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Transport properties of the new vibratory conveyor at operations in the resonance zone

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Abstract: The new vibratory conveyor intended for the precise material dosage was investigated in the hereby study. The transport possibilities in the circum-resonance zone were tested analytically as well as by simulation. The optimal working point of the system, allowing to lower the vibration amplitude of the eliminator on its own suspension, was found. Transport velocities in dependence on the excitation frequency were determined by simulation. The results were verified on the specially designed industrial conveyor whose parameters were determined using analytical and simulation investigations, built according to the patent application.

Keywords: vibrations, vibratory conveyor, dynamic eliminator, transport stopping, feeder, variable output

1 Introduction

The new vibratory conveyor, being the subject of the authors' patent application, allowing accurate dosage of the feed was investigated in the hereby study. It is used for the transport and dosage of loose materials or objects of small dimensions, providing the possibility of a sudden stoppage of transport, regardless of the application of a small and relatively cheap electrovibratory drive [1–3].

It often occurs in the production line that there is either a necessity of the immediate feeding or a sudden

stoppage of the feed material flow. In case of classic vibratory conveyors, their switching off is connected with the necessity of the system passing through successive resonance zones at start-up and coasting of the machine. During such stages, an uncontrolled transport occurs.

Feeders of small sizes are most often made on the suspension of leaf springs and they are usually driven by the expensive electromagnetic excitation [4], which provides a total and immediate control over the trough movement allowing the material dosage [5]. The solution, in which the dosing feeder is driven by the system of counter running vibrators, is also known [6]. In this solution, the dosage by decreasing the transport velocity is realized by a significant decrease in the excitation frequency causing steep angles of the material feed flow at the trough edge. The wide review of various solutions of conveyors allowing for the precise material dosage can be found in paper [7].

The authors investigated the transport possibilities of the patented conveyor in the zone being in front of the antiresonance related to an additional eliminator and in the antiresonance zone. The main problem of this type of operation constitutes a significant amplitude of the eliminator, which is placed on a highly stiff suspension, during the steady state work at a relatively high amplitude of the trough (allowing the typical transport velocity for the given machine class). High amplitudes of the stiff suspension will be causing its cracking. In this article, the authors investigated the transport possibility in the second resonance zone, which significantly decreased the eliminator amplitude in relation to the through amplitude at the material transport with a typical velocity. Such control of the system also provides a possibility of additional increase in the transport velocity, while maintaining safe amplitudes of the eliminator. On the basis of the analytical and simulation investigations performed in this work, the conveyor – of parameters allowing its operation in the second resonance zone – was constructed. Experimental tests of this conveyor fully confirmed previous analyses.

Works concerning the analysed solution cannot be found in the world scientific references, since this is the

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authors' patent. However, several papers concerning conveyors, in which the trough constitutes the dynamic eliminator, can be found. These conveyors are base excited, not as in the hereby article, by the vibration excitation of elements attached to the trough. The first construction of the conveyor operating on the basis of dynamic elimination was described in the paper [8]. In the paper [9], authors analysed the conveyor behaviour around the assumed excitation frequency (being the vibroinsulating Frahm's damper). The authors noticed that in this type of conveyors, due to the attachment of the excitation system to the mass, which – in theory – is not vibrating, the service life of this system as well as of its suspension is prolonged. The same author with the team [10] investigated the possibility of controlling the excitation frequency around the work point. In the paper [11], authors tested the material feed influence on the conveyor operating on the Frahm's eliminator bases. The author substituted the conveyor model, in which the feed was moving in the direction of the trough movement, by the two-mass model, in which the feed was influencing in vertical direction. The advantage of such solutions of base excited conveyors is minimisation of forces transmitted to foundations; however, regardless of a relatively similar construction to the one proposed in this article, they are suitable neither for dosing nor for rapid stopping of the feed flow.

2 Construction of the analysed conveyor

The new vibratory conveyor [1,2], being the subject of the patent application, is presented in Figure 1. This conveyor is built of a classic trough 1, elastically supported 2 on stiff base in the horizontal orientation. It is equipped with the system of two counter running vibrators 3, suspended on the conveyor trough at an angle β . In the steady state, these vibrators are synchronised and rotate in different directions, providing the rectilinear resulting force, passing through the mass centre of the trough system as well as through the centre of its suspension system. Drive motors of inertial vibrators are equipped with inverter 4, by means of which the control system is able to control the rotational speed of electrovibrators 3.

The inertial drive contrary to the electromagnetic drive is much cheaper in terms of cost and exploitation. In practice, this type of drives are not subjected to breakdowns, and when they happen most often they are related to over-used bearings (so the repair is cheap). A negative side of this type of drives constitutes the necessity of maintaining the self-synchronisation of two electrovibrators providing the directional force. In case of operations in the antiresonance zone, at a wrong selection of the

Figure 1: Schematic presentation of the conveyor (according to the invention).

supporting points of these electrovibrators on the trough, maintaining this self-synchronisation might not be successful and machine operations might be impossible [12]. Other disadvantages, as compared to the electromagnetic drive, are long coasting time as well as longer reaction for the frequency change tasks.

The coupling method for the electrovibrators is based on a self-synchronisation, which at the proper parameters, based among others, on moving away of electrovibrators from the centre of gravity, occurs spontaneously [12]. The utilised electrovibrators are three-phased and star connected. Each of them has 40 W – which is power efficient, since usually conveyors weighing approximately above 50 kg need electrovibrators of a power of 150–200 W for their operations. There is a scalar control of a rotational speed (Volts-per-Hertz) and one inverter controls both the motors simultaneously. There is no vector controlling of the speed of the electrovibrators because the self-synchronisation would not be possible.

Additional mass 5, on its own suspension 6, is connected to the main mass 1. The aim of this additional mass is the elimination of trough vibrations at the proper control of the excitation frequency of vibrators, in accordance with the Frahm's eliminator principle [13], when the excitation frequency ω is equal to

$$\omega = \sqrt{\frac{k_T}{m_e}}, \quad (1)$$

stoppage of trough vibrations causes stopping of the feed transport 7.

In the previous works of the authors [14], the possibility of a material transport was investigated for the excitation frequency lower than the antiresonance frequency of the system, while the transport stoppage was tested at the antiresonance frequency. In this article, it was revealed that the conveyor operation is more favourable behind the antiresonance frequency, since this allows a decrease in the maximal amplitudes of the eliminator.

3 Analytical studies of the substituting system of two degrees of freedom

Based on the assumptions that the centres of gravity of the trough and eliminator are overlapping and stiffness of suspensions in x and y directions is the same and that the force originated from synchronised electrovibrators passes through the centre of gravity, the system presented

in Figure 1 can be substituted by the two-mass system presented in Figure 2 [14].

Equations of the system are in the form:

$$\begin{aligned} m_k \ddot{x}_2 + b_2 \dot{x}_2 + k_2 x_2 + b_1 (\dot{x}_2 - \dot{x}_1) + k_1 (x_2 - x_1) &= P_0 \sin(\omega t) \\ m_e \ddot{x}_1 - b_1 (\dot{x}_2 - \dot{x}_1) - k_1 (x_2 - x_1) &= 0, \end{aligned} \quad (2)$$

where x_1 – displacement of the eliminator harmonic movement along coordinate x_1 , x_2 – displacement of the trough harmonic movement along coordinate x_2 , ω – excitation circular frequency of force $P(t)$, m_k – mass of the conveyor trough, m_e – mass of the dynamic eliminator, k_1 and b_1 – total elasticity and damping of suspension in working direction x_1 , respectively, and k_2 and b_2 – total elasticity and damping of suspension in working direction x_2 , respectively.

Values of above, corresponding annotations, are listed in Table 1.

Figure 3 presents amplitude-frequency characteristics of the trough and eliminator; shown amplitudes are of work direction, and represent the strategy of operating before antiresonance. Both masses have the first common resonance at approximately 35 rad/s. Then, the trough amplitude together with ω decreases, which causes, in this case, requirement of relatively large electrovibrators. On the other side, the eliminator amplitude abruptly increases with ω and in the final work zone equals more than 15 mm. Analogous characteristics as in Figure 3 are presented in Figure 4, but for the strategy of controlling behind the antiresonance. The first common resonance occurs at approximately 38 rad/s, and next the trough amplitude slowly approaches zero (117 rad/s). Behind this point, both amplitudes increase with ω , entering the resonance slope. In such a way the identical coefficient of throw is obtained at three times smaller electrovibrators than in a previous strategy and the achieved amplitudes of

Figure 2: Simplified two-mass model of the device.

Table 1: Parameters of the simulated system

Symbol	Value	Unit	Definition
l_1	0.177	m	Linear dimension
l_2	0.177	m	Linear dimension
h	0	m	Linear dimension
h_1	0.045	m	Linear dimension
h_2	0.177	m	Linear dimension
a_1	0.231	m	Linear dimension
a_2	0.155	m	Linear dimension
d	0.35	m	Linear dimension
H	0.12	m	Linear dimension
e_1	0.02	m	Radius of a vibrator unbalance
e_2	0.02	m	Radius of a vibrator unbalance
k_e	320,002	N/m	Stiffness of suspension
k_x	71,690	N/m	Stiffness of suspension
k_y	71,690	N/m	Stiffness of suspension
b_x	90	N s/m	Coefficient of viscous damping
b_y	90	N s/m	Coefficient of viscous damping
b_e	300	N s/m	Coefficient of viscous damping
m_r	28	kg	Mass of the trough
m_1	0.5	kg	Rotating mass I
m_2	0.5	kg	Rotating mass II
m_e	26	kg	Mass of the eliminator
J_1	0.001	kg m ²	Central moment of inertia of the excitation I
J_2	0.001	kg m ²	Central moment of inertia of the excitation II
J_r	2.56	kg m ²	Central moment of inertia of the trough
J_e	0.2	kg m ²	Central moment of inertia of the eliminator
M_{ut}	5	N m	Breakdown torque
ω_{ss}	Variable	rad/s	Synchronous speed
ω_{ut}	Variable	rad/s	Breakdown torque speed
R	0.13	—	Restitution coefficient
μ	0.4	—	Coefficient of friction
k_s	10^8	N/m	Hertz-Staierman stiffness
p	1	—	Hertz-Staierman constant
b_{s1}	0.0009	Nm s	Coefficient of viscous damping in bearing
b_{s2}	0.0009	Nm s	Coefficient of viscous damping in bearing

the additional mass are more than three times smaller, which is beneficial for the suspension.

After determining amplitudes of the trough and eliminator, for parameters shown in Figures 3 and 4, it was revealed that in order to obtain the proper throw coefficient at not very high amplitudes of the eliminator, the system working behind the antiresonance frequency is more beneficial. Eliminator amplitudes are 0.0095 m when the system is tuned to work before the antiresonance frequency and 0.0025 m when the system is working behind this frequency (Figure 4), for the same transport velocity resulting from the throw coefficient. In turn, the throw

Figure 3: Amplitudes of the trough (red line) and the eliminator (blue line) as a function of the excitation frequency, for the strategy of operating before the antiresonance.

coefficient depends on the amplitude and the square of the excitation frequency. In addition, at the work on the slope of the second resonance, the proper throw coefficient can be achieved at a much lower drive (four times lower than in case of work before the antiresonance), which significantly decreases costs of building and maintenance of the machine.

Controlling the strategies can be achieved by controlling the system represented by the scheme, as shown in Figure 10.

4 Simulation investigations

Simulation investigations were performed with taking into consideration the material feed, due to a strong dependence of the eliminator efficiency on the damping in the system [9,15,16]. Calculations performed without

Figure 4: Amplitudes of the trough (red line) and the eliminator (blue line) as a function of the excitation frequency, for the strategy of operating behind the antiresonance.

Figure 5: Trough oscillations (blue line) carrying feed material.

taking into consideration the feed are usually burdened with high errors [17]. In order to determine the transport velocity in dependence on the system excitation frequency – in the analysed conveyor – the system presented in Figure 1 (together with the feed) was studied. Problems with the self-synchronisation can also occur in such system [18]. The analysed system [14] consists of two inertial vibrators with induction motors (described by a static characteristic) which excite the trough suspended on the spiral springs. The additional mass on the elastic suspension is connected to the trough containing the feed material [19,20].

Matrix equation (6), contains equations (7)–(9). These equations were formulated assuming following energies of the system shown in Figure 1.

$$\begin{aligned}
 E_k = & \frac{1}{2}m_r(\dot{x}^2 + \dot{y}^2) + \frac{1}{2}m_e((\dot{x} + \dot{\tau} \cdot \cos(\beta))^2 \\
 & + (\dot{y} + \dot{\tau} \cdot \sin(\beta))^2) \\
 & + \frac{1}{2}m_1(\dot{x} + h_1\dot{\alpha} + \dot{\phi}_1 e_1 \cos(\phi_1))^2 \\
 & + \frac{1}{2}m_1(\dot{y} - a_1\dot{\alpha} + \dot{\phi}_1 e_1 \sin(\phi_1))^2 \\
 & + \frac{1}{2}m_2(\dot{x} + h_2\dot{\alpha} + \dot{\phi}_2 e_2 \cos(\phi_2))^2 \\
 & + \frac{1}{2}m_2(\dot{y} - a_2\dot{\alpha} - \dot{\phi}_2 e_2 \sin(\phi_2))^2 \\
 & + \frac{1}{2}(J_r + J_e)\dot{\alpha}^2 + \frac{1}{2}J_{01}\dot{\phi}_1^2 + \frac{1}{2}J_{02}\dot{\phi}_2^2,
 \end{aligned} \quad (3)$$

$$\begin{aligned}
 E_p = & \frac{1}{2}k_x(x + h\alpha)^2 + \frac{1}{4}k_y(y - l\alpha)^2 + \frac{1}{4}k_y(y + l_2\alpha)^2 \\
 & + \frac{1}{2}k_r\tau^2 - m_1g \cdot e_1 \cdot \cos(\phi_1) + m_2g \cdot e_2 \cdot \cos(\phi_2),
 \end{aligned} \quad (4)$$

$$\begin{aligned}
 N = & \frac{1}{2}b_r(\dot{\tau})^2 + \frac{1}{2}b_x(\dot{x} + h\dot{\alpha})^2 + \frac{b_y}{4}(\dot{y} + l_1\dot{\alpha})^2 \\
 & + \frac{b_y}{4}(\dot{y} + l_2\dot{\alpha})^2,
 \end{aligned} \quad (5)$$

where E_k – kinetic energy, E_p – potential energy, and N – dissipation potential.

The mathematical model of such system consists of matrix equation (6) describing the machine motion, equation (14) describing the electromagnetic moment of drive motors, equations (12) and (13) used for determining the motion of successive feed layers, as well as of dependencies, Equations (10) and (11) describing normal and tangent influences in between feed layers and between the feed and machine body, respectively [21].

$$[M] \cdot [\ddot{q}] = [Q], \quad (6)$$

$$M = \begin{bmatrix} m_r + m_1 + m_2 + m_e & 0 & m_1h_1 + m_2h_2 & m_1h_1 + m_2h_2 & m_1h_1 + m_2h_2 & m_e \cos \beta \\ 0 & m_r + m_1 + m_2 + m_e & -m_1a_1 - m_2a_2 & m_1e_1 \cos(\phi_1) & -m_2e_2 \sin(\phi_2) & m_e \sin \beta \\ m_1h_1 + m_2h_2 & -m_1a_1 - m_2a_2 & m_2(h_2^2 + a_2^2) + m_1(h_1^2 + a_1^2) + J_r + J_e & m_1h_1e_1 \cos(\phi_1) - m_1a_1e_1 \sin(\phi_1) & -m_2h_2e_2 \cos(\phi_2) + m_2a_2e_2 \sin(\phi_2) & 0 \\ m_1e_1 \sin(\phi_1) & m_1e_1 \cos(\phi_1) & m_1h_1e_1 \sin(\phi_1) - m_1a_1e_1 \cos(\phi_1) & m_1e_1^2 + J_{01} & 0 & 0 \\ m_2e_2 \cos(\phi_2) & m_2e_2 \sin(\phi_2) & m_2h_2e_2 \cos(\phi_2) - m_2a_2e_2 \sin(\phi_2) & 0 & m_2e_2^2 + J_{02} & 0 \\ m_e \cos \beta & m_e \sin \beta & 0 & 0 & 0 & m_w \end{bmatrix}, \quad (7)$$

$$\ddot{q} = [\ddot{x} \quad \ddot{y} \quad \ddot{\alpha} \quad \ddot{\phi}_1 \quad \ddot{\phi}_2 \quad \ddot{\tau}]^T, \quad (8)$$

$$Q = \begin{bmatrix} m_2 e_2 \dot{\phi}_2^2 \sin(\phi_2) + m_1 e_1 \dot{\phi}_1^2 \sin(\phi_1) - 2k_x(x + h\alpha) - 2b_x(\dot{x} + h\dot{\alpha}) - T_{1(01)} - T_{1(02)} - T_{1(03)} - T_{1(04)} - T_{1(05)} \\ m_2 e_2 \dot{\phi}_2^2 \cos(\phi_2) - m_1 e_1 \dot{\phi}_1^2 \cos(\phi_1) - k_y(y + l_1\alpha) - k_y(y - l_2\alpha) - b_y(\dot{y} + l_1\dot{\alpha}) - b_y(\dot{y} - l_2\dot{\alpha}) - F_{1(01)} - F_{1(02)} \\ - F_{1(03)} - F_{1(04)} - F_{1(05)} \\ m_1 h_1 e_1 \dot{\phi}_1^2 \sin(\phi_1) + m_1 a_1 e_1 \dot{\phi}_1^2 \cos(\phi_1) + m_2 h_2 e_2 \dot{\phi}_2^2 \sin(\phi_2) - m_2 a_2 e_2 \dot{\phi}_2^2 \cos(\phi_2) - 2k_x h^2 \alpha \\ - 2k_x h \dot{x} - 2b_x h \dot{x} - 2b_x h^2 \dot{\alpha} - k_y(y + l_1\alpha)l_1 + \\ k_y(y - l_2\alpha)l_2 - b_y(\dot{y} + l_1\dot{\alpha})l_1 + b_y(\dot{y} - l_2\dot{\alpha})l_2 + (T_{1(01)} + T_{1(02)} + T_{1(03)} + T_{1(04)} + T_{1(05)})H \\ + 2dF_{1(01)} + dF_{1(02)} - dF_{1(04)} - 2dF_{1(05)} \\ M_{el1} - b_{s1}\dot{\phi}_1^2 \text{sign}(\dot{\phi}_1) - m_1 g e_1 \sin(\phi_1) \\ M_{el2} - b_{s2}\dot{\phi}_2^2 \text{sign}(\dot{\phi}_2) + m_2 g e_2 \sin(\phi_2) \\ - k_\tau \tau - b_\tau \dot{\tau} \end{bmatrix}, \quad (9)$$

where τ – dependent coordinate, $F_{j,(j-1,k)}$ – normal component of the j -ts layer force on $j - 1$ in the k -ts column, $T_{j,(j-1,k)}$ – tangent component of the j -ts layer force on $j - 1$ in the k -ts column, j – material layer index $j = 0$ refers to the machine body, and k – index of the material layer column.

When the successive feed layers (in the given column) j and $j - 1$ are not touching, the contact force in the normal $F_{j,(j-1,k)}$ and tangent direction $T_{j,(j-1,k)}$ in between these layers equals zero. In the opposite case, the contact force occurs in the normal direction in between feed layers j,k and $j - 1,k$ (or in case of the first layer between this layer and the trough [21]). The model of this contact force is as follows:

$$F_{j,(j-1,k)} = (\eta_{j-1,k} - \eta_{j,k})^p \cdot k_s \cdot \left\{ 1 - \frac{1 - R^2}{2} [1 - \text{sgn}(\eta_{j-1,k} - \eta_{j,k})] \right\}. \quad (10)$$

There is also the force, originated from a friction, in the tangent direction:

$$T_{j,(j-1,k)} = -\mu F_{j,(j-1,k)} \text{sgn}(\dot{\xi}_{j,k} - \dot{\xi}_{j-1,k}). \quad (11)$$

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

$$m_{nj,k} \ddot{\xi} = T_{j,(j-1,k)} - T_{j+1,(j,k)}, \quad (12)$$

$$m_{nj,k} \ddot{\eta} = -m_{nj,k} g + F_{j,(j-1,k)} - F_{j+1,(j,k)}, \quad (13)$$

$$M_{el_i} = \frac{2M_{ut}(\omega_{ss} - \dot{\phi}_i)(\omega_{ss} - \omega_{ut})}{(\omega_{ss} - \omega_{ut})^2 + (\omega_{ss} - \dot{\phi}_i)^2}, \quad (14)$$

where M_{el} – moment generated by drive motors, M_{ut} – stalling torque moment of drive motors, ω_{ss} – synchronous frequency of drive motors, and ω_{ut} – stalling frequency of drive motors.

Simulations were performed for the parameters given in Table 1.

The mathematical model of the system contains all movements of the machine in a flat system (in case of the proposed excitation, the motion in the perpendicular direction will not be excited, and thus there is no need of investigating the spatial system). The system has three degrees of freedom of the main body, one for each electro-vibrator and the additional one for the eliminator mass.

The motion of the feed is presented by means of 40 equations. Each feed element has a degree of freedom in the perpendicular and horizontal direction. The perpendicular contact force containing elasticity of foundation and feed (depends which is contacting which) of each element is determined by equation (10). Whereas the horizontal contact force is given by equation (11), which contains coefficient of friction (μ) in between successive layers. In this way the slide is taken into consideration. Displacements in time of the trough as well as – placed on it – feed grains on the direction perpendicular to the trough are presented in Figure 5. The trough performs harmonic oscillations with a high acceleration causing grains detachment from the trough surface enabling the flight phase (Figure 5).

During the contact phase both the trough and feed are moving together, while during the flight phase the trough moves back, which causes mutual displacement of the trough and feed at each cycle. In such a way the feed material is moved on the trough in the expected direction.

4.1 Results of simulation studies

Transport velocity of the conveyor for quasi-stationary states as the frequency function, for various masses of the feed, is presented in Figure 6. For quasi-stationary states, the feed transport occurs at the first resonance

Figure 6: Transport velocity as a function of the excitation frequency in dependence of the feed mass for quasi steady states.

frequency; however, in the real conveyor, such state – at passing through the resonance – is so short that the transport does not occur. At the antiresonance frequency, there is a total lack of transport, while after passing this frequency the transport velocity increases. When the amplitudes of the trough and eliminator are increasing, the only real limitation of the velocity are the structural limitations resulting from too high amplitudes of main masses. It is seen in Figure 6 that at a significant increase in the feed mass, more than three times exceeding the trough mass, the significant decrease in the transport velocity occurs in the work place. There is a possibility of increasing the transport velocity at high masses of the feed by means of increasing the excitation frequency, taking into account the main mass amplitudes' technical limitations.

Diagram in Figure 6 was obtained on the bases of 300 simulations quasi-stable states for various feed masses and various excitation frequencies. Overlapped 2D diagrams formed 3D diagrams allowing to analyse the feed weight and excitation frequency influence on the transport velocity characteristic of the authors' conveyor. Experimental investigations allowing to achieve such precise diagram are – in practice – impossible due to multitude of these tests and dangers of drive overheating as well as destruction of suspension in the steady state during operations, e.g. in the resonance. The conformity of the model with the real tests is confirmed by Figures 7 and 12.

Figure 7 presents the transport velocity for feed mass (3 kg) in dependence of the excitation frequency. The exponential character of the velocity increase – after passing the antiresonance frequency – is seen there. This

Figure 7: Transport velocity as a function of the excitation frequency, obtained by simulation for the working strategy behind the antiresonance and at a small mass of the feed.

is a very positive situation since the velocity change, at frequencies being near the ones at which the transport stops, is small which significantly simplifies the precise dosage. On the other hand, a significant velocity increase occurs along with distancing from the point of transport stoppage allowing for the maximum velocity of the transport.

In Figure 7 (simulation diagram), at the frequency of 150 rad/s, a collapse of the transport velocity occurs. It results from the fact that at this frequency the system: trough – feed transfers from the single-stroke activity (one flight of a feed grain for one move of the trough) to double – stroke operation. According to equations allowing to determine the transport velocity in dependence of the amplitude and frequency of excitation for an

Figure 8: Prototype conveyor of the industrial scale for operations on the resonance slope.

Figure 9: Device view from the side of the drive system.

Figure 10: Scheme of how the machine is controlled: 1 – trough, 2 – eliminator, 3 – supporting frame, 4 – electrovibrators, 5 – variable-frequency drive, 6 – frequency tuning dial, 7 – four-core electrical wire, 8 – DC voltage source 230 V/50 Hz.

Figure 11: Trough carrying glass chippings.

individual grain [22], the velocity at transferring from a single to double stroke operation decreases. This effect is not observed in the real system (Figure 12) since the feed has in this case significantly more layers than in case of the simulation, in which only four layers are utilised. Investigations for the real system [22] also do not show any rapid velocity decreases at transferring from the single-stroke mode to the double-stroke one.

5 Testing of the industrial conveyor

After performing analytical and simulation studies, the conveyor of industrial dimensions was designed and built in accordance with guidelines resulting from the studies. The conveyor presented in Figure 8 has parameters listed in Table 1, used in previous analyses.

The side view of the conveyor, during testing its transport possibilities, is shown in the photograph 8.

Figure 9 presents the device from behind, where there is the excitation module. Inside the conveyor, the eliminator is attached to the backend wall. This eliminator is seen in technical scheme (Figure 10). Conveyor indicates very good transport properties and – what is most important – it indicates a possibility of the precise dosage. The time of the transport stopping from the maximum velocity to the full stop is not more than 1.5 s.

The example of its working can be seen at the YouTube address: <https://youtu.be/PmFnfySpJp8>. Investigations of the feed velocity were performed as follows: fragment of the feed, singled out by its colour (Figure 11), was subjected to a velocity measurement. In the moment when the assumed excitation velocity was stabilised, the observed

Figure 12: Experimental results of the feed transport.

grain was passing the starting point (established on the scale) and the timer started measuring. After passing the middle 0.5 m of the trough length and stopping the timer, the average experimental transport velocity of the feed was obtained.

In order to verify the conformity of the transport velocity between simulation models and the real machine, investigations were performed. For that purpose 3 kg mass was chosen to be tested against various frequencies, while using above-antiresonance-operating strategy. Figure 12 presents the real transport velocities determined on the real conveyor and they should be compared with the simulation results in Figure 7. A very good compatibility of the model with the real conveyor is seen in both figures, apart from the situation when the transport is transferring from a single to double stroke operation, which appears at the frequency of 150 rad/s.

6 Conclusion

- Analytical studies of the two-mass model allowed to select the proper parameters of the conveyor, allowing the reduction in the eliminator amplitudes when the conveyor is working behind the antiresonance zone.
- Simulation investigations of the conveyor proved the possibility of the device working as a feeder dosing loose feed material.
- Testing of the industrial conveyor constructed according to the patent application having parameters determined in the analytical and simulation investigations fully confirmed the ability of the conveyor to the feed transport (within a range of 0–100%).
- The dependence of the transport velocity on the excitation frequency is nearly the same in both the conveyors (numerically investigated one and the industrial conveyor).
- The accuracy of the numerical model will facilitate the planning and realisation of further tests of the machine.

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