Optimizing the Design of Vibrating Conveyors with Two Masses

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Abstract: *The usual construction solutions for vibrating conveyors can generate intense dynamic stresses transmitted in the vibrating system and on the ground, especially for long transport lengths. Zero amplitude points can also occur, which in turn produce clumps of material, thus reducing the useful transport length. Some of the technical solutions to solve these shortcomings are non-harmonic vibrations, whose behavior can be approximated with sufficient accuracy by composite harmonic vibration systems with a construction with one or two vibrating masses. They can provide long transport lengths with low energy consumption and low dynamic loads.*

Keywords: *vibrating conveyors, non-harmonic vibrations*

1. INTRODUCTION

Vibrating conveyors operate on the principle of jump of the material to be transported.

A vibrating conveyor consists of a trough or tube, a suspension, which can be depending on its role, oscillating with a guiding role, or an elastic suspension for damping, as well as a vibration generator.

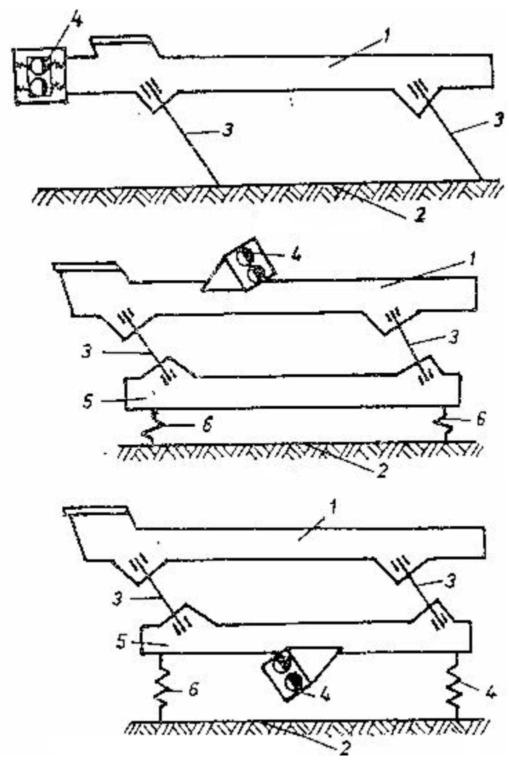

Fig. 1 Construction cases of vibrating conveyors.[4]

In Fig.1: 1 – the gutter, 2 – foundation, 3 – lamellar spring, 4 - unidirectional vibrating generator mechanism, 5 - balancing and/or damping table.

The actual construction of the vibrating conveyors can be with a single vibrating mass (Fig. 1 – case from the top) or with two vibrating masses (Fig. 1 – the two other cases). In the case of two-mass construction, there are constructions with only one driven mass (Fig. 1 – the middle case), or with both driven (Fig.1 – case from the bottom) [4].

The vibrating mechanism can consist of connecting rod-crank mechanisms, electromagnetic vibrators, or inertial vibrators. The vibrating mechanism can act on the rigid or elastic conveyor.

In self-balancing unidirectional vibrators with two rotating eccentric masses, the two eccentric masses rotate synchronously in opposite directions, generating harmonic unidirectional vibrations. The frequencies used are between 15 and 50 Hz (Fig.2).

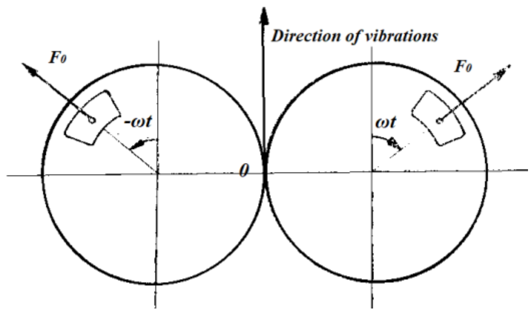

Fig. 2 Single directional inertial vibrator [6]

The electromagnetic vibrators used are of the reactive type. The vibration frequency is usually 100Hz. If a frequency of 50Hz is desired, rectifiers are used to eliminate alternation.

The rigid drive consists of a connecting rod-crank mechanism, having the amplitude equal to the radius of the eccentric. The elastic drive can be spring, inertia, or electromagnetic. The frequencies used are between 5 and 15 Hz.

The gutter suspension is in the form of packages of compression helical springs, lamellar springs, or rigid articulated bars, the latter also having the role of guiding the vibratory movement of the gutter. By means of the elastic support system, the parameters of the material transport vibration are also adjusted.

The constructive solutions for the operation of the conveyors are the generation of vibrations with inertial vibrator with self-balance assembled on the trough supported with helical spring packages, or generation with free inertial vibrator assembled on the trough supported with articulated rigid bars or lamellar springs.

Certain constructive solutions generate intense dynamic stresses transmitted throughout the vibrating system as well as on the ground. At the same time, for long transport lengths, zero amplitude points can appear, which in turn produce agglomerations of material, thus leading to a reduction in the useful transport length.

Technical solutions to solve these shortcomings are systems with non-harmonic vibrations mechanism (Fig.3). Another solution is that having a construction with two vibrating masses (Fig.1). They can provide long transport lengths with low energy consumption and low dynamic loads. The main disadvantage is the more complicated construction.

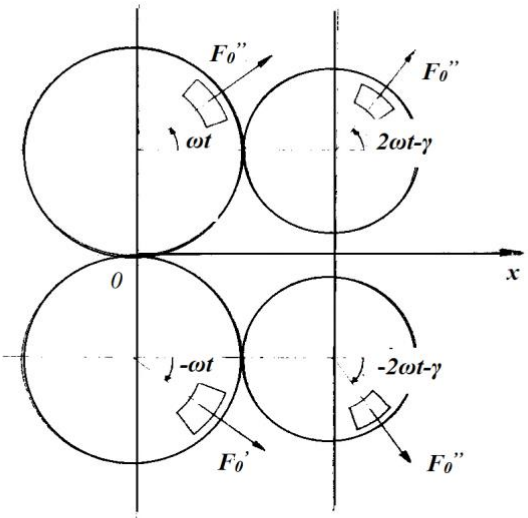

Fig.3 Bi-harmonic unidirectional vibrator. [3]

2. METHODS FOR STUDIES

Although there are types of stochastic models for simulating rather random processes, they do not currently apply to vibrations [7].

Vibrational processes are often difficult to translate into precise mathematical relationships that accurately model the phenomena that occur during oscillations. This is especially true for production processes that involve the transport of materials and products, due to the existence of a multitude of random factors that cannot be quantified. Therefore, simplifying hypotheses are generated, which subsequently generate models by similarity with relationships already verified in practice [2].

Vibrations can be harmonic or non-harmonic.

The main operating parameters of vibrating conveyors, such as amplitude, frequency, speed of material advance, power, depend not only on the size of the vibrating mechanism, but also on the inertial forces that appear in the system and the resistance to advance, such as friction in joints [1].

The actuation of the vibrating conveyors is largely based on the direct harmonic actuation of the transport chute.

However, transport equipment operated by non-harmonic methods also appeared. In this case, two trends are addressed:

- Increasing the transport speed while keeping the transport coefficient K constant.

- Decreasing the amplitude while keeping the transport speed constant.

Since the theoretical law of non-harmonic oscillations cannot be represented in practice, it was found that non-harmonic oscillations in reality can be modeled by mathematical relations obtained from the interference of two harmonic oscillations.

2.1 Increasing the Transport Speed with the Transport Coefficient K Constant

Generaly, vibrating machines work optimally when the maximum transport efficiency is obtained at a minimum power consumption, at the lowest possible dynamic speeds. It has been found from operating practice and experiments that these regimes correspond only to certain pairs of amplitude / frequency values. Thus, low amplitude values are required for high frequencies and vice versa, high amplitudes for low frequencies. [7].

In the case of conveyors, it has been found that the transport speed increases optimally for high amplitudes and low frequencies. Also, the transport speed depends on the type of drive. [6].

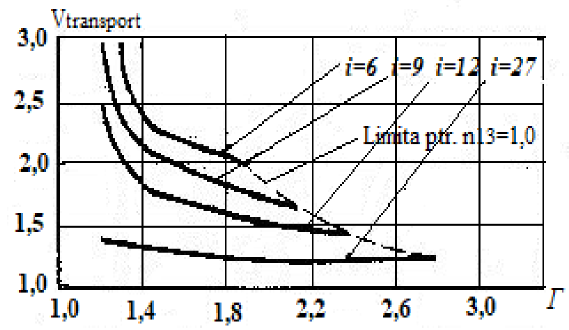

Fig. 3 Nomogram for determining the increase of the transport speed due to the action nonharmonic oscillations ($k = 3$, $\phi_3 = 180°$, $i = 45$). [5]

In this situation there is the expression:

$$S_R = A_1 \cdot \left[\cos \omega\, t + \frac{1}{i \cdot \cos(k \cdot \omega t + \phi_k)} \right] \quad (1)$$

In which:

- S_R – the displacement
- A_1 - amplitude of the basic oscillation.
- i - amplitude ratio ($i = A_1 / A_k$).
- A_k - amplitude of (k-1) oscillations.
- k - frequency ratio to (k-1) basic oscillation.
- ϕ_k - the angle of adjustment of the phases (k-1) oscillations compared to the basic oscillation.

Nomograms such as the one in Fig.4 can be used to calculate the motion parameters.

The nomogram represents the relations between the throwing coefficient $\boldsymbol{\Gamma}$, the coefficient of resistance to advance $\overline{\boldsymbol{n}}$ and the ratio $\boldsymbol{i}$ between the amplitudes of the two harmonic movements that compose the resulting vibration movement. The nomogram also includes the acceptable limits for calculating the motion.

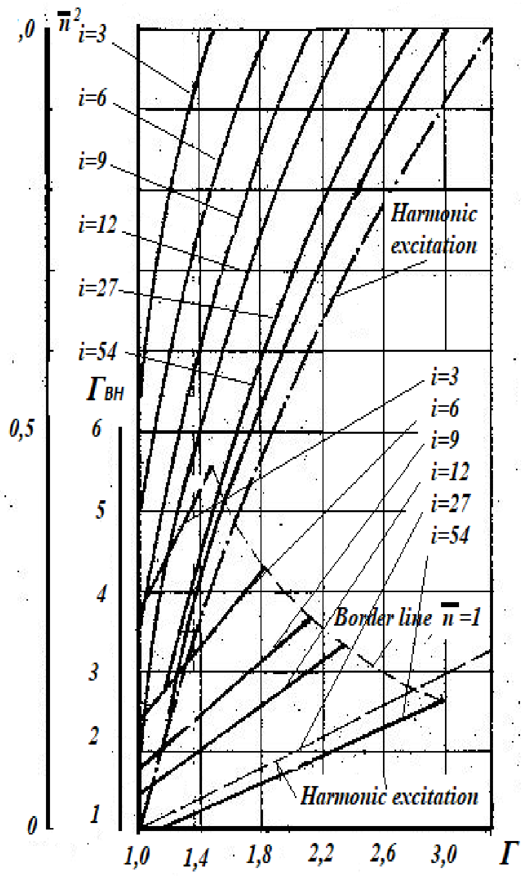

Fig.4 Nomogram for determining the increase of the transport speed of the drive using nonharmonic oscillations (k = 3, ϕ_3 = 180). [5]

The resulting drive is produced by two harmonic movements. Both systems have equal amplitudes.

In Fig.4, parameters are:

- ***Γ*** - throwing coefficient at harmonic drive.
- ***Γ***

Using the nomogram in Fig.4 were determined for an amplification ratio of i = 9 and a throw coefficient Γ = 1.37, and forward resistance coefficient n = 0.5, the following values for the main parameters of the movement:

- $\Gamma_{BH} = 1{,}37$
- $\bar{n}^2 = 0{,}25$
- $\bar{n}^2_{BH} = 0{,}542$
- $\bar{n}_{BH} = 0{,}743$
- $V_{BH} = 2{,}17$,

in which V_{BH} - transport speed increase ratio, with the expression: $V_{BH} = V_{transport\ BH}/V_{transport}$.

2.2 Decreasing The Amplitude with the Transport Speed Constant

For the second case of using vibrating conveyors, that is decreasing the amplitude while keeping the transport speed constant.

In Fig. 5 one can find the actual decrease of the amplitude for the action by interference of two harmonic oscillations A_{BH} compared to the harmonic actuation, both systems having equal transport speeds.

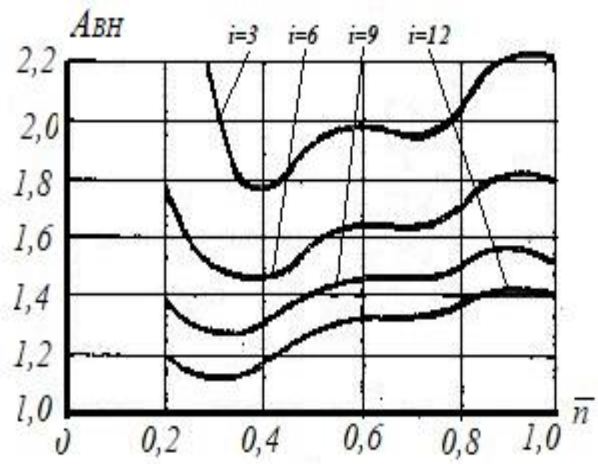

Fig.5 Decreasing the amplitude for the action by interference of two harmonic oscillations (k = 3, ϕ_3 = 180°) compared to the harmonic actuation, both systems having equal transport speeds. [5]

3. CONCLUSIONS

Due to the large number of factors that influence the process and its complexity, it is difficult to determine exact mathematical relationships that accurately describe the transport process itself.

That is why, in a first phase, correction coefficients are introduced to adapt the relations to a situation as close as possible to the real one.

On the other hand, in a successive phase, experiments are performed which generate nomograms that include links between the optimal / critical values of the main parameters that ensure the correct operation of the conveyor.

In this way, optimal operating regimes for the machines can be determined quickly and safely.

In the case presented in this paper, two solutions were shown to reduce the dynamic stresses in the vibrating system by increasing the speed but keeping the coefficient of addition constant, respectively by reducing the amplitudes but keeping the speed constant.

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