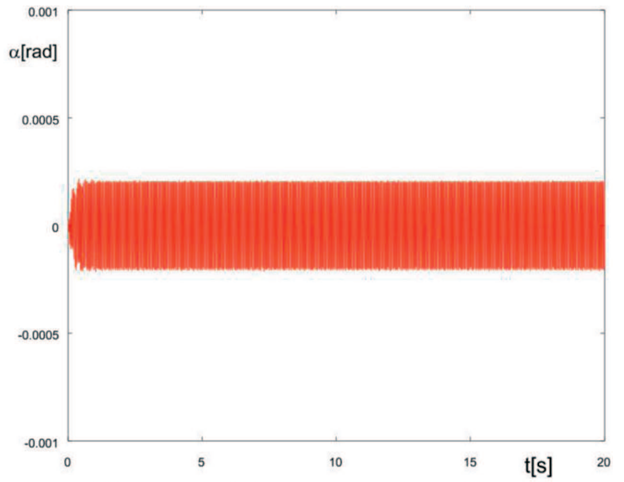

Fig. 8. Machine body oscillations – of the new solution of the vibratory conveyor, $b_{s1} = 2 \cdot b_{s2}$

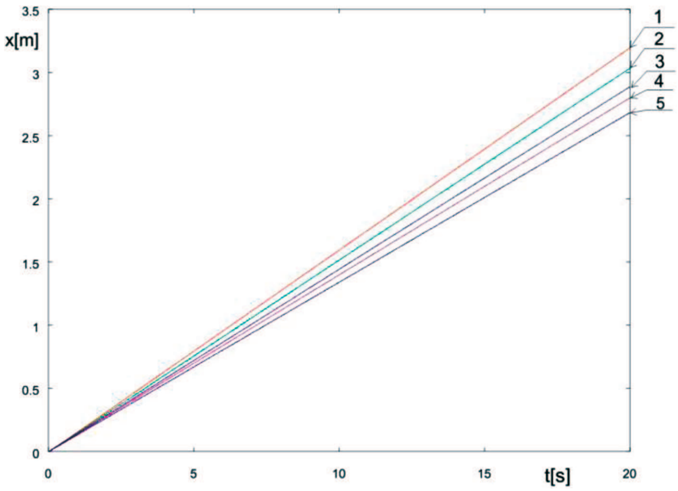

Fig. 9. Transport velocity of successive columns of the feed – for the new solution of the vibratory conveyor, $b_{s1} = 2 \cdot b_{s2}$

A time-history of the transport velocity for the new conveyor, in which vibrators are situated in the main mass centre of gravity (as seen in Fig. 3) is presented in Figure 10. Such placement of vibrators is possible – as it was mentioned earlier – due to the additional degree of freedom allowing the self-synchronisation. In this case the body oscillations decay and the transport velocity is equal for individual columns in spite of strong disphasing of vibrators mainly because of: $b_{s1} = 2 \cdot b_{s2}$. Equations of motion of this system are similar to equations of the system shown in Fig. 2, while distances of vibrators and their mounting from the centre of gravity: $a_1, a_2, a_c, h_1, h_2, h_3$, are equal 0.

Disphasing of vibrators causes the change of the operation direction of the resultant force causing only the change of the trough vibrations direction. The angle calculated from the level of rectilinear trough vibrations influences the feed transport velocity [17,23], however this velocity will be constant along the trough length. Neither collisions between the body and feed nor the gravitation had any influence on the velocity diversification of successive feed columns – at this new structure conveyor.

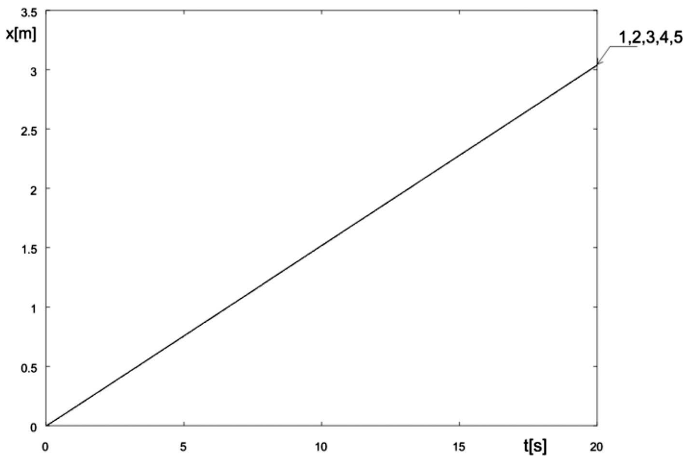

Fig. 10. Time-history of the transport velocity of the successive columns of the feed – for the new solution of the vibratory conveyor – when vibrators are situated in the main mass centre of gravity, $b_{s1} = 2 \cdot b_{s2}$

6. Conclusions

1. Five-times decrease of the angle of the machine body oscillations – related to vibrators disphasing caused by collisions between the body and feed – was obtained for the typical structure parameters of the new vibratory conveyor as compared with the classical structure.
2. When the new vibratory conveyor was applied, the four-times decrease of the oscillation angle of the machine body – related to vibrators disphasing caused by the differences of driving-anti-torque moments – was obtained as compared with the classical structure.
3. The proposed structure of the conveyor causes a significant decrease of the transport velocity differences along the trough – related to disphasing of vibrators – as compared with the classical structure.
4. When the conveyor with vibrators situated in the main mass centre of gravity is applied, the constant transport velocity along the trough length is obtained. It happens regardless of the feed material mass, coefficient of throw, gravitation influence on vibrators as well as diversification of the vibrators (or, in general, driving-anti-torque moments) resistance to motion.

REFERENCES

[1] www.vibra-schultheis.com
[2] www.millpower.com
[3] www.generalkinematics.com
[4] www.jeffreyrader.com
[5] I. B l e c k h m a n, Vibration Mechanic, Moscow, Fizmatlit, 1994.
[6] I. B l e c k h m a n, Yaroshevich, Extension of the Domain of Applicability of the Integral Stability Criterion in Synchronization Problems. Journal of Applied Mathematics and Mechanics **68**, 839-846 (2004).

[7] E. Ł a v e n d e l (red), Wibracji w Technikie Vol. 4. Maszynostrojenije, Moskwa 1981.

[8] J. M i c h a l c z y k, P. C z u b a k, Influence of Collisions with a Feed Material on Cophasal Mutual Synchronisation of Driving Vibrators of Vibratory Machines. Journal of Theoretical and Applied Mechanics **48**, 1 (2010).

[9] J. M i c h a l c z y k, P. C z u b a k, Influence of the asymmetry of vibrators resistance to motion on the correctness of the vibration distribution on working surfaces of vibratory machines. Archives of Metallurgy and Materials **55**, 1 (2010).

[10] J. M i c h a l c z y k, Inaccuracy in self-synchronisation of vibrators of two-drive vibratory machines caused by insufficient stiffness of vibrators mounting. Archives of Metallurgy and Materials **57**, 3 (2012).

[11] S. L i, L. X u, Y. W a n g, The Research of COMAS Vibrating Conveyor Noise Reduction. Applied Mechanics and Materials **184**, 583-586 (2012).

[12] J. M i c h a l c z y k, P. C z u b a k, Methods of Determination of Maximum Amplitudes in the Transient Resonance of Vibratory Machines. Archives of Metallurgy and Materials **55**, 3 (2010).

[13] G. T h e l e n, Probleme beider Bestimmung des Schwingungsverhaltens von uberkutisch abgestimmten Schwingforderern mit Unwuchterregung. Deutsche Hebe und Forder Technik **12**, 1970.

[14] J. M i c h a l c z y k, P. C z u b a k, Natural vibrations of long vibratory conveyers. Archives of Metallurgy and Materials **51**, 1 (2006).

[15] P. C z u b a k, Przenośnik lub podajnik wibracyjny o stałej prędkości transportu. Zgłoszenie patentowe, P.399532, 2012.

[16] J. M i c h a l c z y k, Phenomenon of Force Impulse Restitution in Collision Modelling. Journal of Theoretical and Applied Mechanics **46**, 4 (2008).

[17] T. B a n a s z e w s k i, Przesiewacze, Śląsk 1990.

[18] M. R o u i j a a, C. K r u e l l e, I. R e h b e r g, R. G r o c h o w s k i, P. W a l z e l, Transport and Pattern Formation in Granular Materials on a Vibratory Conveyor. Chem. Eng. Technology **28**, 41-44 (2005).

[19] R. G u i l l e r m o, S o t o - Y a r r r i t u, A. M a r t i n e z, Computer Simulation of Granular Material. Vibrating Feeders, Powder Handling & Processing **13**, 2 (2001).

[20] P. C z u b a k, Equalization of the Transport Velocity in a New Two-Way Vibratory Conveyor, Archives of Civil and Mechanical Engineering, Vol. XI, pp. 573-586, 2011.

[21] J. M i c h a l c z y k, G. C i e p l o k, Numerical Model of Vibrating Screen. Modelowanie Inżynierskie **1**, 32 (2006).

[22] M. G i e r g i e l, Dobór parametrów układów hamowania nadrezonansowych maszyn wibracyjnych. PhD, IMiR, Cracow, 1992.

[23] A. C z u b a k, Przenośniki wibracyjne. Śląsk, Katowice, 1964.

Received: 10 April 2012.