VPS VIBRATIONS IN PHYSICAL SYSTEMS

DOI: 10.21008/j.0860-6897.2023.1.16
Vibrations in Physical Systems, 2023, 34(1), 2023116

Study of transport possibilities in the resonance zone of the new vibratory conveyor equipped with the single electro-vibrator

Weronika ŻMUDA, **Piotr CZUBAK**

AGH University of Science and Technology, Mickiewicza Av. 30, 30-059 Cracow, Poland

Corresponding author: Weronika ŻMUDA, email: wezmuda@agh.edu.pl

Abstract The new vibratory conveyor, which is during patent pending [1], equipped with a single electro-vibrator intended for an accurate material dosing, was investigated in the hereby study. Possibilities of material transportations and dosing both in and out the circum-resonant zone were investigated. Dependencies of the transport velocity of the tested conveyor as functions of the excitation frequency for a certain range of load were determined. The high usefulness of the machine in the production lines requiring accurate material dosing was indicated. It was pointed out that the conveyor can transport the feed at a constant velocity, regardless of the feed mass, which is unique for the conveyors of inertial drive.

Keywords: vibrations, vibratory conveyor, dynamic eliminator, transport stop, feeder, material dosing.

1. Introduction

Vibrating conveyors are used in many industries [2] due to their variety of designs and possible applications [3]. Very often, in the production line, there is a necessity to stop a feed transport or to dosing it. Classic structures driven by the system of electro-vibrators are usually not suitable for intermittent running, since in order to stop the feed transport their electro-vibrators must be switched off. This causes passing through resonance zones where, due to the increased vibrations amplitude, it is not possible to control a transport velocity.

Usually, vibratory conveyors are characterized by utilizing more than one inertial drive. Examples of such conveyors, in which the system of two electro-vibrators [4] allowing the accurate dosing of materials is applied, are patented devices which dynamic properties are analyzed in scientific literature[5, 6]. There are also known conveyors equipped with the expensive electromagnetic drive [7].

Nowadays, more and more emphasis is placed on the energy efficiency and a wide spectrum of possible usage of one solution on many levels. The conveyor presented in this article is characterized by an innovative [1] solution of using only a fraction of a force of a single inertial drive which would be needed for the classic drive. The following paragraphs present the wide spectrum of possible applications of the innovative device that enables both the transport and precise dosing of material of substantial mass without adversely affecting the reduction of the transport speed, regardless of the quantity of the feed. This is a unique property of the conveyors driven by one electro-vibrator. Another advantage of this new solution is a lack of problems related to self-synchronization of electro-vibrators [8] and energy saving system due to use of single motor, which is the problem already known and investigated [9], thereby highly desirable among industries.

The only one, known to the authors, vibrating conveyor allowing to maintain the constant velocity regardless of the feed mass is driven by two counter-running synchronized electro-vibrators [8, 10].

 DOI: 10.21008/j.0860-6897.2023.1.16

2. Scheme of the conveyor and principle of its operation

The scheme of the conveyor is presented in fig. 1a, while the conveyor constructed and build of industrial parameters in fig. 1b.

a) b)

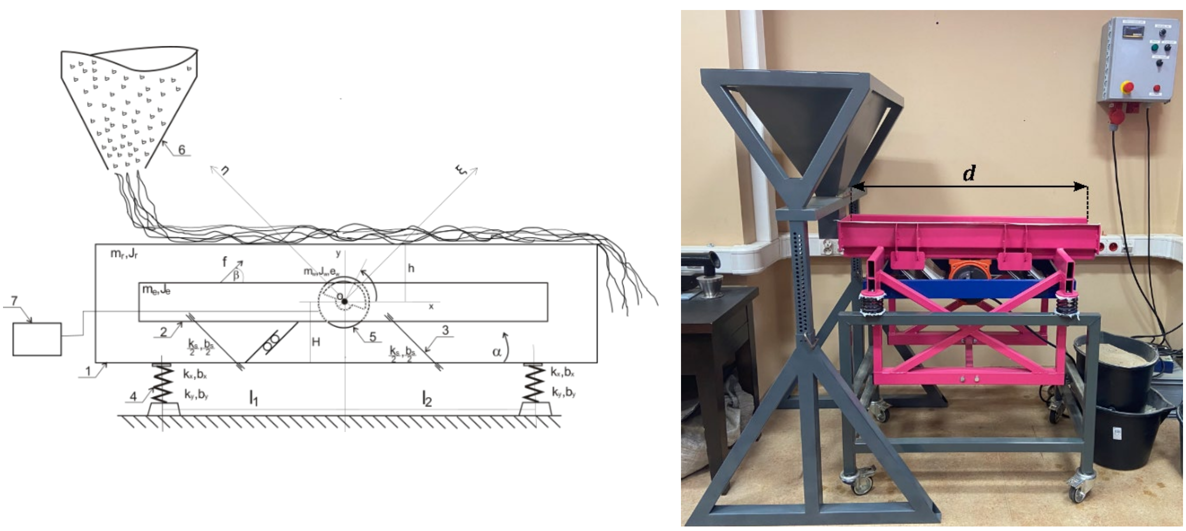

Figure 1. a) Scheme of the conveyor; b) real device.

The conveyor, according to the patent pending [1], is equipped with one electro-vibrator (5) suspended from the trough (1). The additional mass (2) on its own suspension (3) is added to the main mass (1). The aim of this additional mass is to eliminate vibrations of the trough in ξ direction. At the proper control of the excitation frequency of electro-vibrators - according to the Frahm's eliminator rule [11] - the trough vibrations in ξ direction will fade away, while vibrations in the perpendicular direction (η) will not be changed. The dynamic eliminator on its own suspension suspended from the trough eliminates vibrations of the trough in the working direction of its suspension system when the natural frequency of the eliminator on its suspension is equal to the excitation frequency of motor [12, 13].

a) b) c)

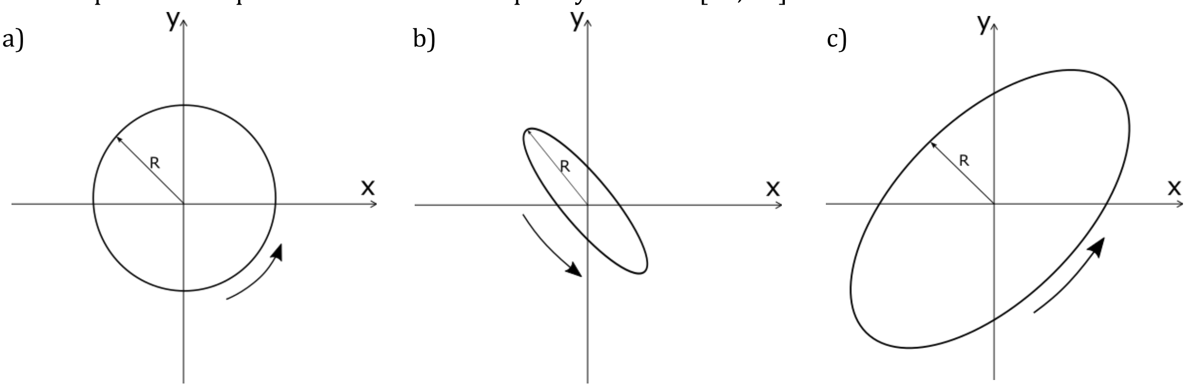

Figure 2. Motion scheme of the trough: a) outside the anti-resonance zone; b) in the anti-resonance zone; c) outside the anti-resonance zone (on the leading slope of second resonance).

The single vibrator, with the mass unbalance reduced to 12% of unbalance of typical conveyor of the same parameters (since - in practice - conveyors are designed with two-vibrators drive this reduction value is related to the sum of unbalances of two electro-vibrators) is applied. The device (suspended by the classic suspension) working outside the anti-resonance zone will perform movements similar to the circle in figure 2a. However, increase of the excitation frequency to the working frequency point causes the trough to vibrate in η direction in more elliptical way (fig. 2b). It happens, because in the anti-resonance point the dynamic eliminator eliminates vibrations in ξ direction but does not do so in the perpendicular one [14]. Further increase of the excitation frequency causes the device to make motion similar to one shown in fig. 2c. This is the working area of a leading slope of the second resonance of the device. The transport occurs then in the ξ direction with the velocity depending on the rotational speed of the drive.

Working characteristic of the conveyor allows to use the device in many ways. The one assumes working in the leading slope of the second resonance of the device with minor unbalanced mass equal to the 12% of the maximal possible one.

The device consists of three masses, as it was presented in Fig. 1. Due to the fact, that engine is rigidly connected to the trough, as well as taking into account that stiffnesses of the main suspension in directions x and y are the same, the machine motion can be considered as the motion in two independent directions ξ and η. It is considered for two masses working individually in ξ direction and collectively in η direction (Fig. 3). For η direction (Fig. 3a) the system works as one common mass being sum of body and eliminator due to characteristics of suspension of the dynamic eliminator. In perpendicular ξ direction (Fig. 3b) system is considered as two individually working masses. This is because of the working direction of the leaf springs. Analysis of the system in both directions states, that being outside the anti-resonance zone the device makes motion similar to circular one presented in Fig. 2a. Increasing of angular velocity of unbalanced mass results in elimination of vibrations of the trough in the working direction of leaf springs (ξ direction) and not eliminating ones in η direction what makes the system moves elliptically (Fig. 2b). Further increase of angular velocity of motor implies the system being positioned on the leading slope of the second resonance (Fig. 2c). In this case amplitudes of motion of the trough in ξ direction are higher than in perpendicular one, what results in the significant velocity of transportation in this direction.

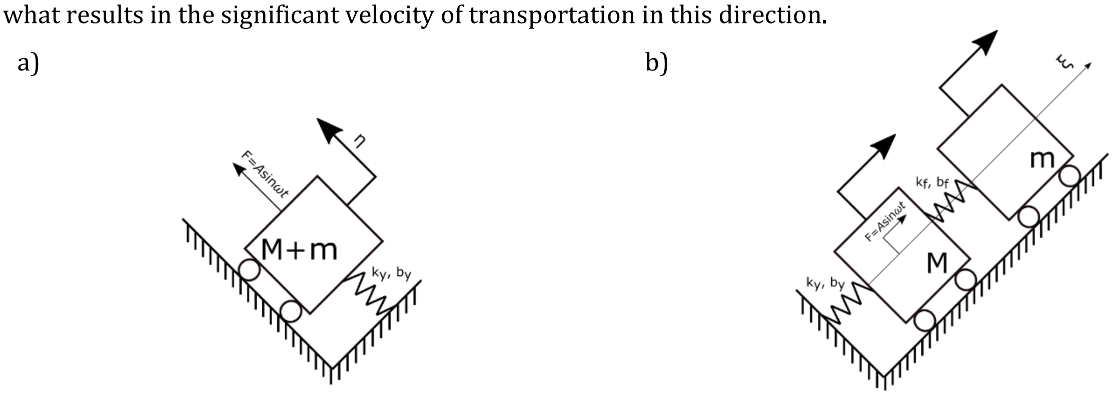

Figure 3. Analysis of motion of the scheme in two independent directions; a) system of one mutual mass in η direction; b) system of two separate masses in ξ direction.

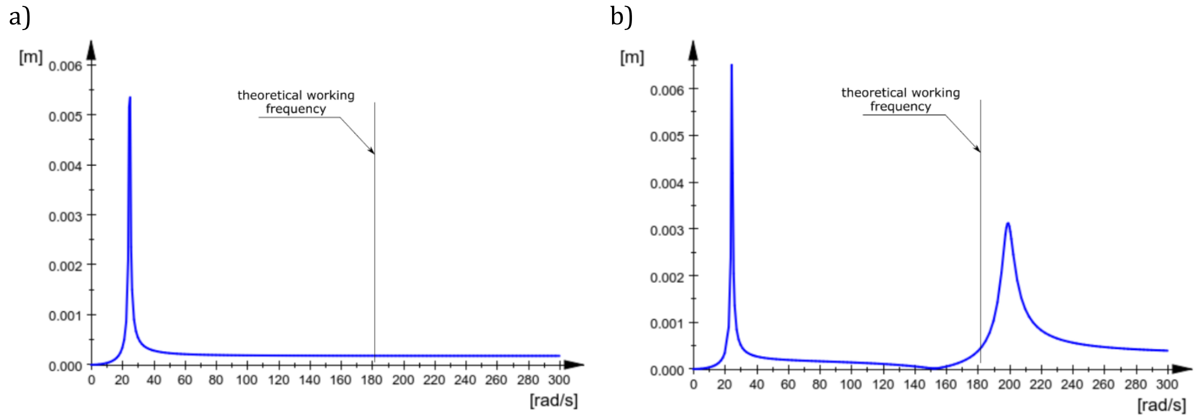

Figure 4. Amplitudes of the trough for reduced unbalance: a) in η direction; b) in ξ direction.

Analytical analysis was performed by applying simplified equations of motion to the MuPAD environment. The formulas enable analysis of the conveyor in steady state; it does not include equations for feed motion and its dumping. To obtain proper outputs for both considered directions of η and ξ, the system was built in form of two separate schemes (Fig. 3a, b) analyzed as two independently working models.

Figure 4a presents the amplitude of the trough in the η direction. It is seen that the amplitude, after passing through the first resonance remains constant at a very low value. In the perpendicular direction amplitudes of the trough arrange differently fig. 4b. The dynamic eliminator eliminates vibrations of the trough in its working direction ξ, which in turn results in fading of the vibrations in ξ direction, but not in η one. Notwithstanding the lack of vibrations in the anti-resonance zone in working direction of leaf springs, vibrations in η direction does not occur. The amplitudes are not high enough to cause transportation.

Going out the anti-resonance zone the system head towards leading slope of the second resonance. The difference between amplitudes in two perpendicular directions increases. This in turn results in increase of the transportation velocity value to the right. Such characteristic allows also to precise dosing of the feed by means of a slight change of the angular velocity. Stoppage of the transport can be achieved by decreasing excitation frequency from theoretical working frequency point to anti-resonance zone by about 20 rad/s, which means relatively short time (just as much as it needs for electro-vibrator to change the angular velocity of unbalanced mass, which due to a small unbalance of engine does not exceed in practice 0.5 s).

Theoretical working frequency point is common for the two considered directions of motion of the conveyor. This is because the presented model (Fig. 4) moves simultaneously in both η and ξ directions. Transportation either to the right or to the left occurs only if both of the following conditions are fulfilled: (i) throw coefficient K cannot be significantly lower than 1 and (ii) amplitude of the trough is greater in one direction than in perpendicular one. According to these rules there is possibility to transport in ξ direction by applying angular velocity in working range between 170 rad/s and 190 rad/s (before second resonance). Choosing angular velocity slightly above 180 rad/s as a working frequency point is also strategically suitable for further analysis of velocity of transportation for changing load of the feed.

Since the main problem of the study constituted revealing that the conveyor maintains constant transport velocity by itself regardless of the feed mass, the analytical analysis of the amplitudes of the system was conducted in dependence of amount of the feed placed on the trough. Fig. 5 represents results of behavior of the trough at four different loads. Since the feed is not connected to the trough and only has moments of contact, it is difficult to assess the amount of feed that should be added in order to determine the dependence of the trough amplitude on the excitation frequency. The authors used dependences determined analytically [15] and by simulation [16]. According to these studies, the mass of the trough is enlarged by the mass of the feed multiplied by the squared sinus of the inclination angle of the trough vibrations direction.

It is seen that a graphical representation for each weight is very similar to each other but differs in the maximum values of amplitudes reached in the second resonance, which also shifts these values to the left under greater amount of the feed. The analytical analysis does not include dumping in the feed. This in turn results in visible shifts in fig. 5 – for the constant value of the angular velocity the amplitude changes, what theoretically should increase velocity of the feed transport, with simultaneous gain of feed mass. However, in practice, when the additional dumping originated from dry friction in the feed is considered, the resonance amplitude is lowered for larger feed, and this causes that the feed can be transported at the constant velocity regardless of its amount without the need of changing the excitation frequency.

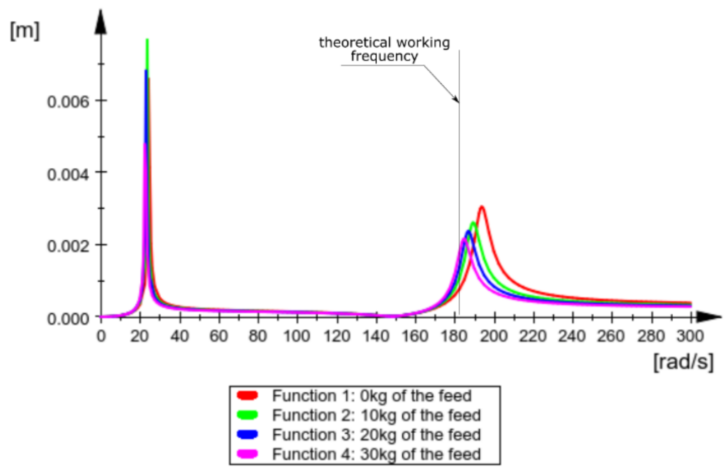

Figure 5. Amplitude of the trough in ξ direction in dependence of the excitation frequency (analytical analysis).

The mathematical model of the system contains the matrix equation describing the machine motion (1), equations concerning electromagnetic moment of the drive motor (10), equations used for determination of movements of successive feed layers (8,9) and dependencies describing normal (6) and tangent (7) influences between feed layers and between the feed layer and machine body. Due to the fact, that the model is five DOF system with additional four consecutive layers of the feed divided into five separate columns, the mathematical model consists of five equations of motion of the conveyor together with the two equations of motion of one particle of the feed. Thus, 40 equations concerning the feed are presented in the shortened scheme consisting of indices *j*, *k*.

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

$$\mathbf{M} = \begin{bmatrix} m_r + m_w + m_e & 0 & 0 & m_w e \sin\varphi & m_e \cos\beta \\ 0 & m_r + m_w + m_e & m_w e \cos\varphi & m_w e \cos\varphi & m_e \sin\beta \\ 0 & 0 & J_r + J_e & 0 & 0 \\ m_w e \sin\varphi & m_w e \cos\varphi & 0 & m_w e^2 + J_w & 0 \\ m_e \cos\beta & m_e \sin\beta & 0 & 0 & m_e \end{bmatrix} \tag{2}$$

$$\ddot{\mathbf{q}} = [\ddot{x} \quad \ddot{y} \quad \ddot{\alpha} \quad \ddot{\varphi} \quad \ddot{f}]^T \tag{3}$$

$$\mathbf{Q} = \begin{bmatrix} -m_w e\varphi^2 \cos\varphi - 2k_x(x + H\alpha) - 2b_x(\dot{x} + H\dot{\alpha}) - T_{101} - T_{102} - T_{103} - T_{104} - T_{105} \\ m_w e\varphi^2 \sin\varphi - k_y(y + l_1\alpha) - k_y(y - l_2\alpha) - b_y(\dot{y} + l_1\dot{\alpha}) - b_y(\dot{y} - l_2\dot{\alpha}) - F_{101} - F_{102} - F_{103} - F_{104} - F_{105} \\ -2k_x H^2\alpha - 2k_x Hx - 2b_x H\dot{x} - 2b_x H^2\dot{\alpha} - k_y(y + l_1\alpha)l_1 + k_y(y - l_2\alpha)l_2 - b_y(\dot{y} + l_1\dot{\alpha})l_1 + b_y(\dot{y} - l_2\dot{\alpha})l_2 + \\ (T_{101} + T_{102} + T_{103} + T_{104} + T_{105})h + F_{101}2d + F_{102}d - F_{104}d - F_{105}2d \\ M_{el} - b_f\dot{\varphi}\, sign\,\varphi \\ -k_s f - b_s\dot{f} \end{bmatrix} \tag{4}$$

where:
$F_{j,j-1,k}$ – normal component of j layer pressure on *j*-1 in k column,
$T_{j,j-j,k}$ – tangent component of j layer pressure on *j*-1 in k column,
j - indicator of the material layer, *j*=0 relates to the machine body,
k – indicator of the column of the material layer.

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

$$F_{j,j-1,k} = 0;\ T_{j,j-1,k} = 0 \text{ for } \eta_{j,k} \geq \eta_{j-1,k}. \tag{5}$$

In the opposite case, the contact force in the normal direction occurs between feed layers *j, k* and *j*-1*, k* (or – in case of the first layer – between the layer and trough), which model is [17]:

$$F_{j,j-1,k} = (\eta_{j-1,k} - \eta_{j,k})^p \cdot k \cdot \left\{1 - \tfrac{1-R^2}{2}\left[1 - \mathrm{sgn}(\eta_{j-1,k} - \eta_{j,k}) \cdot \mathrm{sgn}(\dot{\eta}_{j-1,k} - \dot{\eta}_{j,k)}\right]\right\}, \tag{6}$$

and originated from the friction force in the tangent direction:

$$T_{j,j-1,k} = -\mu F_{j,j-1,k}\mathrm{sgn}(\dot{\xi}_{j,k} - \dot{\xi}_{j-1,k}). \tag{7}$$

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

$$m_{n,j,k}\ddot{\xi} = T_{j,j-1,k} - T_{j+1,j,k}, \tag{8}$$

$$m_{nj,k}\ddot{\eta} = -m_{nj,k}g + F_{j,j-1,k} - F_{j+1,j,k}. \tag{9}$$

Kloss's equation (10) is used as far as mechanical characteristics of the asynchronous engine is concerned.

$$M_{eli} = \frac{2M_{ut}(\omega_{ss} - \dot{\varphi}_i)(\omega_{ss} - \omega_{ut})}{(\omega_{ss} - \omega_{ut})^2 + (\omega_{ss} - \dot{\varphi}_i)^2} \tag{10}$$

where: M_{eli} – electromagnetic moment developed by motor i, assumed in a form corresponding with the static characteristic of a motor, M_{ut} – stalling torque of drive motors, ω_{ss} – synchronous frequency of drive motors, ω_{ut} – frequency of stall of drive motors.

4. Simulation and experimental results

Simulation investigations, including dumping occurring in the feed, were performed for various masses of the feed. The dependence of the transport velocity [18, 19] in the function of the excitation frequency is presented in fig. 6a. The transport does not occur near the first resonant zone due to low excitation frequency. It is seen that there exists range of possible angular velocities that can be chosen, and when the one is fixed, constant transport velocity can be achieved for variable feed masses. According to simulation and analytical analysis performed, set of possible excitation frequencies allowing for this procedure starts at about 170 rad/s and ends at about 185 rad/s. In order to confirm the simulation analysis, the tests were performed on the test stand (fig. 1b). The laboratory tand consists of vibrating conveyor designed and built according to parameters shown in Tab. 1 in which rotational speed of the unbalanced mass was regulated by the inverter and controlled by a speedometer. The velocity of transport was measured on the distance *d* (fig. 1b) by measuring time. To facilitate the measurement, a feed of different color was used. In the case of the tested conveyor it maintains constant transportation velocity up to the load not exceeding 30 kg. For such a mass on the trough the yield of 22 tons/hour is obtained. At higher loads the transportation velocity collapses.

Table 1. Values of parameters simulations were performed for.

Symbol	Value	Unit	Description
$l_1 = l_2$	0.365	m	Distance between center of rotation to the axis of a spring
H	0	m	Distance between center of rotation to the spring position
$b_x = b_y$	58	Ns/m	Damping coefficient of the coil spring
$k_x = k_y$	32000	N/m	Stiffness coefficient of the coil spring
b_s	40	Ns/m	Damping coefficient of the system of leaf springs
k_s	937367	N/m	Stiffness coefficient of the system of leaf springs
m_e	42.5	kg	Mass of the eliminator
m_r	65	kg	Mass of the trough
m_w	6	kg	Unbalanced mass
m_n	0-30	Kg	Mass of the feed
J_e	3.7	$kg \cdot m^2$	Moment of inertia of the eliminator
J_r	4.82	$kg \cdot m^2$	Moment of inertia of the trough
J_w	0	$kg \cdot m^2$	Moment of inertia of the unbalanced mass
β	45	^{o}	Working angle of the leaf springs
R	0.05	unitless	coefficient of restitution of normal impulses at collisions
μ	0.4	unitless	Friction ratio
p	1	unitless	Herz-Sztajerman constant
k	10^8	N/m	Herz-Sztajerman constant
e	0.003	m	Unbalance position
m_{ut}	4.26	N·m	Maximum torque
b_f	0.00009	[Ns^2/m]	Coefficient of bearing resistance

During the tests performed on the experimental industrial research set-up it is very difficult to establish the dependence of the transport velocity on the excitation frequency for all velocities and masses determined in simulation studies. The experimental process is highly time-consuming and controlling the constant feed mass at long-lasting investigations is very difficult. Therefore, in order to confirm the correctness of simulation investigations an experiment was performed. During the experiment the velocity of transportation was measured for the constant excitation frequency in the working point while the mass of the feed on the trough was different. The result of these experiments is presented in fig. 6a and 6b. In the fig. 6b it is clearly seen that velocity of transportation does not change for changes in mass amount on the trough. The output of experimental tests fully confirmed a possibility of maintaining the constant transport velocity for various feed masses.

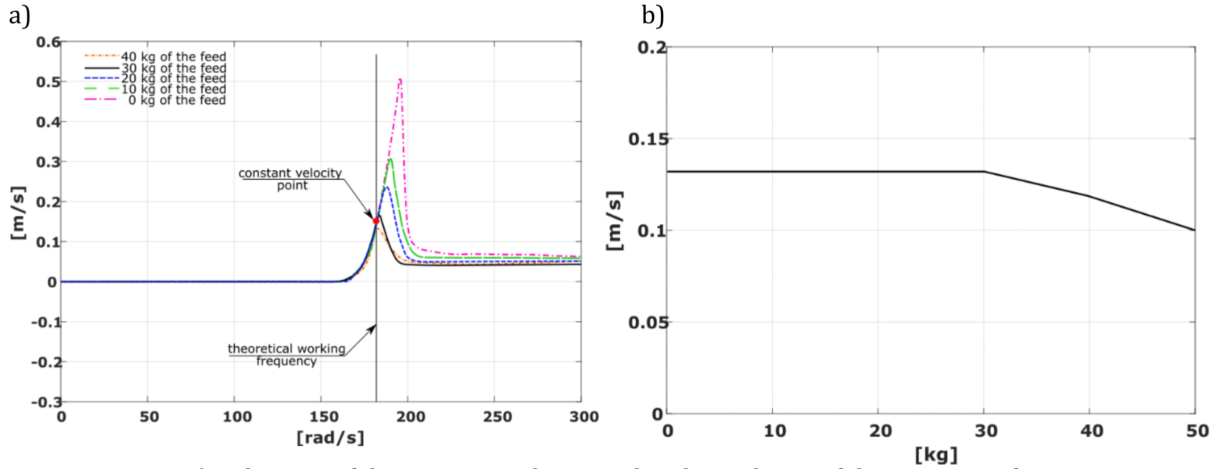

Figure 6. a) Velocities of the transported material in dependence of the excitation frequency (simulation results); b) Transportation velocity of the material on the trough in dependence of the mass (experimental results).

Recording that presents transport possibilities of the device is available to watch at the link: https://youtu.be/IUAouDx26Ls

4. Conclusions

On the bases of the performed analysis, simulation studies and experimental results several conclusions can be drawn. The conveyor indicates good transporting properties allowing to install it in production lines requiring the feed dosing.

1. The dosing is performed by slight changes of the excitation frequency without a necessity of the system passing through its resonance zones.
2. The conveyor is dosing a material regardless of the direction of the electro-vibrator rotations.
3. The device equipped with just one unbalanced motor maintains constant velocity of transportation regardless of the amount of the feed on the trough.
4. Work characteristic of the conveyor when inconsiderable unbalanced mass is applied seems more advantageous on account of amplitudes of eliminator.
5. Maximal amplitudes of eliminators are admissible on account of the material strength.
6. The tests of the transport velocity performed on the specially built device fully confirmed the working principle of the new dosing conveyor and their results were largely consistent with the results of the simulation tests.

Acknowledgments

The work is included in the framework of the Department of Mechanics and Vibroacoustics. 16.16.130.942

Additional information

The author(s) declare: no competing financial interests and that all material taken from other sources (including their own published works) is clearly cited and that appropriate permits are obtained.

References

1. W. Żmuda, P. Czubak; Patent pending No. P. 441218, 2022
2. M. Sturm; Design Optimization of Linear Vibratory Conveyors; PhD dissertation, Technical University of Liberec, Czech Republic, 2018
3. P. Czubak; Vibratory conveyor of the controlled transport velocity with the possibility of the reversal operations; Journal of Vibroengineering, 2016, 18(6), 3539–3547; DOI: 10.21595/jve.2016.17257
4. O. Fishman, T. Haines; Vibratory conveyor with side-mounted drivers; US Patent No. 6,041,915, 2000
5. W. Surówka, P. Czubak; Numerical review of selected solutions of vibratory feeders capable of dosing feed material; Vibrations in Physical Systems, 2020, 31(3), 2020322-1 – 2020322-8; DOI: 10.21008/j.0860-6897.2020.3.22

6. T. Zejer, M. Olesiński, K. Musioł, T. Okoń; Stopping Material Transport on Vibrating Feeder Chute; In: International Colloquium Dymamesi 2021 - Dynamics of Machines and Mechanical Systems with Interactions; Cracow, Poland, March 2 – 3, 2021
7. Z. Despotovic, Z. Stojiljkovic; PSPICE Simulation of Two-Mass Vibratory Conveying System with Electromagnetic Drive; In: EUROCON 2005 - The International Conference on "Computer as a Tool"; Belgrade, Serbia, 21-24 November 2005; IEEE, 1509-1512; DOI: 10.1109/EURCON.2005.1630251
8. I. I. Blekhman; Synchronization in science and technology; ASME press, 1988
9. J. Antoniak; Energy-saving belt conveyors installed in polish collieries; Transport Problems, 2010, 5(4), 5-14
10. P. Czubak, W. Surówka; Transport features of a new, self-attuned conveyor; Transport Problems, 2022, 17(4), 5-16
11. H. Frahm; Device for Damping Vibrations of Bodies; US 989958A, 1909.
12. C. Jiao, J. Liu, Q. Wang; Dynamic Analysis of Nonlinear Anti-Resonance Vibration Machine Based on General Finite Element Method; Advanced Materials Research, 2012, 443-444, 694 – 699; DOI: 10.4028/www.scientific.net/AMR.443-444.694
13. I. J. Sokolov, V. I. Babitsky, N. A. Halliwell; Autoresonant vibro-impact system with electromagnetic excitation; Journal of Sound and Vibration, 2007, 308(3-5) , 375-391; DOI: 10.1016/j.jsv.2007.04.010
14. J. Liu, Y. LI, J. LIU, X, H. Xu; Dynamical analysis and control of driving point anti-resonant vibrating machine based on amplitude stability; Chinese Journal of Mechanical Engineering, 1, 2006, 145-148
15. J. Michalczyk, G. Cieplok; Dobór wibratora maszyny wibracyjnej z uwzględnieniem zakłóceń wywołanych zderzeniami z nadawą (in Polish); Mechanika, 2000, 19(2), 221–232
16. P. Czubak; Reduction of forces transmitted to the foundation by the conveyor or feeder operating on the basis of the Frahm's eliminator, at a significant loading with feed; Archives of Mining, 2012, 57(4), 1121-1136
17. J. Michalczyk; Phenomenon of Force Impulse Restitution in Collision Modelling; Journal of Theoretical and Applied Mechanics, 2008, 46(4), 897-908
18. G. H. Lim; On the conveying velocity of a vibratory feeder; Computers & Structures, 1997, 62(1), 197-203
19. M.L. Chandravanshi, A.K. Mukhopadhyay; Dynamic analysis of vibratory feeder and their effect on feed particle speed on conveying surface; Measurement, 2017, 101, 145-156; DOI: 10.1016/j.measurement.2017.01.031

© 2023 by the Authors. Licensee Poznan University of Technology (Poznan, Poland). This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (http://creativecommons.org/licenses/by/4.0/).