Ensuring self-synchronization of unbalanced vibration exciters for transveyers of overburden spreaders

SYa Levenson* and EG Kulikova**
Chinakal Institute of Mining, Siberian Branch, Russian Academy of Sciences, Novosibirsk, Russia
E-mail: *lev@misd.ru; **shevchyk@ngs.ru

Abstract. Deep-level mining in open pits involves increased volumes of overburden. It is difficult to operate heavy-duty dump trucks at bottom levels of deep open pit mines due to limited size of working areas and high angles of roads. In this case, it is expedient to use more maneuverable crawler dumps or articulated dump trucks of comparatively low capacity and to rehandle mined rocks to larger capacity dump truck at bin trestles with forced discharge constructed on accumulation horizons. This article discusses mined rock discharge from an accumulating bin of the rehandling point by a set of vibratory feeders representing the bottom of the bin. These vibratory feeders are to provide reliable operation under high static and dynamic loads.

1. Introduction
Dump forming is an important stage in mining flow process. Dump are most often formed using heavy dozers and shovels. These machines operate in the zones of dump slope failure and, thus, lack safety [1–5].

![Figure 1. Hydraulic vibratory spreader: 1—load carrier; 2—vibrating transveyers; 3 and 4—hydraulic cylinders; 5—support frame.](image)

The Institute of Mining, SB RAS designed a self-propelling hydraulic vibratory spreader (Figure 1) [6–8] meant for team operation with dump trucks and preventing fall of heavy machines from...
dumps. The spread has load-carrier 1 with several transveyers 2 mounted via hydraulic cylinders on support frame 5.

With the help of the travel mechanism, the spreader is set at the edge of a dump level on a flat horizontal site. A dump truck unloads granular material to load carrier 1 which then swivels and moves overburden down the dump slope [9, 10].

As vibrating transveyers 2, it was decided to use vibratory feed belts designed at the Institute of Mining, SB RAS [11–13]. These feeders represent metal sheets of low bending stiffness, freely placed on the understructure and making undulatory movement under the action of driving force from the inertia vibration exciter.

There are no elastic connections between the load carrier and base frame; for this reason, the capacity of these machines is only limited by the mechanical strength of the base frame. This enables using the vibro-belts in the conditions of dynamic impact of rock flow unloaded from dump trumps on the transveyer surface. Assembling is simple and takes no much labor input. Furthermore, the elastic feeders can implement various regimes of vibration transport.

As recommended [1], the vibration exciter of the vibro-belt is set at the unloading edge of the transneyer at a distance equal to one third its length. For prevention of damping of vibrations transferred along the transveyer surface under overburden flow, it was decided to use one more vibration exciter as shown in Figure 1. In this case, uniform vibration field along the transveyer can be maintained owing to self-synchronization of the inertia vibration exciters.

The theoretical research findings on self-synchronization are amply described by Blekhman, Yaroshevich and other scientists [15–19]. These data and experimental relations obtained on physical models at the Vibrotechnique Laboratory of the Institute of Mining made it possible to engineer a series of dual drive vibrating transportation facilities, e.g. VTU-6 [20]. In the meanwhile, stable synchronous operation of vibration exciter arranged at a long elastic feeder is yet a challenge.

In connection with this, the influence of the planting method of the transveyer on the frame and its wave motion on disagreement of partial frequencies and stability of synchronous operation of vibration exciters was tested experimentally.

2. Modeling and test results
The tests used the physical model of the vibratory transveyer shown in Figure 2. The conveying surface had a bending stiffness $EI = 154 \text{ N}\cdot\text{m}^2$ and a weight of $m = 9 \text{ kg/m}$. The frame structure allowed varying the incline of the transveyer from 0 to 15 deg. Aimed to limit the longitudinal displacement of the conveying surface, replaceable arrester with stiffness of 1.33 or 8.00 MN/m was mounted on the frame.

Vibrations were generated by two inertia vibration exciters RZHF 40 placed in the unloading section (position B1) of the conveying belt and in the loading section (position B2). The static moment of eccentric masses of each vibration exciters was 0.005 kg-m. The rotational velocity of B1 was varied as 49, 36 and 21 Hz and the rotational velocity of B2 was changed relative these values by electronic frequency converters at a step of 0.1 Hz. The exciters rotate clockwise as per the view in Figure 2.

![Figure 2. Vibrating feeder: 1—conveying surface; 2—elastic bottom layer; 3—base frame; 4—vibration exciters B1 and B2; 5—elastic arrester.](image)
The analysis focused on operation of the empty elastic transveyer. The vibration pickups set along the longitudinal axis of the feeder measured vibration velocity components in longitudinal and normal direction to the conveying surface.

With a view to examine transition of vibrations between two operating exciters and their self-synchronization potential, the experimentation assumed the procedure as follows:

Stage 1: Actuation of B1 (Figure 2);
Stage 2: Actuation of B2 15–20 s later and concurrent operation of the exciters for 40–70 s;
Stage: Shutdown of B1 while B2 operates for 10–15 s more.

The vibration velocity was picked up continuously within the whole operating period of the system.

As a result, at Stages 1 and 2, parameters of the elastic wave transferred from the operating exciter to the installation place of the non-operating exciter were determined at different frequencies of the driving force.

Each wave length $l_w$ depends on design parameters of the transveyer and, first of all, on its bending stiffness $EI$ [21]:

$$l_w = \frac{10.6 \cdot \sqrt{\frac{E}{\omega^2 g \cdot a}}}{\sqrt{\frac{m}{a}}}$$

where $g$ is the acceleration of gravity; $a$ is the amplitude displacement in the application domain of the driving force at the circular frequency $\omega$.

**Figure 3.** Distribution of vibration displacement along vibrating surface: 1, 2 and 3, 4—distribution of vibration displacement along vibrating surface during operation of one of vibration exciter, B1 or B2; points of measurements $P_{B1}$ and $P_{B2}$ are at the vibration exciters B1 and B2, respectively; measurement points $T_1$–$T_4$—along the vibrating surface; vibration frequencies: (a) 49 Hz; (b) 21 Hz.
The intensity of the transferred cross vibrations, alongside with the described characteristics, undergoes effect of the conveying belt friction on the underlayer and the ratio of its mass to the mass of the non-operating vibration exciter. Figure 3 presents the curve of the vibration vibration displacement along the vibrating surface at frequencies of 49 and 21 Hz at certain times. Curves 1, 2, 3 and 4 differ by half oscillation period.

The curves show that as with increasing frequency of the driving force, the wave length and the ration of vibro-displacement values at the installations of the operating and non-operating exciters decrease.

The research of the joint operation of the vibration exciters finds that the decrease in the damping of vibrations transferred along the conveying surface results in the higher influence of the first vibration exciter on rotation of eccentric masses of the second vibration exciter. For estimating degree of this influence, we introduced the coefficients to characterize ratio of the vibration amplitude \(a_i\) in the area of the operating exciter to the amplitude \(a_j\) transferred to this are by elastic wave from the other exciter:

\[
k_{12} = \frac{a_1}{a_{12}}, \quad k_{21} = \frac{a_2}{a_{21}}
\]

where \(a_i\) and \(a_j\) are the amplitudes at the points B1 and B2, respectively; \(a_{12}\) and \(a_{21}\) are the vibro-displacement transferred from B2 to B1 or from B1 to B2, respectively.

It is experimentally found that the change in the ratio of the vibration amplitudes near the vibration excites, i.e. in the coefficients \(k_{12}\) and \(k_{21}\), affects disagreement of the partial frequencies assumed as allowable \(\Delta f\) in case of the synchronous operation of the vibration exciters as an unallowable when wobbling arouse. The values of \(k_{12}\), \(k_{21}\) and \(\Delta f\) obtained for the systems with elastic elements of different stiffness, inclination 15° and vibration frequency are given in table 1.

| Arrester stiffness, MN/m | Vibration exciter frequency \(f_1\), Hz | \(k_{12}\) | \(k_{21}\) | \(\frac{a_1}{a_2}\) | Relation of frequencies of vibration exciters, \(f_1\) and \(f_2\) | Allowable frequency disagreement \(\Delta f\), Hz |
|-------------------------|----------------------------------------|--------|--------|-----------------|-----------------------------------------------|-----------------------------|
| 1.33                    | 49                                     | 2.3    | 8.0    | 0.8             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.7                         |
|                         | 43                                     | 1.6    | 4.3    | 1.0             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.4                         |
|                         | 36                                     | 10.8   | 3.2    | 1.5             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.1                         |
| 8.00                    | 49                                     | 2.1    | 9.0    | 0.7             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.6                         |
|                         | 43                                     | 1.6    | 3.5    | 0.9             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.4                         |
|                         | 36                                     | 8.0    | 3.0    | 1.7             | \(f_1>f_2\) \(f_1<f_2\)                       | 0.2                         |

It follows from table 1 that with shorter wave generated by an exciter, the ratio \(\frac{a_1}{a_2}\) is low and the exciters have higher influence on each other, which raises \(\Delta f\). In this case, the synchronous operation of the exciters becomes more stable.

It was assumed that the change in parameters of longitudinal vibrations of the transvayer through the increase in its inclination angle and stiffness would have effect of the process of self-synchronization. However, the tests showed that owing to undulation of the conveying surface, the
longitudinal vibrations transferred to the contact with the elastic arrester is insufficient to cause any response from it. The increase in the arrester stiffness more than by 6 times and in its incline from 0\(^\circ\) to 15\(^\circ\) had observable influence neither on synchronization nor on allowable disagreement of frequencies of the vibration exciters.

Stability of synchronous operation of vibration exciters under loading and the efficiency of the transveyers in handling of granular material by an overburden spreader was tested using an R&D model with capacity of 10 t (Figure 4). The tests proved workability of the vibro-belts in full-scale design of overburden spreaders.

![Figure 4. View of the test vibratory spreader.](image)

3. Conclusion
Operation of double drive feeders with elastic conveying surface in the capacity of transveyers of a hydraulic overburden spreader requires stable and synchronized behavior of the vibration exciters.

It has been found experimentally that stability of the synchronous operation mode hardens with decreasing length of elastic wave transferred by one vibration exciter to the other and dependent on the ratio of design and dynamic parameters of the transveyor.

The longitudinal vibrations of the conveying surface have no influence on the stability of the synchronous behavior of the vibration exciters.

The efficiency of the proposed load carrier in the structure of the overburden spreader has been proved experimentally.

Acknowledgements
The study was supported in the framework of the Basic Research Program, Project No. AAAA-A17-117122090003-2.

References
[1] Molotilov SG, Vasiliev EI, Korteliev OB, Norri VK, Levenson SYa, Gendlina LI and Tishkov AYa 2000 *Intensification of Loading-Hauling–Dumping in Open Pit Mines* Novosibirsk: SO RAN (in Russian)
[2] Zenkov IV, Nefedov BN, Baradulin IM and Kiryushina EV 2014 Modern trends and environmental problems in formation and reclamation of overburden dumps in coal extraction by the open pit method *Ekolog. Promysh*. No 6 pp 22–25
[3] Okuneva AYu and Pereverzeva VYu 2016 Formation of overburden dumps *Education, Science, Production: VIII Youth Forum Proceedings* pp 1207–1212 (in Russian)
[4] Fedorova EA 2011 Regularities in formation of overburden dumps by shovels *Vestn. ChitGU* No 3 (70) pp 110–118
[5] Dudinsky FV 2013 Methodical basis for determining parameters of stripping technology with formation of external dumps by draglines *Vestn. IrkGTU* No 12(83) pp 149–153
[6] Levenson SYa, Gendlina LI, Eremenko YuI, Morozov AV, Protasov SI and Goldobin VA 2009 Utility Patent No 88004 V65G27/00 Vibratory spreader
[7] Levenson SYa, Gendlina LI, Usoltsev VM, Goldobin VA and Morozov AV 2012 Utility Patent No 121800 V65G27/00 Vibratory spreader
[8] Levenson SYa, Gendlina LI, Morozov AV and Usoltsev VM 2014 Equipment for safe formation of overburden dumps by heavy-duty dump trucks Prospects for Innovative Development in Coal Region of Russia: IV Int. Conf. Proc. Pp 181–183 (in Russian)
[9] Morozov AG and Kulikova EG 2011 Formation of overburden dumps using dump trucks and vibratory machines Geology and Subsoil Development: XV Int. Symp. After Academicia MA Usov for Students and Young Scientists Tomsk Vol 2 pp 383–385 (in Russian)
[10] Levenson SYa, Gendlina LI, Morozov AV and Usoltsev VM 2016 Improvement of dumping in open pit mining GIAB pp 96–105
[11] Tishkov AYa, Gendlina LI, Eremenko YuI and Levenson SYa 2000 Vibration action on flowing medium during its discharge from a reservoir J. Min. Sci. Vol 36 No 1 pp 47–51
[12] Spivakovsky AO and Goncharevich IF 1983 Vibrating and Undulating Transportation Machines Moscow: Nauka (in Russian)
[13] Gendlina LI and Kulikova EG 2015 Numerical modeling of dynamics of a vibration feeder for mined rock discharge GIAB No 11 pp 224–230
[14] Tishkov AYa 1974 Theory and practice of engineering ore handling machines based on the principle of a running wave Synopsys of Dr. Eng. Sci. Dissertation Novosibirsk (in Russian)
[15] Blekhman II 1971 Synchronization of Dynamic Systems Moscow: Nauka (in Russian)
[16] Blekhman II, Vasilikov VB and Yaroshevich NP 2013 Possibilities of improvement of vibrating machines with self-synchronizing inertia vibration exciters Probl. Mashinostr. Nadezhn. Mashin No 3 pp 18–22
[17] Gordeev BA, Okhulkov SN, Plekhov AS and Titov DYu 2016 Wobbling during synchronization of two drives mounted on the joint viscoelastic basis Trudy NGTU im. RE Alekseeva No 2(113) pp 75–85
[18] Nagaev RF and Shishkin EV 2003 ASelf-synchronization of inertia vibration exciters in a vibratory cone crusher Obog. Rud No 1 pp 33–36
[19] Yaroshevich TS, Timoshchuk VN and Yaroshevich NP 2011 Dynamic synchronization of unbalanced vibration exciters with multiple frequencies Vestn. SevNTU No 120 pp228–233
[20] Protasov SI, Molotilov SG, levenson SYa and Gendlina LI 1979 Vibrating conveyor test result Rukopis dep. V TSNIEUgol No 1634 Kemerovo (in Russian)
[21] Kreimer VI and Tishkov AYa 1972 Oscillations of a vibrating belt and their damping over its length J. Min. Sci. Vol 8 No 3 pp 345–347