On the possibility of increasing the range of movement of rocks with the help of vibration transport

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Abstract. The results of the experimental studies of the dynamics of the elastic working body of a vibratory conveying device equipped with two inertial vibration exciters are presented. The nature of the change in the mismatch of vibration source rotary frequencies and the degree of their mutual influence with an increase in the linear weight of bulk material has been established. The basic relationships of the structural and dynamic parameters of the vibrating device have been determined, which make it possible to increase the movement range of loose geomaterials.

1. Introduction
The staff of Chinakal Institute of Mining, SB RAS, have been solving the problems of producing and moving various bulk materials extracted and processed at mining enterprises for a long time [1, 2]. Among the mechanical means used for these purposes, vibratory conveying machines occupy a special place due to their design simplicity, low metal consumption, and operational reliability. A number of vibration devices designed to operate not only with bulk materials, but also difficult to transport materials (sticky, caking, etc.) have been developed in the vibrotechnology laboratory of the Institute of Mining, SB RAS. The design feature of these machines lies in the low bending stiffness of the working body which is not fixed to the frame with the help of elastic supporting elements, as in most vibratory conveying devices [3, 4], but freely fits on the base [5–8]. Unbalance vibration exciters of domestic manufacturers (for example, the IV vibrators of Krasniy Oktyabr PJSC) and imported devices (for example, the MVE vibrators of the Italian company OLI) are used as a source of vibrations for such carrying surface. Under the action of its circular driving force, the elastic working body makes wave motion and can move bulk mass in various modes, depending on its properties. In addition, devices with an elastic working body are characterized by low metal consumption and ease of installation.

However, when working under blockage, there is some attenuation of vibrations transmitted along the load-carrying surface from the section for fixing the vibration source, which limits the distance of transporting rock mass. This problem can be solved by changing device design [9] or increasing the power of the vibration drive. The use of the latter method with bending vibrations is ineffective, since it can lead to the appearance of additional stress concentrators in the working body and reduce its durability. It is preferable to disperse driving force and equalize the vibration field along the load-carrying surface by using several less powerful vibration sources.

A necessary condition for the operability and efficiency of most vibration machines with several vibration exciters is rotation synchronization, and sometimes the presence of certain relationships.
between individual vibration exciters rotational phases, for example, the presence of in-phase or antiphase [10]. The mechanical methods of obtaining the required operating mode of vibration sources significantly complicate the design of the vibration device. Instead of using additional mechanical connections and the structural changes of the vibration device, the ability of inertial vibration sources to self-synchronize can be used [10–13]. However, as the results of experimental studies show, bulk material moving along the vibrating surface has a significant effect on the self-synchronization process. Moreover, in contrast to laboratory conditions, in which the storage tank is filled in a certain way [14, 15], in real production conditions, the filling of the storage tank is less orderly, which leads to the uneven loading of the load-carrying surface of a discharging and transporting device. In this regard, obtaining the results of physical modeling performed on real bulk materials under conditions close to production conditions makes it possible to refine theoretical knowledge about the process of the self-synchronization of inertial vibration exciters and is an urgent task for researchers.

In [16] it was noted that the main parameter that determines the operation of vibration exciters in stable synchronous mode is the mismatch of their partial frequencies \( \Delta f \). In turn, it depends on the ratio of the distance between the vibration exciters to the length of the flexural wave transmitted between the sections of their attachment, as well as on the ratio of the magnitudes of the vibration amplitudes created by each vibration source. In this case, it is necessary to establish a rational distance between the vibration exciters, sufficient for their mutual influence on each other, and the ratio of the linear weight of bulk material to driving force, at which the synchronous rotary mode of vibration source rotors is stable.

2. Methods of the research
The studies were carried out at the stand (Figure 1) which includes a vibration device model located inside Storage Tank 2.

![Figure 1. Scheme of the test stand: 1 – elastic working body; 2 – support frame; 3 – vibration exciter; 4 – storage tank; B1, B2, B3 – options for fixing vibration exciters on the unloading, central, and loading sections of the working body, respectively.](image)

As a working body 1, metal sheets with a bending stiffness of 154 N\( \times \)m\(^2\) and 875 N\( \times \)m\(^2\) were alternately used. Its vibrations were created by 2 inertial vibration exciters 3 of the RZHF 40 type, generating circular driving force the maximum value \( P_d \) of which for each was 4.0 kN. One of the
vibration exciters was installed on the unloading section of the working body closer to the outlet window of the hopper (Position B1), and the second was alternately fixed on its central (Position B2) or loading (Position B3) section with a corresponding change in the distance between the vibration exciters \( l \) from 0.34 m to 0.85 m. With the help of the electronic frequency converters of the F1500-G series, unbalance rotation frequency was changed in the range of 43–49 Hz with a step of 0.1 Hz.

The experiments were carried out using sandy loam with clay content of up to 10%. The mass of the measured volume of the material in the storage tank was taken equal to 150, 250, 350, 450, or 550 kg, with a corresponding change in the linear weight \( q = 1.23, 2.04, 2.86, 3.68, \) or 4.46 kN/m.

The vibration sources were started after loading Storage Tank 4 with bulk material in sequence, starting with the vibration exciter in Position B1. In order to maintain constant pressure on the conveying surface, the hopper outlet window remained closed throughout the experiment.

The measuring complex included 6 sensors installed in normal direction to the working body surface at an equal distance from each other along its longitudinal axis. With their help, the speed of transverse vibrations created both by each vibration exciter separately and by their joint action was simultaneously recorded. This made it possible to determine a change in amplitude and the length of the flexural wave transmitted along the load-carrying surface with an increase in pressure from bulk material and the effect of these changes on the mismatch of partial vibration source frequencies.

The mutual influence of vibration exciters was numerically estimated by the ratio of the amplitudes of the oscillations created by them \( A_1/A_2 \), where \( A_1, A_2 \) are oscillation amplitudes in the areas of the fixation of vibration sources B1, B2 (or B3), respectively, as well as by the coefficients:

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k_{12} = \frac{A_1}{a_{12}}, \quad k_{21} = \frac{A_2}{a_{21}},
\]

where \( A_1 \) is the span of the transverse component of vibration displacement at the section where the vibration source is fixed in Position B1 when only this vibration exciter is operating; \( a_{12} \) is the range of oscillations in the area under consideration, created by the driving force of the vibration exciter installed in Position B2 or B3; \( A_2 \) is the span of the transverse component of vibration displacement at the section where the vibration source is fixed in Position B2 or B3 when only this vibration exciter is operating; \( a_{21} \) is the range of oscillations created in the area under consideration only by the driving force of the vibration exciter installed in Position B1.

3. Results of the research

The results of studying the dynamics of the working body with a stiffness of 875 N×m\(^2\) showed that the flexural wave generated by each vibration exciter has a length exceeding that of the working body. In this case, the vibrations of the load-carrying surface of the device are practically the same as those of a rigid body. Therefore, the vibration exciter fixed in Position B2 (\( l = 0.34 \) m) has a greater load from the working body and the bulk material located on it than on the vibration exciter in Position B3 (\( l = 0.84 \) m). As a result, despite the closer arrangement of the vibration sources, the ratio \( A_1/A_2 \) is greater (Figure 2, Curves 1,a and 1,b), and the degree of influence of the vibration sources on each other is less. In this case, the stability of the synchronous mode of their operation decreases. If the second vibration exciter is fixed in Position B3, then there is an increase in the damping of vibrations transmitted from B3 to B1 and vice versa, which also reduces the degree of the influence of the vibration exciters on each other.

With a decrease in the stiffness of the working body to 154 N×m\(^2\), the operation of each vibration exciter is accompanied by a bending of its fastening section and an increase in oscillation amplitude in the section of driving force application. As a result, the ratio \( A_1/A_2 \) decreases (Figure 2, Curves 2,a and 2,b). Moreover, if the linear weight of the bulk material \( q \) does not exceed 0.7 of the driving force amplitude value \( P_1 \), the vibration exciters operate under practically equal conditions and the ratio of the amplitudes of the oscillations they create is equal to unity, regardless of the distance between the vibration sources, which favorably affects the degree of their mutual influence.
However, with an increase in bulk material mass, the damping of oscillations transmitted along the working body increases.

![Figure 2](image2.png)

**Figure 2.** Dependence of the oscillation amplitude ratio in vibration source fixation areas on the ratio of the linear bulk material weight $q$ to the driving force amplitude $P_d$: working body rigidity: $I$ – 875 N×m², 2 – 154 N×m²; distance between vibration exciters: $a$ – 0.34 m; $b$ – 0.85 m

It was found that in the used frequency range (43–49 Hz) the effect of the vibration exciter located farther from the outlet hopper window (Position $B_2$ or $B_3$) on the operation of the vibration source $B_1$ is greater than the effect of $B_1$ on $B_2$ ($B_3$), regardless of the distance between them. As obtained dependences show (Figure 3), the values of the coefficients $k_{12}$ (Figure 3, Curves $I_a$ and $I_b$) are close.
to each other, less than the values of $k_{21}$ (Figure 3, Curves 2,a and 2,b) and practically do not depend on the size of material mass in the storage tank.

With an increase in the material mass at which the ratio $q/P_A$ reaches 0.6–0.7, the damping of oscillations transmitted along the elastic working body is proportional to the decrease in amplitude in the areas of driving force application, therefore, the influence coefficients $k_{21}$ change little. Under these conditions, doubling the distance between the vibration exciters leads to a fourfold decrease in the degree of influence of the vibration exciter $B1$ on the second vibration source (Figure 3, Curves 2,a and 2,b).

Therefore, at a distance between the vibration exciters of 0.34 m, a twofold increase in linear load on the vibrating working body led to a decrease in the value of partial frequencies mismatch by 40% (Figure 4, Curve 1), and with the removal of the vibration exciters more than twice, $\Delta f$ tends to zero (Figure 4, Curve 2). In the latter case, synchronous mode, if attained, was unstable.

![Figure 4. Dependence of partial frequencies mismatch ($f_1 - f_2$) at which synchronous mode is maintained, on the ratio of the linear bulk material weight $q$ to the driving force amplitude $P_A$ : distance between the vibration exciters: 1 – 0.34 m; 2 – 0.85 m](image)

As a result of experimental studies, the most stable synchronous operation of the vibration exciters was achieved when they were installed on the working body with a rigidity of 154 N×m² at a distance of 0.34 m from each other. Taking into account the flexural wave length $l_B$ also recorded experimentally, these parameters provide the ratio $l/l_B = 0.5–0.6$ which can be considered decisive in the design of a vibratory conveying device with an extended working body and several vibration exciters.

4. Conclusions

1. Due to design reliability and simplicity, vibratory conveying machines are effectively used to move bulk materials in the harsh conditions of mining enterprises.

2. In order to increase haulage, machines with an elastic working body can be equipped with 2 or more inertial vibration exciters of low power, operating synchronously due to self-synchronization effect.

3. The driving force of each vibration source is determined from the rational ratio with the linear bulk material weight $q/P_A = 0.6–0.7$.

4. The distance between vibration exciters should be set taking into account the length of the flexural wave formed in the elastic working body during device operation, based on the ratio $l/l_B = 0.5–0.6$. 

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