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FEED MATERIAL INFLUENCE ON THE DYNAMICS OF THE SUSPENDED SCREEN AT ITS STEADY STATE OPERATION AND TRANSIENT STATES

WPŁYW NADAWY NA DYNAMIKĘ PRZESIEWACZA PODWIESZONEGO W STANIE PRACY USTALONEJ I OKRESACH PRZEJŚCIOWYCH

The influence of the feed material on steady vibrations as well as on the transient resonance during the start-up and coasting of the suspended screen, was analysed in the hereby paper. The influence of the feed material presence on the possibility of performing the first half-turn of the vibrator was also established. The original, feed material simulation model reflecting its layer properties and diversification of feed particles vibrations along the screen riddle – specially developed for this aim – was applied in investigations.

Keywords: vibrating screens, limited power systems, loose feed material, transient resonance

W pracy poddano analizie wpływ nadawy na przebieg drgań ustalonych i rezonansu przejściowego podczas rozruchu i wybiegu przesiewacza podwieszonego. Ustalono też wpływ jaki obecność nadawy wywiera na możliwość wykonanie pierwszego półobrotu wibratora. Do badań zastosowano oryginalny, opracowany w tym celu model symulacyjny nadawy, odzwierciedlający jej własności warstwowe i zróżnicowanie drgań cząstek nadawy wzdłuż rzeszota przesiewacza

Słowa kluczowe: przesiewacze wibracyjne, układy o ograniczonym wzbudzeniu, nadawa sypka, rezonans przejściowy

1. Introduction

The proper technological process of a screen requires maintaining the correct value of the coefficient of throw, it means the amplitude and vibration frequency as well as, usually, a translatory character of the machine body motion (Banaszewski, 1990). The last condition is attributed both to relations in the system: feed – machine body and to the proper self-synchronisation of vibrators (Michalczyk & Cieplok, 2014).

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Majority of these factors are influenced by the carried feed (Švacha, 1971; Michalczyk, 1995; Michalczyk & Czubak, 2010). In addition there is a significant feed influence on transient processes, including the transient resonance during a start-up (Michalczyk & Czubak, 2010) or overcoming a circum resonant stall (Michalczyk & Cieplok & Bednarski, 2010). On account of calculation difficulties related to a variable structure of a system, analytical calculating methods currently applied in selecting work parameters of screens and other machines of a similar work character are mainly based on the analysis of an unloaded machine motion (Blechman, 1994; Goździecki & Świątkiewicz, 1975; Turkiewicz & Banaszewski, 1982; Banaszewski, 1990; Goncharevich & Frolov & Rivin, 1990). That is why simulation methods are important and the demand for simulation models representing the dynamics of the system: machine – feed, still growing.

Out of works undertaking the problem of a feed influence on the vibratory machine operation – with a special considering of machines for material classifying and short distance transport – the work of *V*. Plavinski should be mentioned (Plavinskij, 1969), which presents the experimental tests results concerning the feed influence on the machine vibrations amplitude in a steady state. The dependence of the equivalent amplitude (a motion is not exactly harmonic) of the riddle vibration was proposed in this study for the case of over-resonance machine:

$$A_{eq} = \frac{m_n e}{m_o + m_n + \lambda m} \tag{1}$$

where:

λ — according to Fig. 1,
m — feed mass,
m_o — riddle mass,
m_n, e — vibrator unbalanced mass and eccentric, respectively.

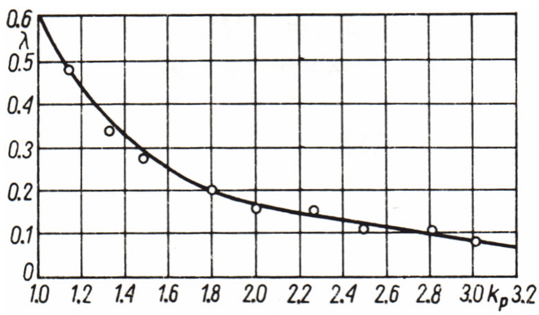

Fig. 1. Nomogram for the coefficient λ determination
k_p — coefficient of throw, determined as a ratio of the amplitude component, perpendicular to the machine work surface, of the acceleration in vibrating motion to the acceleration of gravity component, perpendicular to this surface. To determine preliminarily (without a feed taken into account) this coefficient value, $\lambda = 0$ should be assumed at calculating the amplitude from equation (1).

Other studies from this scope are e.g. investigations of A. Czubak (Czubak & Michalczyk, 2001) concerning experimental determination force interactions of a feed on a vibrating surface, works of T. Banaszewski (Banaszewski, 1990) presenting investigations of a similar character and range, but in relation to the sieve screen deck as well as theoretical (Michalczyk, 1995) and simulation studies (Czubak & Michalczyk, 2001). The last of the mentioned works contains the description of the feed influence both on the machine body motion and vibrator running and its excitation force waveform, but it concerns only the machine performing a perpendicular translatory motion. The newest studies are focused on problems of modelling loose feed materials by means of discrete elements (DEM), usually spherical or spherically derivative (Sinnott & Cleary, 2009; Krulle et al., 2007), however without taking into account the feed influence of the machine motion and drive system running.

2. Mathematical model of the system

The typical mining vibration screen, suspended type, presented schematically in Fig. 2, constitutes the subject of the analysis.

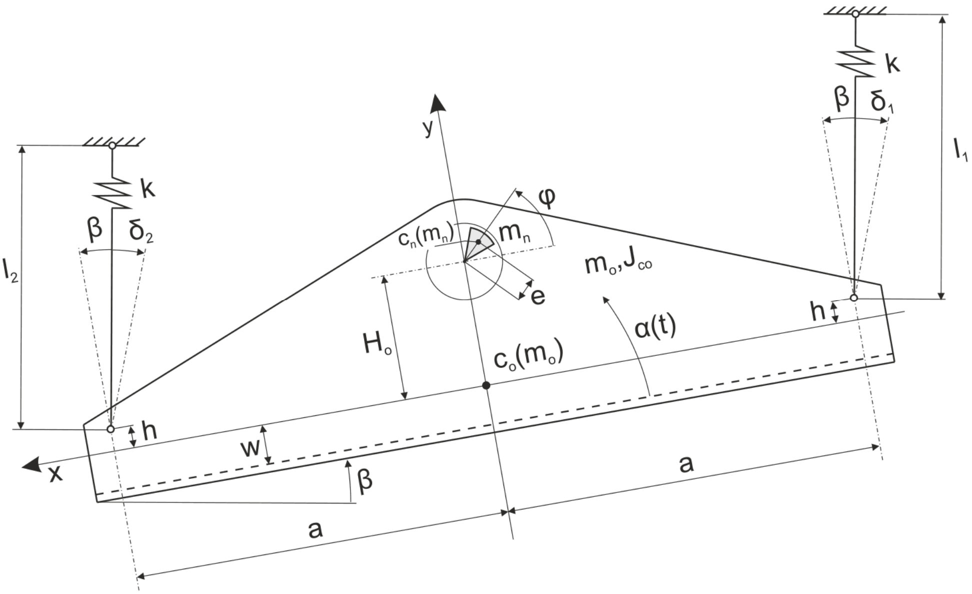

Fig. 2. Scheme of the suspended screen

where: m_o, J_{co} – mass and the central moment of inertia of the machine body with not rotational vibrator elements, c_o – mass center of m_o, m_n, e – vibrator unbalanced mass and its eccentric, c_n – mass centre of m_n, k – elastic constants of suspension system springs, β – riddle inclination angle, δ_i – out of plumb angles of ropes during vibrations. i = 1,2, x,y,α – absolute co-ordinate system assumed at the static balance point, used for determining coordinates $x(t)$, $y(t)$ of the mass centre c_o and the machine body angle of rotation $\alpha(t)$.
φ – vibrator angle of rotation (calculated in the absolute system, as in Figure 2), l_1, l_2 – total length of the suspension rope at the static balance point of the system, m – feed mass together with undersize particles (if they are carried by a riddle), remaining notations as in Figure 2.

Mathematical model of the machine without a feed was presented in paper (Michalczyk & Cieplok, 2015). This model was supplemented in the present paper, with dynamic equations describing the feed motion. The **feed model** was based on experimental tests (Czubak, 1964), in which it was shown that the loose feed was moving on the vibrating surface in a form of mutually displacing layers. The form of the feed model was based on the experimentally verified model presented in paper (Czubak & Michalczyk, 2001).

Dynamic equations of motion of the screen are written as:

$$m_o\ddot{x} = P_x + m_o g\sin(\beta) + S_{x1} + S_{x2} + \\ -\left(T_{00,11} + T_{00,12} + T_{00,13} + T_{00,13}\right) + Q_x \tag{2a}$$

$$m_o\ddot{y} = P_y - m_o g\cos(\beta) + S_{y1} + S_{y2} + \\ -(F_{00,11} + F_{00,12} + F_{00,13} + F_{00,14}) + Q_y \tag{2b}$$

$$J_{co}\ddot{\alpha} = P_x H_o + M_o + S_{x1}h + S_{x2}h + S_{y1}a - S_{y2}a + \\ +\left(T_{00,11} + T_{00,12} + T_{00,13} + T_{00,13}\right)\cdot w + \\ + F_{00,11}\cdot d_2 + F_{00,12}\cdot d_1 - F_{00,13}\cdot d_1 - F_{00,14}\cdot d_2 \tag{2c}$$

$$m_n[\ddot{x} + \ddot{\alpha}H_o + \ddot{\varphi}e\sin(\varphi) + \dot{\varphi}^2 e\cos(\varphi)] = -P_x + m_n g\sin(\beta) \tag{2d}$$

$$m_n[\ddot{y} + \ddot{\varphi}e\cos(\varphi) - \dot{\varphi}^2 e\sin(\varphi)] = -P_y - m_n g\cos(\beta) \tag{2e}$$

$$J_{cn}\ddot{\varphi} = M - M_o + P_x e\sin(\varphi) + P_y e\cos(\varphi) \tag{2f}$$

Where, apart from notations in Fig. 2, the following was assumed:

J_{cn} — central moment of inertia of unbalanced vibrator elements with the reduced moment of inertia of the engine rotor and power transmission elements (a drive placed on separately – was assumed), M_o — anti-torque moment in vibrator bearings;

$$M_o = c\dot{\varphi}^2\,\mathrm{sgn}(\dot{\varphi} - \dot{\alpha}) \tag{2g}$$

M — engine drive moment,
P_x, P_y — components of vibrator interaction forces acting on the machine body,
S_{xi}, S_{yi} — rope tension components S_i, $i = 1,2$,
$T_{00,ij}, F_{00,ij}$ — feed interaction forces acting on the sieve,
Q_x, Q_y — components of interaction forces of a feed material – fed and collected from a sieve.

Remaining notations as in Fig. 2.

Components of the rope tension were determined at the assumption that:

a)

$$l_1, l_2 > 30A_{H\max} \tag{3}$$

where: $A_{H\max}$ — maximal resonance amplitude of the rope fastening points to the machine body – in the horizontal direction.

b) Static deflection of springs is larger than the maximal resonance amplitude of the rope fastening points to the machine body – in the vertical direction $A_{V1,2\max}$:

$$\frac{mg}{2k} > A_{V1,2\max} \tag{4}$$

$$m = m_o + m_n \tag{5}$$

That time, tensions S_1, S_2 of ropes l_1, l_2 can be determined from the dependence:

$$S_{1lin} = k\left\{1 - \frac{\psi}{4}[1 - \mathrm{sgn}(\dot{f}_1)]\right\}\left(\frac{mg}{2k} + f_1\right),$$
$$S_1 = S_{1lin} \text{ if } S_{1lin} \geq 0,\ S_1 = 0 \text{ if } S_{1lin} < 0 \tag{6a}$$

$$S_{2lin} = k\left\{1 - \frac{\psi}{4}[1 - \mathrm{sgn}(\dot{f}_2)]\right\}\left(\frac{mg}{2k} + f_2\right),$$
$$S_2 = S_{2lin} \text{ if } S_{2lin} \geq 0,\ S_2 = 0 \text{ if } S_{2lin} < 0 \tag{6b}$$

where the description of elastic-damping forces of the hysteresis type was assumed acc. to (Michalczyk, 2008), while the suspension rope elongation from the static equilibrium position and their time derivatives were described by approximate dependencies:

$$f_1 \cong -(y + \alpha a)\cos(\beta) + (x + \alpha h)\sin(\beta) \tag{7a}$$

$$f_2 \cong -(y - \alpha a)\cos(\beta) + (x + \alpha h)\sin(\beta) \tag{7b}$$

$$\dot{f}_1 \cong -(\dot{y} + \dot{\alpha} a)\cos(\beta) + (\dot{x} + \dot{\alpha} h)\sin(\beta) \tag{8a}$$

$$\dot{f}_2 \cong -(\dot{y} - \dot{\alpha} a)\cos(\beta) + (\dot{x} + \dot{\alpha} h)\sin(\beta) \tag{8b}$$

Tension forces components are of the form:

$$S_{1x} = S_1[-\sin(\beta + \delta_1) \cong -S_1[\sin(\beta) + \delta_1 \cos(\beta)] \tag{9a}$$

$$S_{2x} = S_2[-\sin(\beta + \delta_2)] \cong -S_2[\sin(\beta) + \delta_2 \cos(\beta)] \tag{9b}$$

$$S_{1y} = S_1 \cos(\beta + \delta_1) \cong S_1[\cos(\beta) - \delta_1 \sin(\beta)] \tag{9c}$$

$$S_{2y} = S_2 \cos(\beta + \delta_2) \cong S_2[\cos(\beta) - \delta_2 \sin(\beta)] \tag{9d}$$

where deflection angles of ropes are determined by dependencies:

$$\delta_1 = [(x + \alpha h)\cos(\beta) + (y + \alpha a)\sin(\beta)]\frac{1}{l_1} \tag{10}$$

$$\delta_2 = [(x + \alpha h)\cos(\beta) + (y - \alpha a)\sin(\beta)]\frac{1}{l_2} \tag{11}$$

The feed was modelled in a layer-column form at the assumption of a continuity of layers along the riddle, Fig. 3:

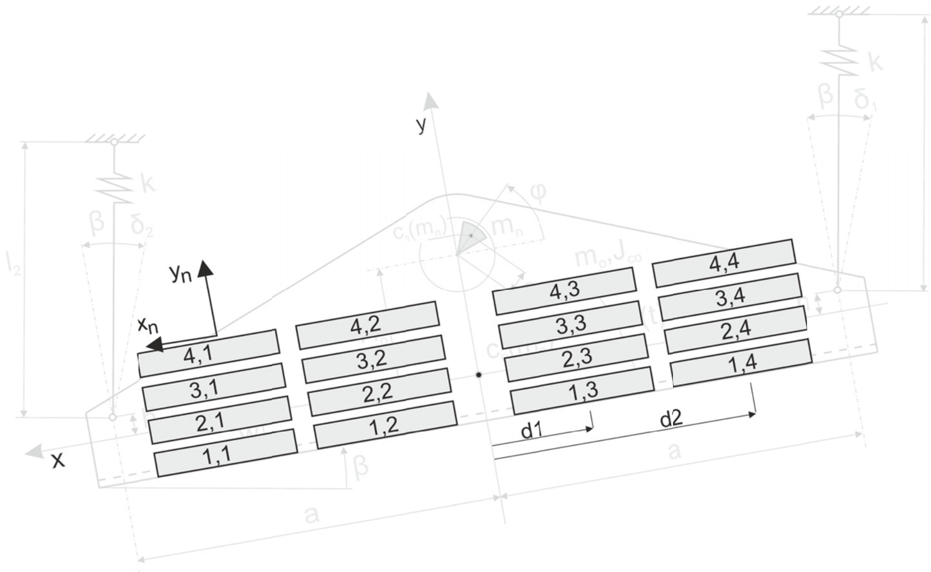

Fig. 3. Model of the feed

Contact forces $F_{ij,ij}$ of the impact type in y direction, where: i – row number, j – column number, occurring between the sieve of screen and the feed bottom layer were described by dependencies given by (Michalczyk, 2008). For the contact of the first layer columns (i = 1) with the sieve they have a form:

$$F_{00,11} = (y - d_2 \cdot \alpha - y_{11}) \cdot k_0 \cdot \left(1 - \frac{1-R^2}{2}[1 - \mathrm{sign}(\dot{y} - d_2 \cdot \dot{\alpha} - \dot{y}_{11})]\right) \tag{12a}$$

$$F_{00,12} = (y - d_1 \cdot \alpha - y_{12}) \cdot k_0 \cdot \left(1 - \frac{1-R^2}{2}[1 - \mathrm{sign}(\dot{y} - d_1 \cdot \dot{\alpha} - \dot{y}_{12})]\right) \tag{12b}$$

$$F_{00,13} = (y + d_1 \cdot \alpha - y_{13}) \cdot k_0 \cdot \left(1 - \frac{1-R^2}{2}[1 - \mathrm{sign}(\dot{y} + d_1 \cdot \dot{\alpha} - \dot{y}_{13})]\right) \tag{12c}$$

$$F_{00,14} = (y + d_2 \cdot \alpha - y_{14}) \cdot k_0 \cdot \left(1 - \frac{1-R^2}{2}[1 - \mathrm{sign}(\dot{y} + d_2 \cdot \dot{\alpha} - \dot{y}_{14})]\right) \tag{12d}$$

where: R — coefficient of restitution, $d_{1,2}$ — as in Fig. 2, while e.g. $F_{00,12}$ symbol denotes the force perpendicular to the riddle surface transferred by the sieve to the feed element placed in the first layer, second column.

When there is no contact these forces have to meet the zeroing conditions:

$$if\ y - d_2 \cdot \alpha \le y_{11} \text{ then } F_{00,11} = 0 \tag{13a}$$

$$\text{if } y - d_1 \cdot \alpha \le y_{12} \text{ then } F_{00,12} = 0 \tag{13b}$$

$$if\ y + d_1 \cdot \alpha \le y_{13} \text{ then } F_{00,13} = 0 \tag{13c}$$

$$\text{if } y + d_2 \cdot \alpha \le y_{14} \text{ then } F_{00,14} = 0 \tag{13d}$$

Similar dependencies describe forces in *y* direction between layers. The example of these forces notation for the first column elements, *j* = 1, is given below :

$$F_{11,21} = (y_{11} - y_{21}) \cdot k_1 \cdot \left(1 - \frac{1-R^2}{2}[1 - \text{sign}(\dot{y}_{11} - \dot{y}_{21})]\right)$$

$$\text{if } y_{11} \le y_{21} \text{ then } F_{11,21} = 0 \tag{14}$$

$$F_{21,31} = (y_{21} - y_{31}) \cdot k_1 \cdot \left(1 - \frac{1-R^2}{2}[1 - \text{sign}(\dot{y}_{21} - \dot{y}_{31})]\right)$$

$$\text{if } y_{21} \le y_{31} \text{ then } F_{21,31} = 0 \tag{15}$$

$$F_{31,41} = (y_{31} - y_{41}) \cdot k_1 \cdot \left(1 - \frac{1-R^2}{2}[1 - \text{sign}(\dot{y}_{31} - \dot{y}_{41})]\right)$$

$$\text{if } y_{31} \le y_{41} \text{ then } F_{31,41} = 0 \tag{16}$$

Tangent forces originated from the Coulomb friction can be written in a form shown (as an example) for the first column (*j* = 1):

$$T_{00,11} = -\mu \cdot F_{00,11} \cdot \text{sign}(\dot{x}_{1-4} - \dot{x} + w \cdot \dot{\alpha}) \tag{17a}$$

$$T_{11,21} = -\mu \cdot F_{11,21} \cdot \text{sign}(\dot{x}_{2-4} - \dot{x}_{1-4}) \tag{17b}$$

$$T_{21,31} = -\mu \cdot F_{21,31} \cdot \text{sign}(\dot{x}_{3-4} - \dot{x}_{2-4}) \tag{17c}$$

$$T_{31,41} = -\mu \cdot F_{31,41} \cdot \text{sign}(\dot{x}_{4-4} - \dot{x}_{3-4}) \tag{17d}$$

Equations of motion in *y* direction, for the first column elements (*j* = 1) are of a form:

$$m_{11} \cdot \ddot{y}_{11} = -m_{11} \cdot g \cdot \cos(\beta) + F_{00,11} - F_{11,21} \tag{18a}$$

$$m_{21} \cdot \ddot{y}_{21} = -m_{21} \cdot g \cdot \cos(\beta) + F_{11,21} - F_{21,31} \tag{18b}$$

$$m_{31} \cdot \ddot{y}_{31} = -m_{31} \cdot g \cdot \cos(\beta) + F_{21,31} - F_{31,41} \tag{18c}$$

$$m_{41} \cdot \ddot{y}_{41} = -m_{41} \cdot g \cdot \cos(\beta) + F_{31,41} \tag{18d}$$

Equations of motion in x direction for the layers (on the basis of the layers continuity condition) can be written in a form:

$$(m_{11}+m_{12}+m_{13}+m_{14})\cdot\ddot{x}_{1-4}=(m_{11}+m_{12}+m_{13}+m_{14})\cdot g\cdot\sin(\beta)+ \\ +(T_{00,11}+T_{00,12}+T_{00,13}+T_{00,13})-(T_{11,21}+T_{12,22}+T_{13,23}+T_{14,24}) \quad (19a)$$

$$(m_{21}+m_{22}+m_{23}+m_{24})\cdot\ddot{x}_{2-4}=(m_{21}+m_{22}+m_{23}+m_{24})\cdot g\cdot\sin(\beta)+ \\ +(T_{11,21}+T_{12,22}+T_{13,23}+T_{14,24})-(T_{21,31}+T_{22,32}+T_{23,33}+T_{24,44}) \quad (19b)$$

$$(m_{31}+m_{32}+m_{33}+m_{34})\cdot\ddot{x}_{3-4}=(m_{31}+m_{32}+m_{33}+m_{34})\cdot g\cdot\sin(\beta)+ \\ +(T_{21,31}+T_{22,32}+T_{23,33}+T_{24,34})-(T_{31,41}+T_{32,42}+T_{33,43}+T_{34,44}) \quad (19c)$$

$$(m_{41}+m_{42}+m_{43}+m_{44})\cdot\ddot{x}_{4-4}=(m_{41}+m_{42}+m_{43}+m_{44})\cdot g\cdot\sin(\beta)+ \\ +(T_{31,41}+T_{32,42}+T_{33,43}+T_{34,44}) \quad (19d)$$

Influence of the feed material on riddle motion, connected with its charging and dumping. Apart from – taken into account above – influences of layers being on the sieve, the influences related to a feed mass variability due to its feeding and loosing in a form of undersize and oversize particles should be imposed – in a general case – on the screen riddle.

In the steady state these forces are of a quasi-steady character and do not have any dynamic effect. They only insignificantly change the riddle balance point in a space, which allows their omission at creating the system model.

4. Simulation investigations

The following values of parameters were assumed:

The total mass of feed material was changed within a range: 0.1 to 0.4 of the screen riddle mass m_o, $R = 0$, k_0, $k_1 = 10000000$ N/m, $u = 0.4$

The screen parameters were assumed as in paper (Michalczyk & Cieplok, 2015).

$m_0 = 400$ kg, $m_n = 11$ or 20 kg, $e = 0.1$ m, $J_{c0} = 150$ kg*m^2, $H_0 = 0.1$ m, $h = 0.05$ m, $a = 0.8$ m, $k = 87600$ N/m, $l_1 = l_2 = 0.85$ m.

Simulation investigations allowed to determine the feed material influence on the screen motion: a) In transient states: Tables 1,2,3, Figures 4 and 5, b) In a steady state: Figures 6 to 15.

a) Transient states:

- Transient resonance:

TABLE 1

List of maximum values of individual coordinates in transient states, for the coefficient of throw equal 3.2

Feed mass/ m_o [%]	**x_{max} [m]**	**y_{max} [m]**	**$alfa_{max}$ [rad]**	
1	**2**	**3**	**4**	**5**
0% – start-up	0.0079449	0.010058	0.0034480	—
10% – start-up	0.0057850	0.007251	0.0041914	—
20% – start-up	0.0108500	0.006576	0.0076367	—

1	2	3	4	5
30% – start-up	0.0089382	0.006708	0.0109440	—
40% – start-up	0.0131410	0.007673	0.0083142	—
				Coasting time [s]
0% – coasting	0.0096527	0.023649	0.012854	32
10% – coasting	0.0083220	0.017899	0.012664	33.7
20% – coasting	0.0096602	0.014907	0.012797	36.6
30% – coasting	0.0118180	0.012804	0.012248	39.8
40% – coasting	0.0118410	0.010994	0.011696	44.5

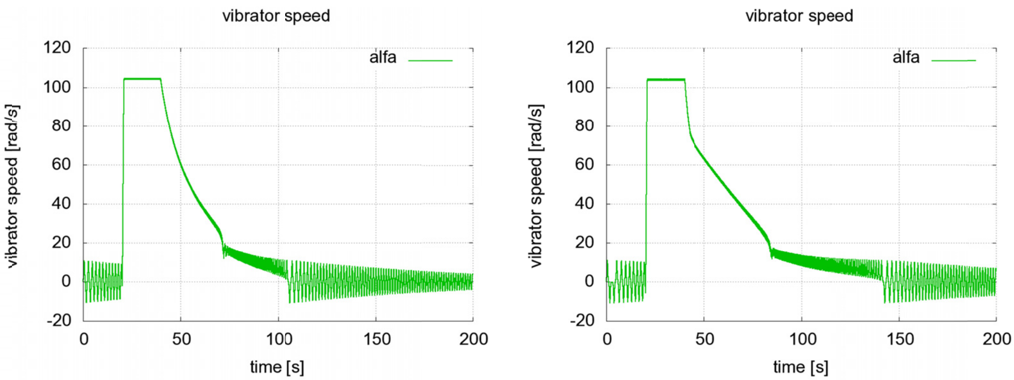

Fig. 4a-b. Diagrams present the vibrator angular speed for the system without and with the feed material, of a mass being 40% of the screen riddle mass, for the coefficient of throw being 3.2 – during the successive phases of motion: start-up, steady state, free coasting

TABLE 2

List of maximum values of coordinates in transient states for the coefficient of throw being 5.4

Feed mass/m_o [%]	x_{max}**[m]**	y_{max} **[m]**	***alfa***$_{max}$ **[rad]**	
0% – start-up	0.007885	0.017186	0.006145	—
10% – start-up	0.008895	0.009561	0.010509	—
20% – start-up	0.010177	0.010332	0.011636	—
30% – start-up	0.013026	0.013160	0.019095	—
40% – start-up	0.015569	0.015128	0.015739	—
				Coasting time [s]
0% – coasting	0.013605	0.032368	0.021164	26.5
10% – coasting	0.011539	0.020229	0.017705	23.9
20% – coasting	0.015614	0.016667	0.017959	26.8
30% – coasting	0.015956	0.015605	0.019374	34.5
40% – coasting	0.012909	0.014844	0.020723	55.5

- **Problem of the first half-turn**
  In some cases (especially at increased resistances to motion in the winter time) the vibrator start-up can be stopped due to a difficulty of the first lifting of an unbalanced

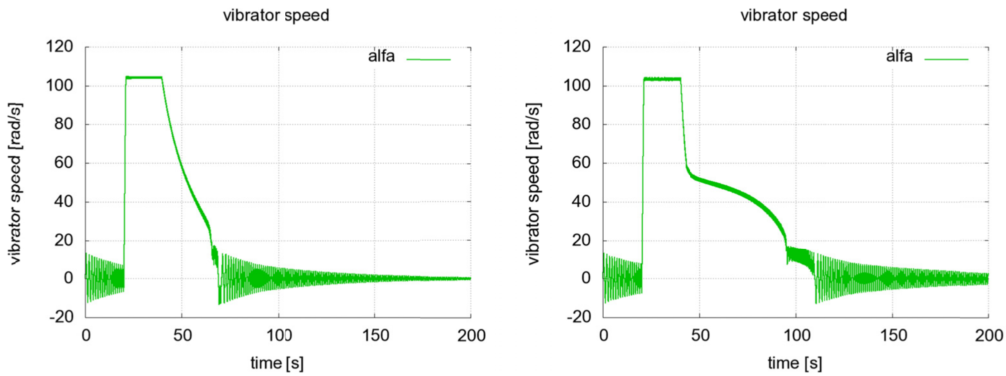

Fig. 5a-b. Diagrams present the vibrator angular speed for the system without and with the feed material, of a mass being 40% of the screen riddle mass m_o, for the coefficient of throw being 5.4 – during the successive phases of motion: start-up, steady state, free coasting

mass. Since this problem is discussed in references (e.g. Banaszewski.1990) without taking into account the machine body fluctuations and the feed material influence, the simulation method was applied in the hereby paper to asses the feed material influence on the possibility of performing the first half-turn. Table 3 shows the engine driving moment values required for performing the first half-turn at increased resistances to motion, in dependence of the ratio of the feed material mass to the machine riddle mass.

TABLE 3

Engine driving moment value needed for performing the first half-turn in dependence of the mass of the feed material

No.	Feed mass m_o [%]	Minimal driving moment [Nm], for unbalanced mass 11 kg, e = 0.1 m	Minimal driving moment [Nm], for unbalanced mass 20 kg, e = 0.1 m
1	0	8.02	14.52
2	10	8.02	14.53
3	20	8.02	14.53
4	30	8.02	14.53
5	40	8.02	14.53

As can be seen from the given data, the feed material presence does not influence significantly the easiness of performing the vibrator first half-turn.

b) Steady state

The steady state analysis of the screen operation was performed for two values of the coefficient of throw – 3.2 and 5.4, for various feed material amounts from a range: 0.001 (which represents the lack of the feed) to 0.4 of the riddle mass. Vibration waveforms of individual columns and layers in the perpendicular direction towards the sieve (y) for various coefficients of throw and feed masses, are presented in diagrams below. Values of amplitudes of the screen riddle along x and y axes as well as the average transport velocity of the feed material along the riddle are given below the diagrams.

Case 1 – The lack of the feed material (the feed mass constitutes 0.001 of the screen riddle mass m_o), coefficient of throw: 3.2.

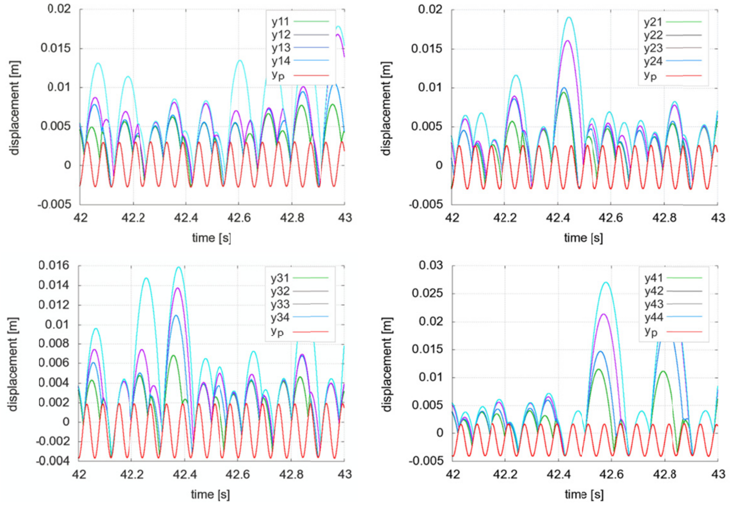

Fig. 6. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: $Ax = 0.0027$ [m], $Ay = 0.0028$ [m]

Case 2 – The mass of the feed constitutes 0.1 of the screen riddle mass m_o, coefficient of throw: 3.2.

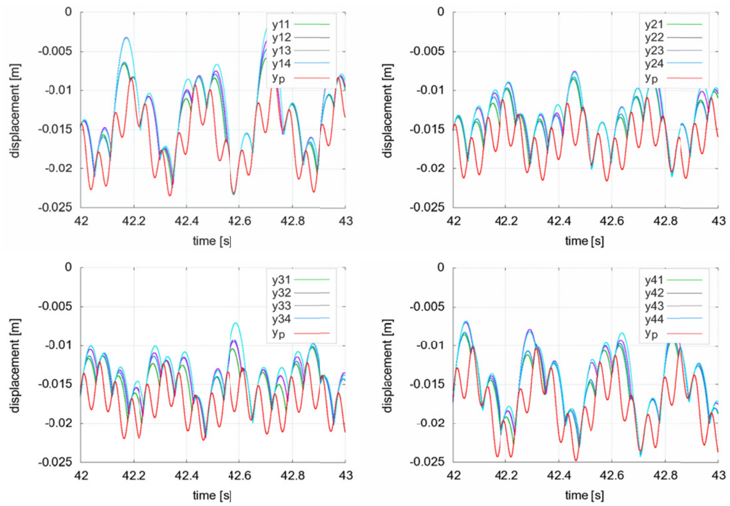

Fig. 7. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: $Ax = 0.0027$ [m], $Ay = 0.0027$ [m]. The average transport velocity in the steady state equals in this case 0.025 [m/s]

<u>Case 3</u> – The mass of the feed constitutes 0.2 of the screen riddle mass m_o, coefficient of throw: 3.2.

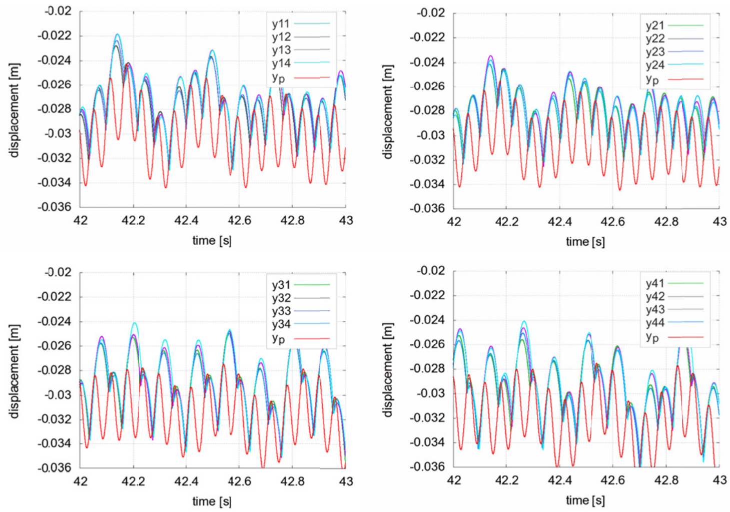

Fig. 8. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0026 [m], Ay = 0.0026 [m]. The average transport velocity in the steady state equals in this case 0.28 [m/s].

<u>Case 4</u> – The mass of the feed constitutes 0.3 of the screen riddle mass m_o, coefficient of throw: 3.2.

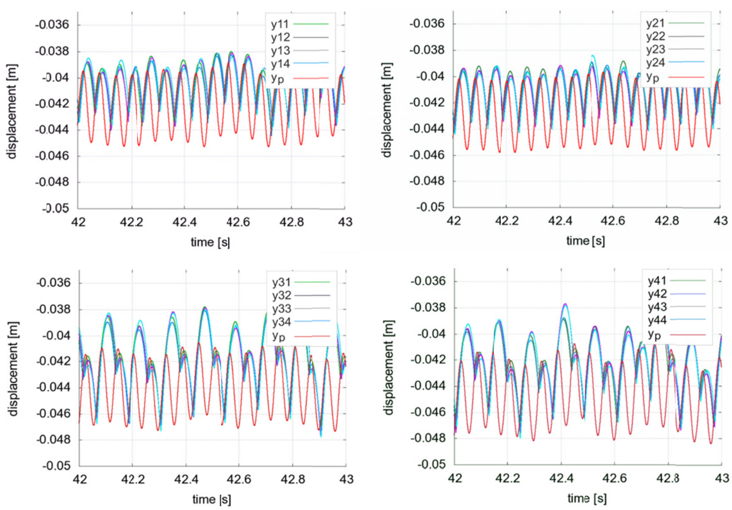

Fig. 9. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0025 [m], Ay = 0.0027 [m]. The average transport velocity at a steady state equals in this case 0.30 [m/s].

<u>Case 5</u> – The mass of the feed constitutes 0.4 of the screen riddle mass m_o , coefficient of throw: 3.2.

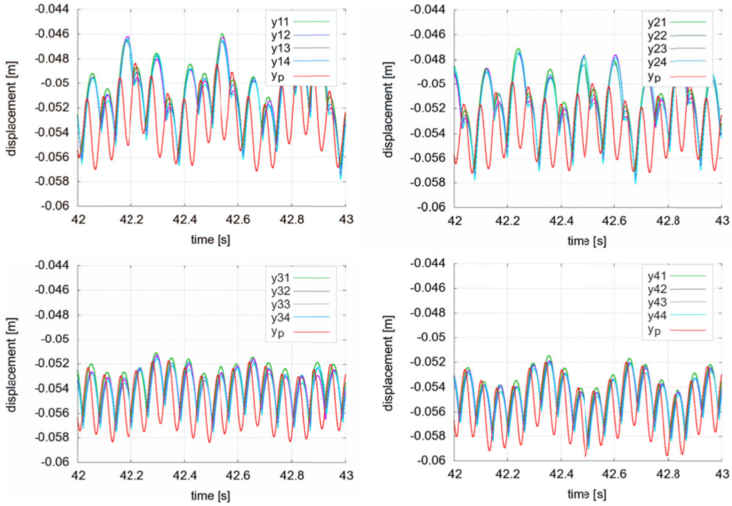

Fig. 10. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0025 [m], Ay = 0.0025 [m]. The average transport velocity in the steady state equals in this case 0.31 [m/s]

<u>Case 6</u> – The lack of the feed (the mass of the feed constitutes 0.001 of the screen riddle mass m_o), coefficient of throw: 5.4.

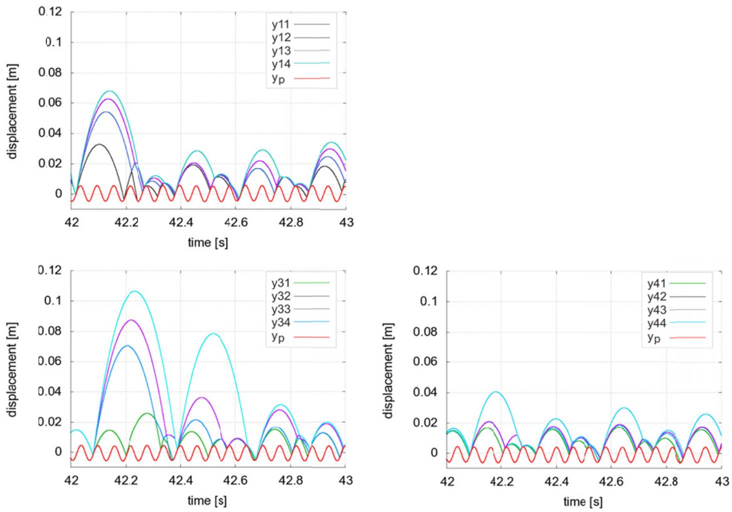

Fig. 11. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0048 [m], Ay = 0.0049 [m]

Case 7 – The mass of the feed constitutes 0.1 of the screen riddle mass m_o, coefficient of throw: 5.4.

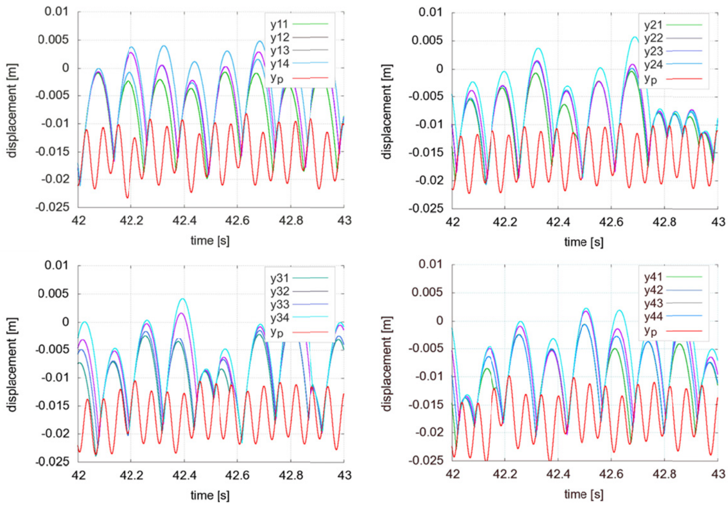

Fig. 12. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0048 [m], Ay = 0.0049 [m]. The average transport velocity in the steady state, equals in this case 0.42 [m/s]

Case 8 – The mass of the feed constitutes 0.2 of the screen riddle mass m_o, coefficient of throw: 5.4.

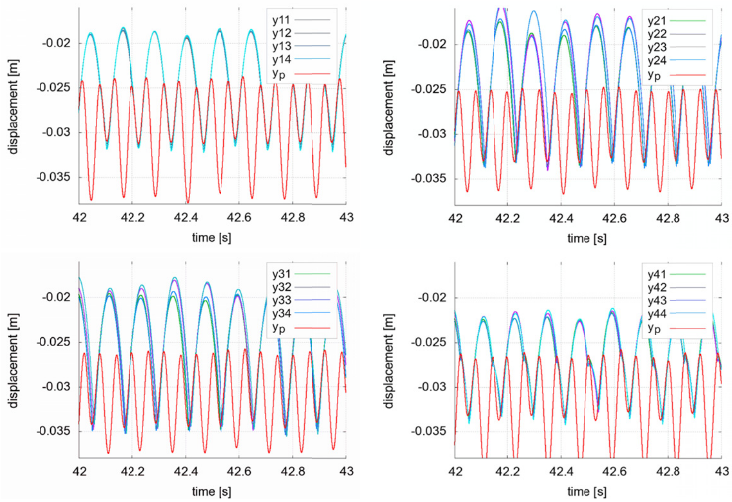

Fig. 13. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0047 [m], Ay = 0.0047 [m]. The average transport velocity in the steady state, equals in this case 0.45 [m/s]

<u>Case 9</u> – The mass of the feed constitutes 0.3 of the screen riddle mass m_o , coefficient of throw: 5.4.

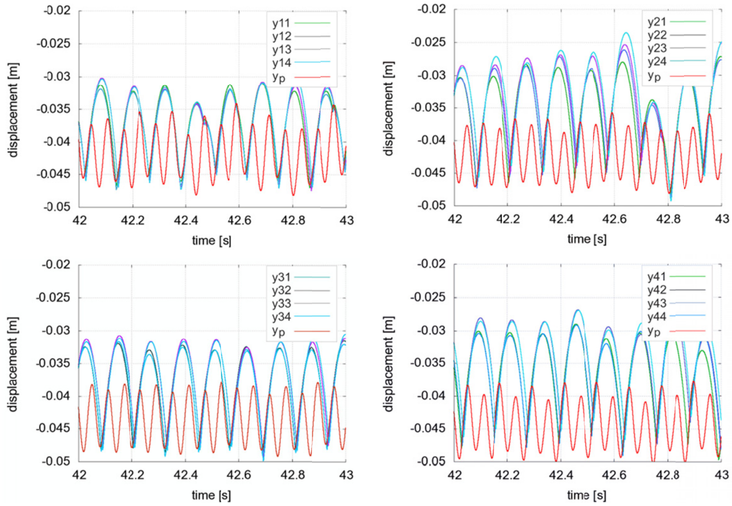

Fig. 14. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0.0046 [m], Ay = 0.0047 [m]. The average transport velocity in the steady state, equals in this case 0.31 [m/s]

<u>Case 10</u> – The mass of the feed constitutes 0.4 of the screen riddle mass m_o, coefficient of throw: 5.4.

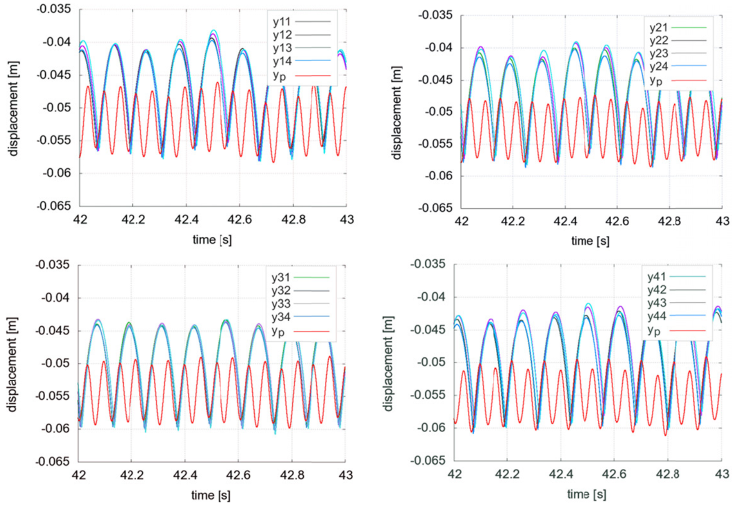

Fig. 15. Displacements of individual columns of the feed y_{ij} and the displacement of the screen riddle y_p in the contact place with the given column of the feed. Amplitudes of the machine body vibrations: Ax = 0,0046 [m], Ay = 0,0045 [m]. The average transport velocity in the steady state, equals in this case 0.22 [m/s]

5. Conclusions

The conclusions shown below are drawn on the bases of the results of the performed simulation investigations of the feed material influence on the operations of the suspended screen.

1°. The feed material presence does not change the fact that in transient states the vibration amplitudes, in all motion directions, are usually higher during coastings than during start-ups.

2°. The feed material presence significantly lowers (up to 2-times for coasting) maximal vibration amplitudes in the transient resonance in the direction y, perpendicular to the sieve. This is important since vibrations in this direction are usually sources of high dynamic reactions influencing the environment. This effect is probably related to the increased – due to the feed material presence – damping and inertia in the system limiting the maximum values of amplitudes.

3°. In the remaining directions of motion: x and α the feed material presence does not explicitly influence maximal amplitudes in transient states. However – by decreasing vibration amplitudes – it decreases the vibratory moment being of the damping character and due to that causes a prolongation of the vibrator coasting.

4°. Changes in the feed material amount on the riddle, in the range from 0 to 40% of its mass, do not visibly influence the engine driving moment required for performing the first half-turn.

5°. The feed material presence reduces the vibrations amplitudes of the riddle – in the steady state – more intensely when the coefficient of throw is lower than 3.3 (up to 9% when the mass of the feed constitutes 40% of the mass of the riddle m_o) and less intensely (only up to 6%) when this coefficient exceeds 3.3.

6°. An increased mass of the feed material makes flying of sieved particles more steady and causes their shaking process more regular and in conformity with the theory of a material point movement on vibrating surfaces (single-stroke motion for the coefficient of throw values lower than 3.3 and double-stroke motion for its higher values).

7°. Increasing the mass of the feed material in a range from 0.1 to 0.4 of the mass of the riddle influences the transport velocity differently in dependence of the coefficient of throw value. For the coefficient of throw being 3.2 increasing the feed amount above 0.1 of the riddle mass increases the average transport velocity up to app. 25%. When the coefficient of throw is 5.4 increasing of the feed amount causes at first a small increase of the average transport velocity of the feed along the sieve of the suspended screen followed by its significant decrease to a half of maximuml value.

References

Banaszewski T.,1990. *Przesiewacze.* Wyd. Śląsk.

Blechman I.I., 1994. *Vibratory Mechanics*. Nauka, Moskva.

Czubak A., 1964. *Przenośniki wibracyjne*. Śląsk, Katowice.

Czubak A., Michalczyk J., 2001. *Teoria transportu wibracyjnego*. Wyd. Pol. Świętokrzyskiej, Kielce.

Goncharevich I., Frolov K., Rivin E., 1990. *Theory of Vibratory Technology.* Hemisphere Publ. Corp. N. York, Washington, London.

Goździecki M., Świątkiewicz H., 1975. *Przenośniki*. WNT, Warszawa.

Krulle C.A., Gotzendorfer A., Grochowski R., Rehberg I., Rouijaa M., Walzel P., 2007. *Granular Flow and Pattern Formation on a Vibratory Conveyor*. Traffic and Granular Flow'05, ISBN 978-3-540-47640-5 Springer Berlin Heidelberg New York.

Michalczyk J., Cieplok G., Bednarski Ł., 2010. Procesy przejściowe maszyn wibracyjnych i układów wibroizolacji. WNT, W-wa.

Michalczyk J., Czubak P., 2010a. *Influence of Collisions with a Material Feed on Cophasal Mutual Synchronization of Driving Vibrators of Vibratory Machines*. Journal of Theoretical and Applied Mechanics, No 1. Vol. 48.

Michalczyk J., Czubak P., 2010b. *Methods of Determination of Maximum Amplitudes in the Transient Resonance of Vibratory Machines*. Archives of Metallurgy and Materials, No 3. Vol. 55.

Michalczyk J., Czubak P., 2010. *Przyczyny błędów w oszacowaniu amplitud maksymalnych w rezonansie przejściowym.* Modelowanie Inżynierskie Nr 37.

Michalczyk J., Cieplok G., 2015. *Maximal Amplitudes of Vibrations of the Suspended Screens During the Transient Resonance*. Arch. Min. Sci. **61**, 3, 537-552.

Michalczyk J., 2008. *Phenomenon of Force Impulse Restitution in Collision Modelling*. Journal of Theoretical and Applied Mechanics, Vol. 46, No 4.

Plavinskij V. and others, 1969. *Masiny niepreryvnowo diejstvija.* Nauka, Moskva.

Turkiewicz W., Banaszewski T., 1982. *Wzrost amplitudy drgań podczas zatrzymywania przesiewaczy z napędami bezwładnościowymi.* Archiwum Górnictwa, z. 4.

Sinnott M.D., Cleary P.W., 2009. *Vibration-induced arching in a deep granular bed.* Granular Matter, Vol. 11, Iss. 5, p. 345-364.

Švacha M., 1971. *Nova typova rada vytlokuczich rostu a teoreticki rozbor vibracji.* Slevarenstvii, No 2/4.