Self-regulating Resonance Vibration Damping System

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Abstract

Vibrating conveyors are widely used in the industry for transportation of bulk materials. A special application of vibrating conveyors is dosing such materials precisely, which is often required by technological processes. The application of the conveyors driven by inertia vibrators for this purpose requires both the conveyor's operation close to its natural frequency and overcoming the resonance frequencies during the start-up and coasting of the device. Transitioning through the resonance zones and operating with near-resonance frequencies may cause uncontrolled spillage of the feed from the conveyor chute, which is the reason why such solutions have been thus far impossible to apply in industrial practice. The article describes the concept of the resonant vibration damping system designed to minimize the amplitude of vibration in transient states and to enable operation of the conveyor close to the resonant frequency. The basis of the presented solution is an automatic change in the amplitude of the damper's movement - small within the steady state of operation, and large within the near-resonant frequency.

Keywords: vibratory conveyor, resonance vibration damping

1. Introduction

Obtaining a precise dosing function from the vibrating conveyor driven by two self-synchronizing vibrators with opposing rotation requires the speed of material transport along the conveyor chute to decrease gradually, as the angular speed of the vibrators decreases, until transport stops. In addition, the transported material must not, in an uncontrolled manner, fall off the conveyor chute during its start-up and coasting.

The research shows that the conveyor frequency, at which material transport stops, is close to the frequency of the machine's own angular vibration.

Assuming the support of the conveyor on steel springs with low damping properties, high vibration amplitudes in the resonance zone and transient resonance when switching off and switching on the vibration drive should be taken into account.

It is therefore desirable to minimize the conveyor vibration in resonance zones. The vibration amplitude in the resonance zone can be reduced by increasing the damping in the suspension system. However, this cannot be done by a positive parallel viscous damper in the suspension because in the steady state it would emit too much energy, which would in turn cause both unnecessary losses and lead to dangerous heating of the damper.

The condition for obtaining the appropriate dynamic properties of the conveyor that allows for the precise dosing function is to increase the damping properties of the steel spring suspensions elements in the low operating frequency range, while limiting the damping in the nominal operating range. This task can be accomplished by using the system described in patent application P.425086 [1], shown in Figure 1.

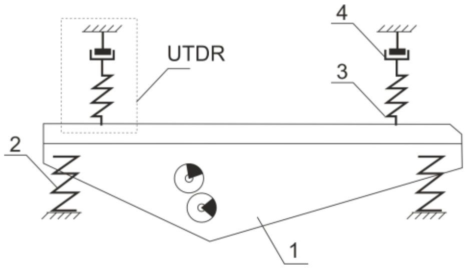

Figure 1. Conveyor with resonance vibration damping system (UTDR). 1-vibrating conveyor, 2-main suspension of the conveyor, UTDR - resonance vibration damping system, 3-UTDR spring, 4-UTDR damper

The resonance vibration damping system (UTDR) is made of a spring connected in line with a viscous damper and both parts connected in parallel to the main suspension of the conveyor. The resonance vibration damping system is attached to the mobile transport system with one end and to the stationary support structure of the device with the other. An appropriate selection of additional spring rigidity and the damper's suppression constant create a self-regulating system that dissipates energy when working at low frequencies and offers very low resistance in the operating frequency range.

2. Theoretical analysis of UTDR's functioning

Let's consider the steady-state operation of any suspension system from Fig. 1 (front or back end of a conveyor, suspended or supported), assuming that the machine's body is excited by a vibration drive, whose amplitude in the vertical direction changes harmonically over time according to the dependence:

$$y_o(t) = A\sin(\omega t) \tag{1}$$

where: $y_o(t)$ - vertical direction changes of the machine's body, A – vibration amplitude, ω – vibration frequency.

The dynamic equation of mass movement m_t can be written as:

$$m_t\ddot{y} = k_t(y_o - y) - b_t\dot{y} \tag{2}$$

where: m_t - movable internal mass of the viscous damper, k_t - elasticity coefficient of the damping system's spring, y_0 - vertical displacement of the machine's body, y - vertical displacement of the moving internal mass of the viscous damper (m_t), b_t - damper's constant.

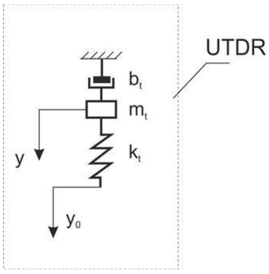

Figure 2. UTDR - resonance vibration damping system

Taking into account the formula (1) in this relationship, we obtain the equation as follows:

$$m_t\ddot{y} + b_t\dot{y} + k_t y = k_t A \sin(\omega t). \tag{3}$$

We seek the solution to this equation in the form:

$$y(t) = C\sin(\omega t + \gamma). \tag{4}$$

After substituting this form in equation (3) and dividing both sides by A, the ratio of vibration amplitude of damper C to the vibration amplitude of the machine in the vertical direction can be obtained:

$$\frac{C}{A} = \frac{1}{\sqrt{\left(1-\left(\frac{\omega}{\omega_{nt}}\right)^2\right)^2 + \left(\frac{b_t\omega}{k_t}\right)^2}}, \tag{5}$$

where

$$\omega_{nt} = \sqrt{\frac{k_t}{m_t}}. \tag{6}$$

Because the energy dissipated in the viscous damper is equal to:

$$N_t = \frac{1}{2} b_t \omega^2 C^2. \tag{7}$$

The reduction in the vibration amplitude of the damper C in relation to the amplitude A of machine vibration in the vertical direction for a given vibration frequency ω causes the reduction of power dissipated in the damper, according to the following relationship:

$$\frac{N_t(C)}{N_t(A)} = \left(\frac{C}{A}\right)^2. \tag{8}$$

The value of power dissipated in the damper can be written as a function of vertical vibration parameters, A and vibration frequency ω, in the formula:

$$N_t = \frac{1}{2} b_t \omega^2 A^2 \cdot \left(\frac{C}{A}\right)^2. \tag{9}$$

Shown below are the values of the ratio $(C/A)^2$, as a function of the machine vibration frequency f for a conveyor with the given parameters (total value from two support points at the beginning or end of the chute) of: b_t = 1256 Ns/m, m_t = 3kg, k_t = 49298 N/m:

Table 1. Dependence of the expression $(C/A)^2$ on the frequency of the conveyor.

f [Hz]	1	2.5	4	7.5	15	24
$(C/A)^2$	0.98	0.88	0.74	0.45	0.16	0.07

It can be concluded from the above, that the proposed system ensures practically full utilization of the damper in the range of natural frequencies f = 1 to 4 [Hz], at the same time reducing the generated power 10 times, protecting the system against unnecessary power loss and overheating of the damper. The method of selecting UTDR parameters is presented later in the article.

3. Model of vibrating conveyor with UTDR

Tests of the vibration damping system were carried out on a digital model of a vibration conveyor suspended on tendons, equipped with the resonance vibration damping system and a layered feed model.

This model was built on the basis of proven and tested simulation models of vibrating machines published in 2001-2017 by the Vibromechanics Team of the Department of Mechanics and Vibroacoustics at AGH University of Science and Technology in Kraków. Specifically, the machine model driven by two self-synchronizing electro-vibrators was based on work [2]. The model introduces a generalization, allowing different elastic properties of the elastic support system at both ends of the machine. The conveyor was loaded with multi-layer bulk feed model, the basis and experimental verification of which are presented in monograph [3]. The model of the elastic support system of the machine for a conveyor suspended on elastically supported tendons was based on the model presented in work [4].

The resulting conveyor model is shown in Fig. 3. The model has 7 degrees of free movement associated with the conveyor and 40 degrees of free movement associated with the layered feed model. The torque parameters of the motors are described by Kloss characteristics. Due to the limited volume of the article, the derived equations of motion are not quoted. Simulation tests were carried out for the following model parameters:

m=1518 [kg], J_0 =3803 [kgm^2], m_1=m_2=44.8 [kg], e_1=e_2=0.038 [m], J_{01}=J_{02}=0.101 [kgm^2], a_1= 1.135 [m], a_2=0.958 [m], h_1=0.449 [m], h_2=0.756 [m], l_1=1.594 [m], l_2=2.006 [m], h= 0.128 [m], H=(-)0.123 [m], k_1=300 000 [N/m], k_2=263 234 [N/m], k_y/k_x=4, b_{y1}=167 [Ns/m], b_{y2} =133 [Ns/m], b=100 [Ns/m] (for x), Σm_{nij}=500 [kg], μ=0.4, R=0.05 (restitution rate), d=1(feed pitch), M_{ut}=40.5 [Nm], $\omega_{ss}(t)$=157 – 0.2*t [rad/s], ω_{ut}=125 [rad/s], p=2.5, b_{s1}=b_{s2}=10^{-4} [Ns/m], d_1=d_2= 0.112 [m].

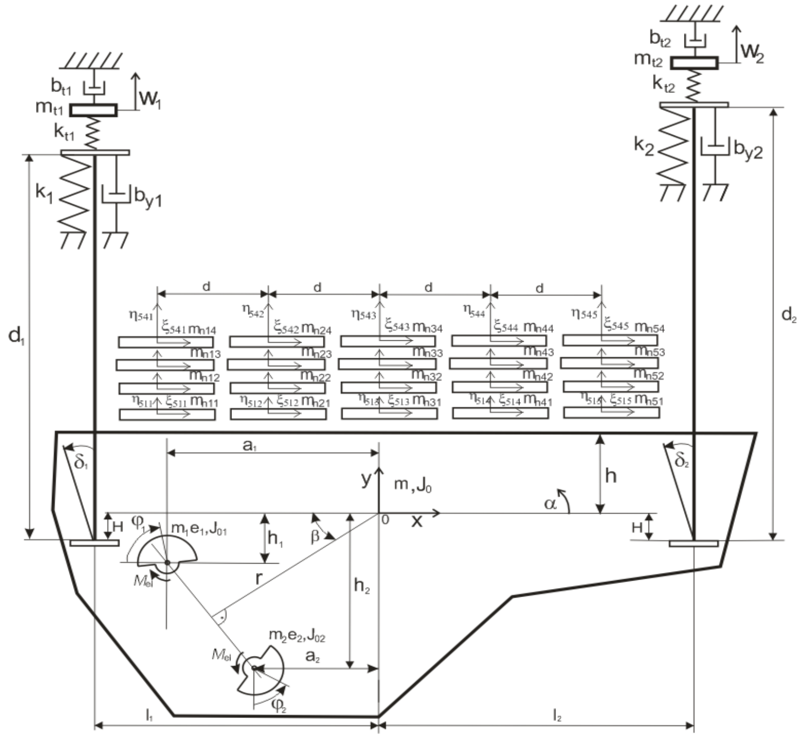

Figure 3. Model of vibrating conveyor

The parameters of the resonance vibration damping system were selected in the following procedure:

1) Due to the need to protect against spilling additional feed during transient resonance during coasting, the ratio of maximum resonance amplitude to the steady state amplitude was assumed to be 2.5, which requires [5] the value of the relative damping coefficient in the machine-damper ξ = 0.2. This value in the studied example corresponds to the values of permanent viscose dampers: b_{t1}=6 708 Ns/m b_{t2}=5 330 Ns/m.

2) For the given dampers, the total vibrating masses of the dampers were estimated together with the masses connected to each of the two supports adopted in the model as: m_{t1}=m_{t2}=3 kg.

3) The ratio of damper vibrations amplitude C to amplitude A of the machine vibrations (5) allows determining the total value of the constant kt of the damper springs' elasticity. The prerequisite of the value of this ratio adopted as C/A = 0.922 ensures 85 % utilization of energy dissipation by the damper for the frequency of vertical resonance vibrations and an 11-fold decrease in energy dissipation during nominal operation. As a consequence, according to the rule of equality of momentum from elastic forces relative to the centre of mass of the machine, this allows determining the elasticity constants k_{t1}=313 779 N/m, k_{t2}=249 454 N/m for both supports of the machine.

4. Numerical simulation results

In order to assess the effectiveness of the resonance vibration damping system, the following charts summarize selected results of simulation tests obtained for the system without UTDR (red) and with UTDR (green). Firstly, the impact of UTDR on the start-up process, nominal work for a period of 150 seconds and free coasting of the machine was checked.

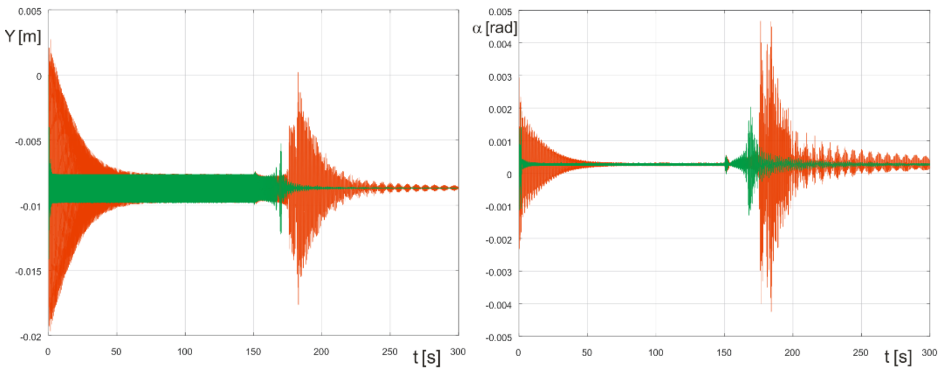

Figure 4. Impact of UTDR on vertical (left) and angular vibrations of the conveyor (right) during start-up, nominal operation and free coasting (with UTDR - green line, without UTDR - red line)

The application of the vibration damping system resulted in a reduction in the duration of transient periods in the vertical Y direction by approximately: 15x for start-up, 5x for coasting, and a decrease in maximum amplitudes: 1.6x for start-up, 2.5x for coasting. At the same time, for the α coordinate, it shortened the duration of the transition periods by: 6x for starting, 1.7x for coasting, and the reduction of maximum amplitudes: 1.8x for starting, 2.7x for coasting.

In the second case, the UTDR was examined in quasi-steady states. The simulations included switching on the conveyor and reaching the rated speed, nominal work for a period of 100 s, then working at a slowly changing, linearly decreasing power frequency until the transport of material over the conveyor chute stops, finally stopping the vibrators.

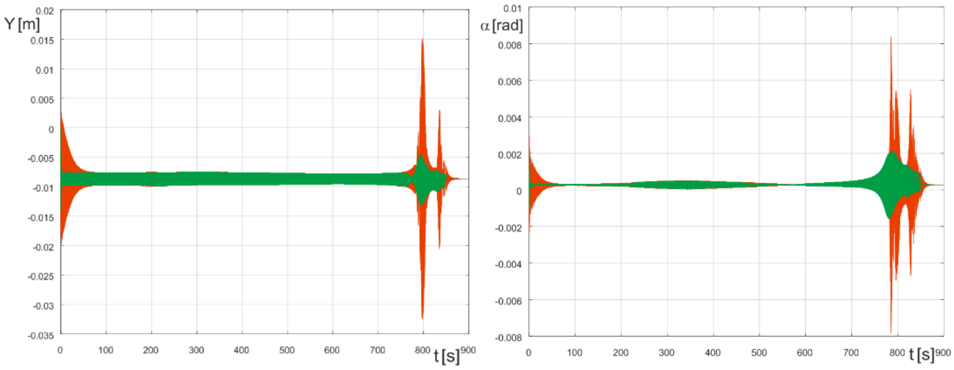

Figure 5. Impact of UTDR's operation on conveyor's vertical and angular vibrations during start-up, steady-state operation and quasi-steady coasting with linearly decreasing angular speed of vibrators. (with UTDR - green line, without UTDR - red line)

The application of the UTDR system resulted in a 5.8-fold reduction in the maximum vertical vibration amplitude in the quasi-stationary resonance and a 4.5-fold reduction in the maximum angular vibration amplitude in the quasi-stationary resonance.

The impact of UTDR on the feed transport process is shown in Figure 6. The change (decrease) due to the introduction of the resonance vibration damping system has affected the feed transport speed for operating frequencies lower than nominal, in direct proportion to lowering the rotational speed of vibrators, which, however, did not change the frequency of stopping the transport, which in both cases was 7.4 Hz (corresponding to the time instant of 600 s on the chart). The nominal transport speed is shown between 40th and 140th second of the simulation.

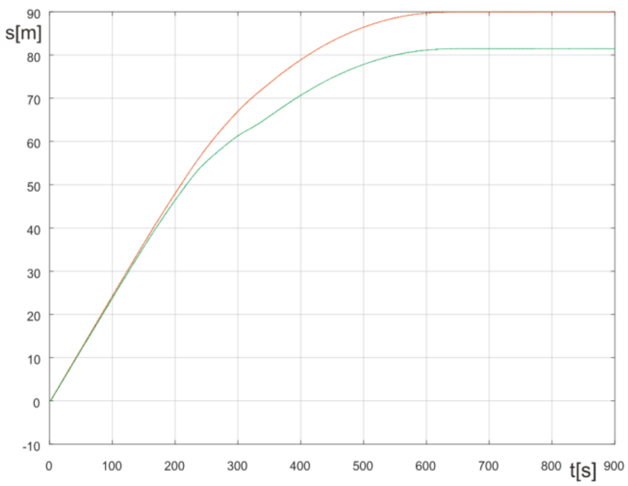

Figure 6. Feed transport along the conveyor with (green line) and without UTDR (red line).

5. Conclusions

a) The use of UTDR, whose principle characteristic is the automatic change in the amplitude of the damper's movement – low in the steady-state operation, and high at frequencies near resonance – allows for accurate dosing of the feed carried by conveyors with rotary vibrators by reducing the frequency of vibrators.

b) The analysis of this system carried out by means of computer simulation has proven the full usefulness of the UTDR system for the implementation of a precise feed dosing. In particular, the analysis showed that the system does not interfere with the conveyor operation at nominal conditions, but allows to limit to practical values the amplitude of vibrations in transient resonance states and during the material dosing process.

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