A Vibrating Jaw Crusher with Auteresonant Electric Motor Drive of Swinging Movement

E A Zagrivniy and D A Poddubniy a

Saint-Petersburg Mining University, 21-st line, Saint-Petersburg, 199106, Russian Federation

E-mail: a poddubniy.da@yandex.ru

Abstract. The article relates to a vibrating jaw crusher with pendulum vibrating exciter auteresonant electric motor drive and with elastic element rational force distribution, with limited peak-to-peak swing. Its design and its math model are presented. Also disclosed is the operating principle of a vibrating jaw crusher and the control algorithm for controlling the crushing jaw for maintaining the operating mode at resonant frequency.

1. Introduction

Traditional vibrating jaw crushers are provided with electromechanical vibrating exciters based on self-synchronized unbalanced rotors with the alternating current electric drive. Such vibrating jaw crushers are inefficient due to power losses in the behind-resonance zone, for operating those at the resonant frequency of a mechanical system is not possible because of Sommerfeld effect. Therefore, electric motor power 2-5 times exceeds the calculated one, thus making the drive efficiency low [1,2,3].

2. Materials and methods

The Mining University in Saint-Petersburg developed vibrating jaw crushers with autoresonant reciprocating motion electric motor drive of pendulum exciters that do not have the abovementioned disadvantage. The crusher comprises a body hosting two moving jaws and spring suspensions, two pendulum exciters, two petal couplings, two electric motors and elastic elements that may be torsional, or an elastic system based on several torsion springs.

A torsion used as an elastic element may result in increased load torque on petal coupling, for it links a pendulum vibrating exciter and a motor shaft, which, in turn, has its opposite end connected to the torsion attached at one side to the crusher body. A crusher of this type has the torque acting on the petal coupling in all operating modes regardless technological load, reaching maximum value, when rotor speed curve passes zero, to be defined by multiplying the acceleration and the sum pendulum mass moment and the electric motor drive torque. This is because the elastic element (torsion) and the pendulum vibrating exciter are located opposite relatively the petal coupling.

To eliminate the abovementioned disadvantages, the vibrating jaw crusher is designed, as presented at the figure 1.
The proposed vibrating jaw crusher with pendulum vibrating exciter auteresonant electric motor drive is capable of operating at the resonant frequency. Pendulum vibrating exciters of each crushing jaw 5 are connected to electric motor 14 rotor shaft via petal couplings 10 by the intermediate shaft 16 and the coupling 13. Each pendulum vibrating exciter 5 is firmly attached to two torsional springs (7 and 8). Those springs have one end firmly attached to the jaw, and the second end attached to the vibrating exciter. Also, every intermediate shaft 16 is firmly attached to the fixture ring 15, which, in turn, is fixed to one end of the third torsion spring 17 (an elastic element of the electric motor rotor), and the second end is fixed to the body 4. Using an elastic system like this one allows cutting notably the mass moment at each coupling, resulting from torque by the pendulum vibrating exciter [4,5]. Because of symmetry, calculating the crusher necessitates considering only one half of the selected structure scheme.

The electromechanical system of the vibrating jaw crusher disclosed herein comprises three vibration systems work table equal own frequencies being excited by the same source (figure 1). The first mechanical vibration system is created by the moving jaw 1 of the crusher, with the ram 2, the pendulum vibrating exciter 5 with the shaft 9, torsional springs 7-8 and the body 4 of the crusher. The weight of
the jaw with the ram and the pendulum vibrating exciter \( m_1 \) and the elastic pendulum element with stiffness factor \( C_1 \) provide the defined own vibration frequency:

\[
\omega_{11} = \frac{C_1}{m_1}.
\]  

(1)

The second, exciting, mechanical vibrating system is created by the rotor of the electric motor 14 with the intermediate shaft 16, the petal half-coupling 10, the connecting coupling 13, the elastic element of the electric motor 17 rotor. The own vibration frequency of the system:

\[
\omega_{22} = \frac{C_2}{J_2},
\]

(2)

where \( C_2 \) is the stiffness factor of the torsion spring, \( N\cdot m/\text{rad} \); \( J_2 \) is the sum mass moment of the system’s rotating elements, \( \text{kg} \cdot \text{m}^2 \).

The third vibrating system is created by the pendulum vibrating exciter 5, the pendulum shaft 9, the petal half-coupling 10 and the elastic element of the pendulum vibrating exciter (7, 8). The own vibration frequency of the system is in accordance with:

\[
\omega_{33} = \frac{C_3}{J_3},
\]

where \( C_3 \) is the stiffness factor of the torsion ring, \( N\cdot m/\text{rad} \); \( J_3 \) is the sum mass moment of the system’s rotating elements, \( \text{kg} \cdot \text{m}^2 \).

Therewith, all own frequencies of the three systems approximately equal:

\[
\omega_{11} \approx \omega_{22} \approx \omega_{33}.
\]

(4)

Such a device and such settings provide string inertial association between the vibrating systems with minimum power spending to be taken from the power mains, when operating at the resonance frequency, with high efficiency factor \([6]\). It is worth mentioning separately, that such layout provides minimal loading torque onto the petal coupling, which is to be defined by the technological load only. In addition, having the three vibrating systems expands the operating resonant frequency range.

3. Mathematical model

Building the system math model needs the second type Lagrange equations to be used \([7,8]\):

\[
\frac{d}{dt}\left( \frac{\partial T}{\partial q_i} \right) - \frac{\partial T}{\partial q_i} + \frac{\partial F}{\partial q_i} + \frac{\partial \Pi}{\partial \dot{q}_i} = Q_i, \quad i = 1, \ldots, n,
\]

(5)

where \( T \) and \( \Pi \) represent kinetic and potential power of the system, correspondingly; \( F \) is the dissipative function; \( q \) is the generalized system coordinates; \( Q_i \) is the generalized external force; \( n \) is the number of freedom degrees.

With \( q_1 = x \) and \( q_2 = \phi \), as the generalized coordinates, the system (5) would be as follows:

\[
\frac{d}{dt}\left( \frac{\partial T}{\partial x} \right) - \frac{\partial T}{\partial x} + \frac{\partial F_x}{\partial x} + \frac{\partial \Pi}{\partial \dot{x}} = Q_x.
\]

\[
\frac{d}{dt}\left( \frac{\partial T}{\partial \phi} \right) - \frac{\partial T}{\partial \phi} + \frac{\partial F_\phi}{\partial \phi} + \frac{\partial \Pi}{\partial \dot{\phi}} = Q_\phi.
\]

(6)

A number of allowances had been accepted in connection therewith (figure 2): elastic elements are linear; electromagnetic transitional processes are omitted because they take too little time compared to mechanical vibration period; power losses in the electric drive are defined by the electric motor efficiency factor; friction force in bearings and during ram motion is zero.
Figure 2. The dynamical calculation diagram of vibrating jaw crusher with pendulum vibrating exciter electric motor drive:

(1–11th elements’ numeration matches the one at the figure 1); \( \mu \) – viscous friction coefficient; \( F_1 \) – force of elasticity of suspension elements; \( c_1 \) and \( c_2 \) – stiffness factors of the elastic elements of the suspension of the crushing jaw and the rotor with a pendulum exciter; \( M \) – weight of crushing jaw with a ram and a shaft of the pendulum vibrating exciter; \( m \) – weight of a pendulum vibrating exciter; \( g \) – acceleration of gravity; \( J_1 \) and \( J_2 \) – moment of inertia of the pendulum vibrating exciter and rotor; \( OK \) is the distance equal to \( a \), where \( K \) – center of gravity of the pendulum exciter.

The kinetic power of the system is defined by the expression:

\[
T = T_1 + T_2 + T_3, \tag{7}
\]

where \( T_1 \) is the kinetic power of the crushing jaw 1 with the ram 2 and the shaft 9 of the pendulum vibrating exciter 5; \( T_2 \) is the kinetic power of the pendulum vibrating exciter 5; \( T_3 \) is the kinetic power of the rotor of the driving motor 14.

The crushing jaw with the ram moves progressively, therefore

\[
T_1 = \frac{1}{2} M \cdot \dot{x}^2. \tag{8}
\]

The kinetic power of the pendulum vibrating exciter is defined in accordance with the expression

\[
T_2 = \frac{1}{2} m \cdot \dot{x}^2 + m \cdot a \cdot \dot{x} \cdot \dot{\phi} \cdot \cos \phi + \frac{1}{2} J_1 \cdot \dot{\phi}^2. \tag{9}
\]

The kinetic power of the rotor is

\[
T_3 = \frac{1}{2} J_2 \cdot \dot{\phi}^2. \tag{10}
\]

The kinetic power of the system is equal to

\[
T = \frac{1}{2} \dot{x}^2(M + m) + m \cdot a \cdot \dot{x} \cdot \dot{\phi} \cdot \cos \phi + \frac{1}{2} \dot{\phi}^2(J_1 + J_2). \tag{11}
\]

The potential power of the system is

\[
P = P_1 + P_2 + P_3 + P_4, \tag{12}
\]

where \( P_1 \) is the potential power of the pendulum vibrating exciter 5; \( P_2 \) is the potential power of the elastic element of the pendulum (7 and 8); \( P_3 \) is the potential power of the elastic element of the electric motor 17 rotor; \( P_4 \) is the potential power of the elastic elements 11 of the jaw suspension.
For the pendulum vibrating exciter
\[ P_1 = m \cdot g \cdot a \cdot (1 - \cos \varphi). \] (13)

For the elastic element of the pendulum vibrating exciter
\[ P_2 = \frac{1}{2} C_3 \cdot \varphi^2. \] (14)

For the elastic element of the electric motor rotor
\[ P_3 = \frac{1}{2} C_2 \cdot \varphi^2. \] (15)

For the jaw suspension elastic elements
\[ P_4 = \frac{1}{2} C_1 \cdot x^2. \] (16)

Therefore, the potential power of the system is
\[ P = m \cdot g \cdot a \cdot (1 - \cos \varphi) + \frac{1}{2} C_3 \cdot \varphi^2 + \frac{1}{2} C_2 \cdot \varphi^2 + \frac{1}{2} C_1 \cdot x^2. \] (17)

By putting the results into the Lagrange equation (6) and making appropriate conversions, we obtain the differential equations describing the system motion:
\[
\left\{ \begin{align*}
(M + m) \ddot{x} + \mu \dot{x} + C_1 \cdot x &= m \cdot a \cdot (\dot{\varphi}^2 \cdot \sin \varphi - \dot{\varphi} \cdot \cos \varphi) \\
(J_1 + J_2) \ddot{\varphi} + C_2 \cdot \varphi + C_3 \cdot \dot{\varphi} &= M_x - m \cdot a \cdot (g \cdot \sin \varphi + \ddot{x} \cdot \cos \varphi)
\end{align*} \right. \] (18)

The distinctive features of the system include using nontraditional pendulum vibrating exciter autoresonant electric motor drive that allows obtaining autoresonant vibrations of the rotor with the peak-to-peak values of 60°, 120° and 180°, when using three-phase stator with one pair of poles, and 30°, 60° and 90° with two pairs of poles [9,10].

Peak-to-peak limitation feature allows creating vibrating machines capable of working in all modes without exceeding the limits of mechanical tensions in heavy-loaded systems.

Worth mentioning is the useful feature of the system disclosed – the capability to automatically decrease the amplitude of the vibrations by the pendulum vibrating exciter maintaining the amplitude of vibrations by the jaw when the load open source reduced – something that is the property of systems with two degrees of freedom (vibration dynamic attenuation effect with $\mu \to 0$, $\varphi \to 0$).

4. Conclusion

Providing resonant vibrations of the moving jaws and pendulum vibrating exciters uses such a method for exciting and regulation of the autoresonant vibrations that each vibration half-period has the speed modified, and when the rotor speed curve passes zero relatively the stator, the voltage is applied to the stator windings to form electromagnetic torque to be modified in a co-phase manner with the rotor vibration speed, and the defined amplitude value controlled by changing the voltage via negative feedback by the vibration speed amplitude value during every half-period of the vibrations. Implementing those conditions relies on the reciprocating motion electric drive. It may, for example, be the one with three phase stator from the regular asynchronous electric drive equipped with the magnetic rotor with a single pair of poles. The electric drive allows obtaining fixed angle of the rotor vibrations, wherein the value may depend on particular winding connection scheme.

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