Influence of the Excitation Frequency on Operations of the Vibratory Conveyor Allowing for a Sudden Stopping of the Transport

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Abstract

Transport possibilities of the new vibratory conveyor in dependence on the excitation frequency of inertial vibrators were investigated in the hereby paper. The comprehensive model of the machine together with the loose feed material was tested. Simulations allowed to determine the dependence of the transport velocity on the excitation frequency. The time needed for a sudden stopping of the transport was also investigated. The new controlling strategy, realised by means of the excitation frequency, was proposed for situations when there is a necessity of sudden stopping of the transport.

Keywords: vibratory conveyor, dynamic eliminator, transport stopping, variable efficiency, feeder

1. Allowing the conveyor for dynamic stop

The new vibratory conveyor, allowing to control the transported material velocity in dependence on the excitation force-frequency, was analysed in the hereby paper. This conveyor is used for transporting loose materials or objects of small dimensions, providing the possibility of sudden stopping of the transport, in spite of the application of a small and relatively cheap electro-vibratory drive [1,2].

Very often there is a need of stopping the conveyor operations in the production line. In case of vibratory conveyors, switching off of the conveyor means that the system will have to pass through successive resonance zones during the machine starting and coasting. Conveyors, being in these states, can transport larger amounts of materials than in the steady-state, and - in addition - are stalling, which does not allow for a sudden stopping of the transport.

Feeders of small dimensions are the most often produced on suspensions of flat springs and are driven by electromagnetic excitation, which offers a total and immediate control over the trough movements. Such structures are typical for the needs of dosing of the transported material [3]. The structures of larger overall dimensions of significant inertia as well as relatively high costs of electromagnetic drives constitute their limitations.

The conveyor of arbitrary dimensions, adapted for controlling the transportation velocity, is the conveyor produced by the Danish company Alvibra, patented as US20130268114A1 [4]. Its concept is based on the older idea, from 1994, (US5836204A) [5], where a two-mass system was forced to vibrate in the antiphase. The mentioned conveyor, in spite of an efficiency control, is not suitable for operating as a dosing feeder. The system of such low damping, in case of slowing down the transport velocity, will reduce its vibrations very slowly, in dependence of the ratio: the inertia to the system damping.

Another kind of conveyors, e.g. known from the patent description US 3064357 [6], is equipped with the system of sensors connected with the control system which, by analysing the feed layer thickness and transportation velocity, knows how much material was transported and is able to switch off the machine in the proper moment. The difficulty - in this case - constitutes an amount of the feed material transported during coasting, when machine vibrations are much higher than in the steady state.

2. Structure of the analysed conveyor

The new vibratory conveyor, being the subject of the patent application [1], is presented in figure 1. This vibratory conveyor is built of a classic trough 1, elastically 2 supported on a stiff base in a horizontal position. It is equipped with a system of two counter running vibrators 3, suspended to the trough at β angle. In a steady state, vibrators are synchronized and counter rotating providing the rectilinear resulting force, passing through the mass center of the trough system as well as through the center of its suspension system.

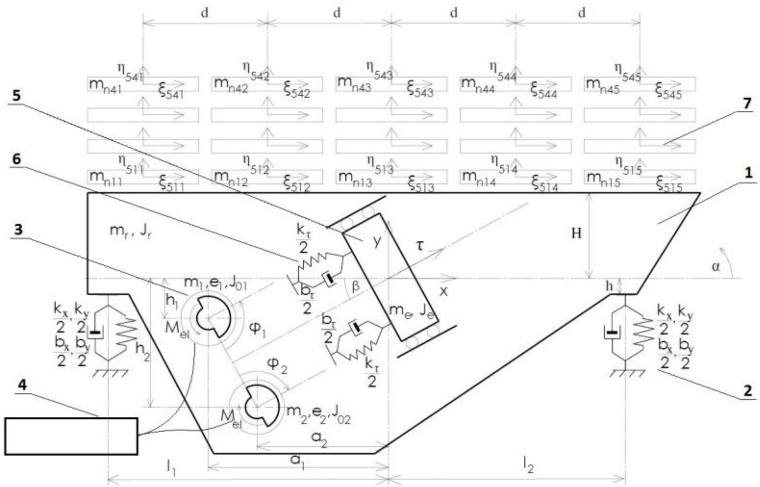

Figure 1. Scheme of the conveyor loaded with feed material - according to the invention

Motors of inertial vibrators are equipped with inverter 4, by means of which the controlling system is able to control the rotational speed of electrovibrators 3. 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 controlling of the excitation frequency of vibrators, according to the Frahm's eliminator rule. When the trough vibrations stop, the feed transport 7 also stops.

The conception predicts a possibility of the material transportation with a velocity typical for the given machine class, when the excitation frequency is significantly below the partial frequency of the eliminator additional mass 5 on its suspension 6. In order to stop the feed flow the excitation frequency should be changed in such a way as to equalize with the partial frequency of mass 5 on its suspension. When constants m_e and k_τ are selected in such a way, that the partial frequency is equal the frequency of the excitation force $\omega = \sqrt{k_\tau / m_e}$ vibrations of mass m_e are stabilizing and the force in spring k_τ equalises - in case of a lack or small damping in the system - the excitation force stopping the vibrations of the main mass, i.e. the trough. This effect occurs independently of the natural vibrations frequency of the basic system, before connecting the eliminator. The Frahm's eliminator system operates in the antiresonant basin, surrounded by both sides by the resonant zones, in addition to which the width of this basin depends on the ratio of the eliminator mass to the main mass [7].

Stopping the trough vibrations and in consequence stopping the feed transportation can be obtained in a short time and damping of trough vibrations is nearly total. It should be emphasised that the feed transport stops already at small trough vibrations, even before their complete damping, on account of a small coefficient of throw. An advantage of this solution constitutes the possibility of stopping the feed flow without the necessity of switching off the device, and in consequence without the necessity of the system passing through resonance zones [8]. This conveyor is very suitable for the accurate feed dosing, especially in the mode of often stopping of the transport. An additional advantage of this solution constitutes the fact that - when the transport stops - the feed distributes uniformly along the whole trough length. The transport restarting occurs when the excitation frequency of unbalanced masses decreases to the conveyor operational frequency in the steady-state. That time vibrations of the mass of the dynamic eliminator 5 will not significantly influence vibrations of the main mass 1 and the transport will restart.

Symbols below that are described and listed in figures and table stand for:

ω - rotational frequency of the vibrators,
m_r - mass of the conveyor trough,
m_e - mass of the dynamic eliminator,
m_1, m_2 - unbalanced masses of the vibrators,
e - mass eccentricity,
k_τ ,b_τ - summary of elasticity and damping of springs in the work direction τ,
k_x,k_y,b_x,b_y - summary of elasticity and damping of springs in the main suspension, rest of the symbols are visualized by the figures.

3. Simulation investigations

The simulation investigations were performed with taking into consideration feed materials, on account of a high dependence of the eliminator efficiency on the damping in the system [9,10]. Calculations performed without taking into consideration feed materials are usually burdened with large errors [11,12]. In order to find out the controlling strategy of the analysed conveyor, the determination of the transport velocity in dependence on the excitation frequency of the system is essential. This also provides the possibility of determining the width of the zone in which the feed transport decays.
The analysed system [2] contains two inertial vibrators, excited by induction motors described by a static characteristic. These vibrators are exciting the trough suspended on the system of spiral springs. Into the trough with feed the additional mass on elastic suspension is connected. The movement of this mass versus the trough is only possible along axis τ.

The mathematical model of such system is composed of the matrix equation (1) describing the machine motion, equation (7) describing the electromagnetic moment of drive motors, as well as dependencies (5) and (6) describing normal and tangent influences between feed layers and between the feed and machine body [13].

$$[M]\cdot[\ddot{q}]=[Q] \tag{1}$$

$$M=\begin{bmatrix} m_{11} & m_{12} & m_{13} & m_{14} & m_{15} & m_{16} \\ m_{21} & m_{22} & m_{23} & m_{24} & m_{25} & m_{26} \\ m_{31} & m_{32} & m_{33} & m_{34} & m_{35} & m_{36} \\ m_{41} & m_{42} & m_{43} & m_{44} & m_{45} & m_{46} \\ m_{51} & m_{52} & m_{53} & m_{54} & m_{55} & m_{56} \\ m_{61} & m_{62} & m_{63} & m_{64} & m_{65} & m_{66} \end{bmatrix} \tag{2}$$

where:

$m_{11}=m_r+m_1+m_2+m_e$; $m_{13}=m_{31}=m_1h_1+m_2h_2$; $m_{14}=m_{41}=m_1e_1\sin(\varphi_1)$;
$m_{15}=m_{51}=m_2e_2\sin(\varphi_2)$; $m_{16}=m_{61}=m_e\cos\beta$; $m_{22}=m_r+m_1+m_2+m_e$;
$m_{23}=m_{32}=-m_1a_1-m_2a_2$; $m_{24}=m_{42}=m_1e_1\cos(\varphi_1)$; $m_{25}=m_{52}=m_2e_2\sin(\varphi_2)$;
$m_{26}=m_{62}=m_e\sin\beta$; $m_{33}=m_2(h_2^2+a_2^2)+m_1(h_1^2+a_1^2)+J_r+J_e$;
$m_{34}=m_{43}=m_1h_1e_1\sin(\varphi_1)-m_1a_1e_1\cos(\varphi_1)$;
$m_{35}=m_{53}=m_2h_2e_2\cos(\varphi_2)-m_2a_2e_2\sin(\varphi_2)$;
$m_{44}=m_1e_1^2+J_{01}$; $m_{55}=m_2e_2^2+J_{02}$; $m_{66}=m_e$;
$m_{12}=m_{21}=m_{36}=m_{63}=m_{45}=m_{54}=m_{46}=m_{64}=m_{56}=m_{65}=0$

$$\ddot{q}=[\ddot{x}\quad \ddot{y}\quad \ddot{\beta}\quad \ddot{\varphi}_1\quad \ddot{\varphi}_2\quad \ddot{\tau}]^T \tag{3}$$

$$Q=\left[q_{11}\ q_{21}\ q_{31}\ q_{41}\ q_{51}\ q_{61}\right]^T \tag{4}$$

where:

$$q_{11} = m_2 e_2 \dot{\phi}_2^2 \sin(\phi_2) - m_1 e_1 \dot{\phi}_1^2 \cos(\phi_1) - 2k_x (x + h\alpha) - 2b_x (\dot{x} + h\dot{\alpha})$$
$$-T_{1(01)} - T_{1(02)} - T_{1(03)} - T_{1(04)} - T_{1(05)};$$
$$q_{21} = -m_2 e_2 \dot{\varphi}_2^2 \cos(\varphi_2) + m_1 e_1 \dot{\varphi}_1^2 \sin(\varphi_1) - k_y (y + l_1 \alpha) - k_y (y - l_2 \alpha) -$$
$$b_y (\dot{y} + l_1 \dot{\alpha}) - b_y (\dot{y} - l_2 \dot{\alpha}) - F_{1(01)} - F_{1(02)} - F_{1(03)} - F_{1(04)} - F_{1(05)};$$
$$q_{31} = -m_1 h_1 e_1 \dot{\varphi}_1^2 \cos(\varphi_1) - m_1 a_1 e_1 \varphi_1^2 \sin(\varphi_1) + m_2 h_2 e_2 \dot{\varphi}_2^2 \sin(\varphi_2) + m_2 a_2 e_2 \dot{\varphi}_2^2 \cos(\varphi_2) -$$
$$2k_x h^2 \alpha - 2k_x h^2 \alpha - 2k_x hx - 2b_x h\dot{x} - 2b_x h^2 \dot{\alpha} - k_y (y + l_1 \alpha) l_1 + k_y (y - l_2 \alpha) l_2 -$$
$$b_y (\dot{y} + l_1 \dot{\alpha}) l_1 + b_y (\dot{y} - l_2 \dot{\alpha}) l_2 (T_{1(01)} + T_{1(02)} + T_{1(03)} + T_{1(04)} + T_{1(05)}) H_r +$$
$$2dF_{1(01)} + dF_{1(02)} - dF_{1(04)} - 2dF_{1(05)};$$
$$q_{41} = \mathrm{M}_{el1} - b_{s1} \dot{\varphi}_1^2 \operatorname{sgn}(\dot{\varphi}_1) - m_1 g e_1 \cos(\varphi_1);$$
$$q_{51} = \mathrm{M}_{el2} - b_{s2} \dot{\varphi}_2^2 \operatorname{sgn}(\dot{\varphi}_2) - m_2 g e_2 \cos(\varphi_2);$$
$$q_{61} = -k_\tau \tau - b_\tau \dot{\tau}$$

where:

τ – dependent coordinate,
$F_{j,(j-1,k)}$ – normal component of the *j*th layer pressure on *j*-1 in the *k*th column,
$T_{j,(j-1,k)}$ – tangent component of the *j*th layer pressure on *j*-1 in the *k*th column,
j – index of the material layer, j=0 is related to the machine body,
k – index of the column.

If successive layers of the feed (in a given column) *j* and *j*-1 are not in contact, the contact force in the normal $F_{j,(j-1,k)}$ and tangent $T_{j,(j-1,k)}$ directions in between these layers is equal zero. If not, the contact force occurs in the normal direction between layers *j*,*k* and *j*-1,*k* of the feed (or in case of the first layer: between a layer and trough [13]), which model is of the following form:

$$F_{j,(j-1.k)} = (\eta_{j-1,k} - \eta_{j,k})^p \cdot k \cdot \left\{ 1 - \frac{1 - R^2}{2} \left[1 - \operatorname{sgn}(\eta_{j-1,k} - \eta_{j,k}) \cdot \operatorname{sgn}(\dot{\eta}_{j-1,k} - \dot{\eta}_{j,k}) \right] \right\} \tag{5}$$

and the force - originated from friction - acting in the tangent direction:

$$T_{j,(j-1,k)} = -\mu F_{j,(j-1,k)} \operatorname{sgn}(\dot{\xi}_{j,k} - \dot{\xi}_{j-1,k}) \tag{6}$$

$$\mathrm{M}_{el} = \frac{2\mathrm{M}_{ut} (\omega_{ss} - \dot{\varphi}_{i1}) \cdot (\omega_{ss} - \omega_{ut})}{(\omega_{ss} - \omega_{ut})^2 + (\omega_{ss} - \dot{\varphi}_i)^2} \tag{7}$$

M_{el} - moment generated by drive motors

M_{ut} - stalling torque of drive motors,

ω_{ss} - synchronous frequency of drive motors,

ω_{ut} - frequency of the stalling torque of drive motors.

Table 1. Parameters of the simulated system

Quantity	Symbol	Unit	Value
Dimension	l_1	m	1
Dimension	l_2	m	1
Dimension	h	m	0
Dimension	h_1	m	0.4
Dimension	h_2	m	0.8
Dimension	a_1	m	0.8
Dimension	a_2	m	0.4
Dimension	d	m	0.8
Dimension	H_r	m	0
Dimension	e_1	m	0.029
Dimension	e_2	m	0.029
Spring constant	k_e	N/m	2464900
Spring constant	k_x	N/m	1e5
Spring constant	k_y	N/m	1e5
Coefficient of elasticity	k	N/m	1e8
Dissipation constant	b_x	N·s/m	400
Dissipation constant	b_y	N·s/m	400
Dissipation constant	b_e	N·s/m	100
Dissipation constant	bs_1	N·s/m^2	9e–5
Dissipation constant	bs_2	N·s/m^2	9e–5
Mass	m_r	kg	360
Mass	m_1	kg	20
Mass	m_2	kg	20
Mass	m_e	kg	100
Moment of inertia	J_1	kg· m^2	0.01
Moment of inertia	J_2	kg· m^2	0.01
Moment of inertia	J_r	kg· m^2	50
Moment of inertia	J_e	kg· m^2	30
Breakdown torque	M_{ut}	N·m	50
Coefficient of friction	R	-	0.13
Coefficient of friction	μ	-	0.4
Synchronous radian frequency	ω_{ss}	rad/s	variable
Breakdown radian frequency	ω_{ut}	rad/s	variable

4. Results of simulation investigations

The system control assumed in paper [2] predicted the transport stop, however before stopping an increase of the transport velocity occurred. The same situation was at the transport restarting, where before obtaining the nominal velocity, the transport velocity was higher. Different frequency of the system operations - for the same parameters of the conveyor - is assumed in the new control strategy. When the steady-state frequency will change in such a way as to achieve the highest possible transport velocity, the system will not be able to increase this velocity when the frequency will be changing.

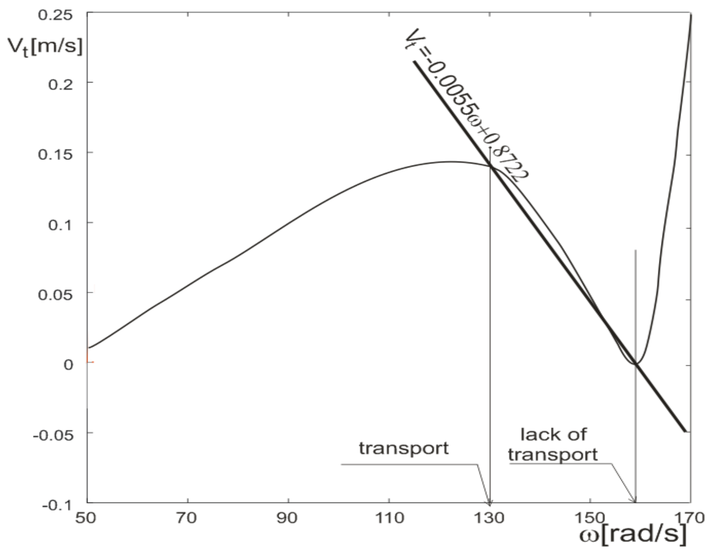

Figure 4. Transport velocity as a function of the excitation frequency

The diagram in figure 4 presents the transport velocity as the excitation frequency function for quasi-steady states. It is seen there, that for the frequency ω being above 130 [rad/s] the dependence of the transportation velocity on the excitation frequency can be - with a good approximation - given by a function: $V_t = -0.0055\omega + 0.8722$.

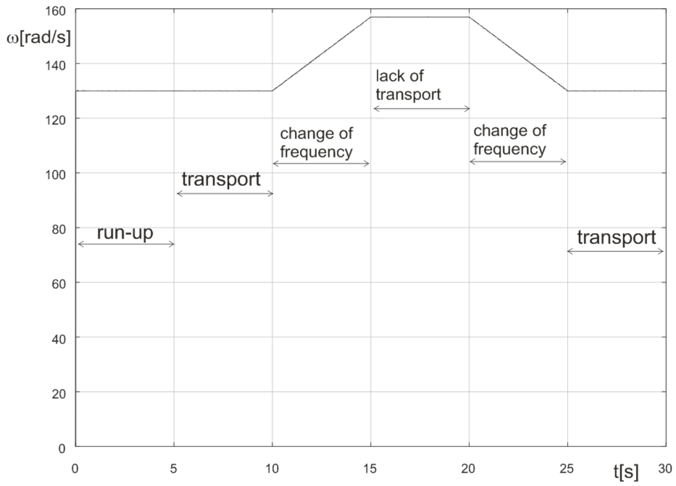

Figure 5. The angular velocity of vibrators

Utilising the effect of linear slowing down of the transport velocity (in Fig. 4 - after exceeding the excitation frequency 130 [rad/s] by the system), it was proposed to control the excitation frequency - in the range 130-157 [rad/s]. Figure 5 presents the proposed changes in the excitation frequency within the whole work range. The obtained reactions of the system are shown in figure 6. It can be seen, that the transport velocity of the feed smoothly changes - from its maximal value to the transport stopping - in 5 seconds. In a similar fashion, the velocity is changing - between the 20th and 25th second of operation - when the transport is restarting.

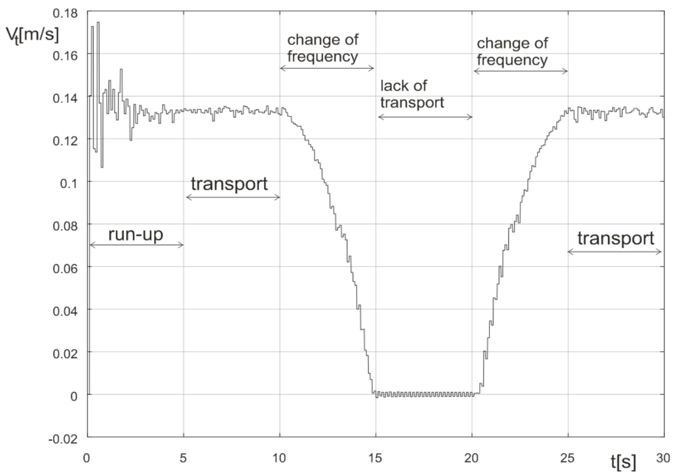

Figure 6. Transport velocity of the feed

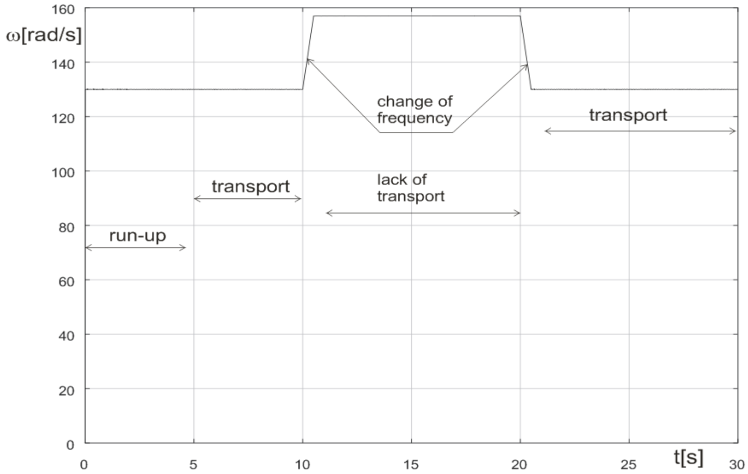

Figure 7. The angular velocity of the excitation

One of the main aims of the simulation procedure was the verification of the possibility of sudden stopping of the feed transportation. The diagram below (Fig. 7) presents the conveyor excitation frequency, at testing the total feed stopping in 0.5 s, while the diagram in figure 8 presents the feed transportation velocity in this case. Investigations indicate that this type of conveyor is not able to stop the transport in such a short time, and the residual transport occurs still 2 seconds after an attempt of its stopping. In a similar fashion, at another (very fast) restarting the feed flow rate will not be steady for approximately 2 seconds.

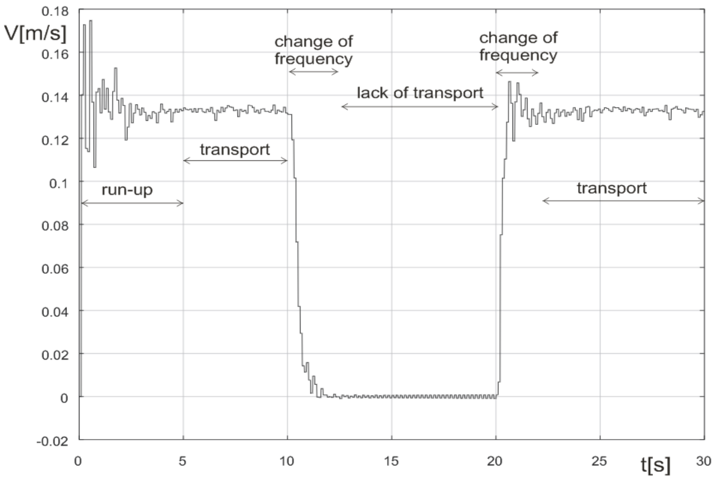

Figure 8. The feed transportation velocity

In spite of the fact that attempts indicated too long reaction time of the feeder, there is a method of performing the sudden stopping. The strategy, which should be undertaken in controlling the device, is a gradual slowing down of the feed flow rate, achieved by a gradual reduction of the excitation frequency. Then in the situation requiring the total transport stopping it will be possible to obtain this stopping even in times shorter than 0.5 s. Simulations presenting the assumed strategy are shown in figures 9 and 10.

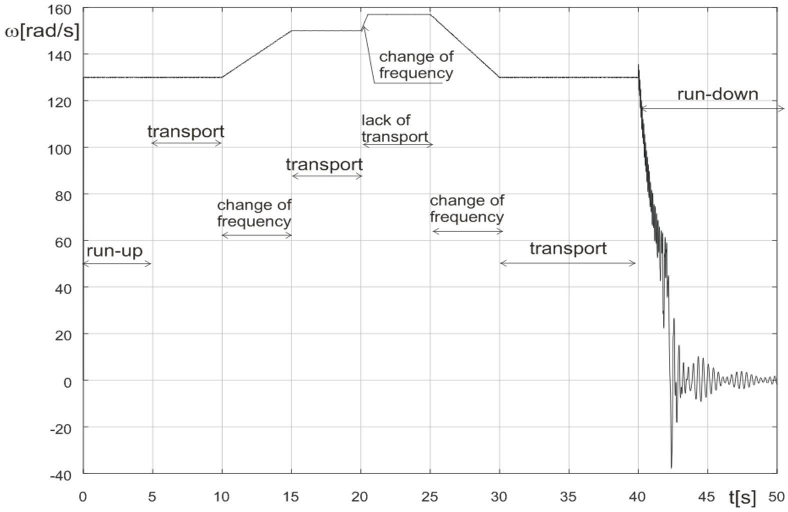

Figure 9. The angular velocity of the excitation

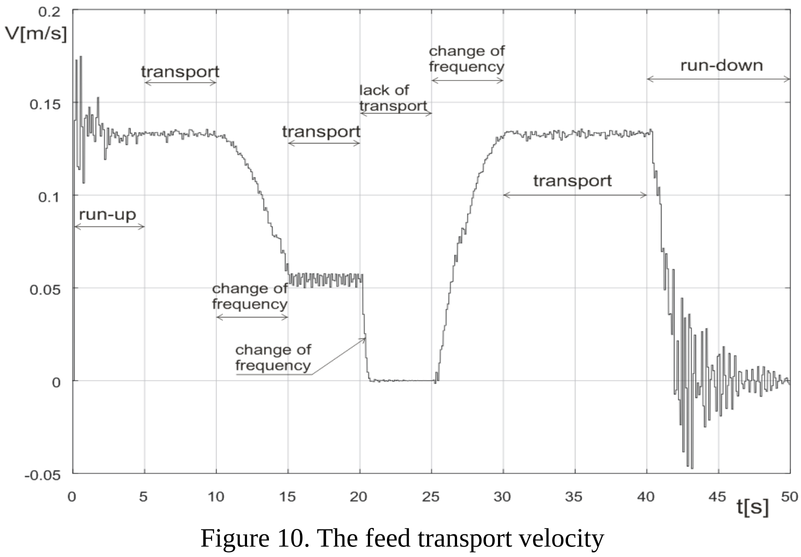

Figure 10. The feed transport velocity

5. Conclusions

- Simulation tests of the conveyor, according to the new concept, proved the possibility of its application as a feeder dosing loose feed materials.
- It was shown that the stopping of the feed flow from its full rate requires 2.5 s.
- At assuming the proper control strategy, it is possible to stop the transport in times below 0.5 s.
- The new feeder allows a smooth control of the efficiency within a range of 0-100%, as well as provides a possibility of the transport velocity approximation, as the linear function of the angular frequency of the drive.

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