Application of vibratory-percussion crusher for disintegration of supertough materials

E V Shishkin¹, S V Kazakov²

¹ Saint-Petersburg Mining University, 2, Line 21 Vasilyevsky Island, St. Petersburg, 199106, Russia
² REK 'Mekhanobr-tekhnika' (ZAO), 3, Line 22 Vasilyevsky Island, St. Petersburg, 199106, Russia

E-mail: Shishkin_ev@spmi.ru, atom2@inbox.ru

Abstract. This article describes the results of theoretical and experimental studies of a vibratory-percussion crusher, which is driven from a pair of self-synchronizing vibration exciters, attached to the shell symmetrically about its vertical axis. In addition to that, crusher’s dynamic model is symmetrical and balanced. Forced oscillation laws for crusher working members and their amplitude-frequency characteristics have been inducted. Domains of existence of synchronous opposite-phase oscillations of crusher working members (crusher’s operating mode) and crusher capabilities have been identified. The results of mechanical and technological tests of a pilot crusher presented in the article show that this crusher may be viewed as an advanced machine for disintegration of supertough materials with minimum regrinding of finished products.

1. Introduction

Crushing and grinding of various solid materials are the most common and energy-consuming processes, absorbing about 10 % of the whole electric power generated in Russia. Such huge power costs stem from the fact that the existing capacity of crushing and grinding facilities applied in Russia and abroad, using conventional disintegration techniques, is limited by their kinematic features, not allowing for reduction of power consumption of production processes, cutting down non-recoverable losses of valuable components and improvement of commercial product quality. In reference with the above-mentioned, the task of power consumption reduction related to disintegration is considered one of the most important issues of minerals and man-made raw materials processing. This problem can be solved only by developing fundamentally new high-efficiency machinery and raw material processing technologies. It appears that in this regard vibrating disintegration techniques hold maximum promises, e.g. using the vibratory cone crusher on the basis of a dual-mass system without rigid kinematic links between crushing media with vertical oscillation of working members [1, 2]. This machine utilizes a practical material breakage principle – mineral release on the smallest new surface. The vibratory-percussion crusher, developed by the Research and Engineering Corporation «Mekhanobr-Tekhnika» is designed for crushing various types of natural and man-made mineral raw material. This machine is characterized by such advantages as a high reduction ratio and low content of small size fractions in the crushed material, which in its turn has high practical value for the products manufactured of this material.
2. The mechanical and mathematical model of the crusher

The crusher, which concept layout is shown in Figure 1, consists of a shell and a crushing head connected to the shell by hard spring packs. Both units can move relative to each other along a vertical axis. The shell is supported by a fixed base using vanishing hard springs. The shell also carries two symmetrically located inertial vibration exciters, generating excitation force, fluctuating in time in accordance with the harmonic law \( F = H \sin \omega t \), where \( H \) is the force amplitude and \( \omega \) is the angular frequency [3 – 6]. With such installation of the vibration exciters, the self-synchronization margins turn out to be sufficiently high and weakly depend on the machine operation mode. Apart from this, the fact that no stiff kinematic links between two self-synchronized vibration exciters are present makes the machine considerably easier to maintain and reliable in operation. At that, the crusher dynamic scheme is symmetrical and balanced, hence the alternating loads applied to the foundation may be ignored [2, 7].

\[
\begin{align*}
    m_1 \ddot{y}_1 + \beta (\dot{y}_1 - \dot{y}_2) + c(y_1 - y_2) &= 2m_e \omega^2 \sin \omega t, \\
    m_2 \ddot{y}_2 + \beta (\dot{y}_2 - \dot{y}_1) + c(y_2 - y_1) &= 0,
\end{align*}
\]  

Figure 1. The concept layout of a vibratory cone crusher.

Differential equations for center-of-mass motion of the shell and the crushing head take the form [8]:

\[
\begin{align*}
    m_1 \ddot{y}_1 + \beta (\dot{y}_1 - \dot{y}_2) + c(y_1 - y_2) &= 2m_e \omega^2 \sin \omega t, \\
    m_2 \ddot{y}_2 + \beta (\dot{y}_2 - \dot{y}_1) + c(y_2 - y_1) &= 0,
\end{align*}
\]

where \( y_1, y_2 \) are the vertical center-of-mass displacements of the shell and the crushing head; \( m_1, m_2 \) – the masses of the shell and the crushing head; \( m_e, e \) – the mass and the eccentric mass misalignment of the inertial vibration exciter; \( c \) – the effective coefficient of springs rigidity connecting the shell and the head. At that, the rigidity of elastic shock absorbers is neglected. Apart from that, the presence of a viscous interaction force is assumed between the shell and the crushing head at the coefficient \( \beta \) (roughly accounts for availability of the material in the crushing chamber within the framework of the model under consideration) [9, 10]. The value of this coefficient may be selected through experimental determination of energy losses over crusher’s synchronous oscillations period. In this case, the dynamic model under consideration may be also tentatively used for analysis of vibro-impact operating mode. It is also assumed, that the vibration exciter rotors rotate smoothly with a preliminarily unknown synchronous angular frequency \( \omega \) and the same phase shifts \( \alpha_1 \) and \( \alpha_2 \), that is, \( \alpha_1 - \alpha_2 = 0 \). Such mode of rotation of the vibration exciters is called the synchronous-sinphase (crusher’s operating mode).

The solutions of this system, namely the center-of-mass oscillation amplitudes of the shell and the crushing head in a non-impact mode, are expressed by the following formulas:
450 vibratory cone crusher

\[ y_1 = \frac{m_1 e}{m_1 + m_2} \left[ \left( \frac{m_2 \omega^3}{m_1} k^2 - \omega^2 \right) \frac{1}{\Delta} - 1 \right] \sin \omega t - \frac{m_1 \omega^3}{m_1} \frac{2n\omega}{\Delta} \cos \omega t \],

\[ y_2 = -\frac{m_1 e}{m_1 + m_2} \left[ \frac{\omega^3 (k^2 - \omega^2)}{\Delta} + 1 \right] \sin \omega t - \frac{2n\omega^3}{\Delta} \cos \omega t \].

Here the following symbols are introduced:

\[ \Delta = (k^2 - \omega^2)^3 + 4n^2 \omega^2, \]

\[ k = \sqrt{\frac{c}{m}} \] – natural frequency of consolidated oscillation of the dual-mass system under consideration,

\[ n = \frac{\beta}{2m} \] – viscous damping relative factor,

\[ m = \frac{m_1 m_2}{m_1 + m_2} \] – reduced mass of the dual-mass system under consideration.

3. Mechanical and technological tests of the crusher

Based on the results of theoretical studies and full-scale experiments on VKD-300 vibratory cone crusher, the improved design of the machine on the basis of the dual-mass system with 450 mm crushing head diameter was developed. The plain view of the crusher is shown in Figure 2. For the purpose of effective material breakage the elastic crusher system was tuned to operation in the superresonance frequency range. With such setting the working members of the machine synchronously oscillate in reverse-phase with high frequency. Figure 3 shows frequency response functions of the shell and the crushing head with account for their phases, where checkpoints corresponding to the amplitudes obtained during pilot tests are shown in addition to ideal curves. The graphs of oscillation amplitudes against vibration exciter rotational speed, obtained as a result of theoretical and experimental studies of the crusher and shown in Figure 3, prove that in the superresonance frequency range \((\omega < n_2)\) the shell and the crushing head move in phase, and approaching the resonance, the oscillation frequency of the shell progressively fades down, while that of the crushing head actively increases. It is also essential that when \(\omega = n_2\) the shell is motionless \((a_1 = 0)\), hence the antiresonance phenomenon takes place. In the superresonance frequency range \((\omega > n_2)\), the shell and the crushing head start moving in a reverse phase, which in fact is the crusher operation mode, wherein the shell amplitude has grown compared to the subresonance range, but remains virtually unchanged with increasing rotation speed, whereas the crushing head oscillations asymptotically approach the X-axis.

Figure 2. VKD-450 vibratory cone crusher.
Technological tests of VKD-450 crusher with adjustment of movement of the shell and the crushing head were performed using a supertough material – electrocorundum. Electrocorundum is a manufactured mineral distinguished for its high alumina content (over 99 %). It is highly strong and hard and ranks second after diamond by Mohs scale. The main technological task during testing was to obtain the crusher performance exceeding 3 t/h of final product containing sharp-edged fine grains with -50+0 mm feedstock.

4. Conclusion
On the basis of performed theoretical and experimental studies of VKD-450 crusher operation, it may be concluded that synchronous opposite-phase movement of the shell and the crushing head (crusher’s operating mode) takes place in the super-resonance range ($\omega > 117 \text{ s}^{-1}$); in the dynamic system under consideration, the resonance starts when disturbing frequency value agrees with the rated value of natural frequency of the system free oscillations ($\omega \approx 130 \text{ s}^{-1}$), whereas the antiresonance appears at disturbing frequency equal to partial oscillation frequency of the crushing head ($\omega \approx 117 \text{ s}^{-1}$).

As a result of technological tests (crushing of super-tough material – electrocorundum), the 4.400 kg/h performance was achieved while saving sharp-edged grains in the final product.

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