

# Vibration cone crusher for disintegration of solid materials

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**Abstract.** Traditionally, during the disintegration of natural and technogenic mineral raw materials, as well as other materials, crushing machines with a kinematic drive of working bodies are used. However, such a drive has a low degree of crushing and high energy costs. In addition, if we are talking about the disintegration of natural and technogenic raw materials containing mineral aggregates, then in these machines it is practically impossible to ensure sufficient selectivity of the liberation. The aim of the research is to develop an energy-efficient method of destroying various materials with the implementation of the process of selective liberation of minerals with the simplest design of the machine (without rigid kinematic connections of the working bodies, without oil stations and special slide bearings). In the course of the research, a mathematical analysis of the dynamics of the vibration crusher was carried out to determine the required mechanical parameters, which made it possible to adjust the body and the crushing cone of the machine to high-frequency synchronous-antiphase oscillations in the operating frequency range. As a result of the conducted experimental tests, the reliability of theoretical studies was confirmed and the dependences of technological indicators on the controlled parameters of the machine were determined in the operating frequency range.

## 1. Introduction

The main tasks in the disintegration of solid rock and technogenic materials are the opening of minerals with a minimal newly formed surface and a reduction in the specific power consumption [1, 2]. The most common crushing machines in the industry are the machines with a kinematic drive of active members, which have a low degree of crushing and high energy costs. In addition, it is practically impossible to ensure selective crushing and protection of crystals from breaking in these machines due to the lack of force interaction between the pieces and the uncertain nature of the forces that arise when the material is crushed. Currently, the industrial use of a new class of vibration machines characterized by high disintegration efficiency has been mastered as a result of a long cycle of research and development work in REK "Mekhanobr-tehnika". One of such effective vibration machines is the cone inertial crusher (CIC) [3]. The crushing cone of the machine is driven by an unbalanced vibrator, which makes it possible to exclude rigid kinematic links between the active members, and the crushing force developed by the vibration exciter shall be determined by mechanics and does not depend on the properties of the material being processed. The crusher is dynamically balanced, the entering of the unbreakable body in the working chamber does not cause breakdowns of

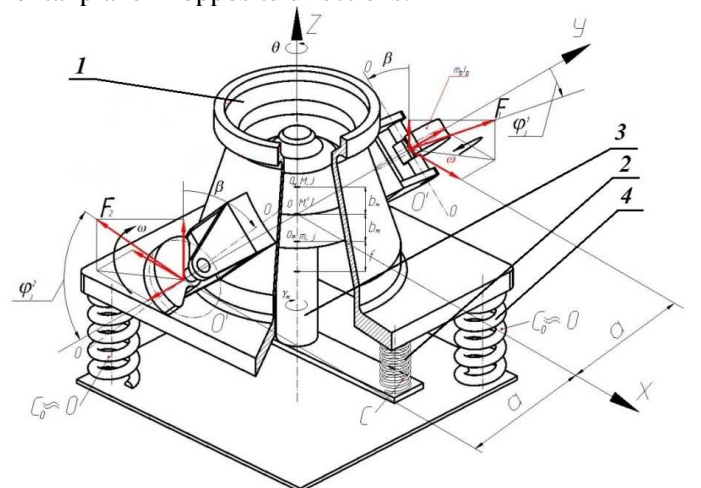


the mechanisms. However, in the presence of high technological parameters, productivity and degree of crushing, this machine has drawbacks. So, having quite large possibilities of adjustment of both mechanical and dynamic parameters, the CIC does not always succeed in reducing the output of small classes to the necessary degree. When concentrating ores containing rare elements (e.g., diamonds), this can have negative consequences. In addition, the machine has a fairly extensive system of lubrication of the elements, which not only complicates its design, but also increases the likelihood of failure of the babbit friction bearing, which is the support of the crushing cone, in case of a possible failure in the supply of oil. The limitations include the inability of the crusher to run without load (without material in the crushing chamber). With the aim of the implementation of the process of selective opening of minerals with the simplest design of the machine (without rigid kinematic links of active members, oil stations and special friction bearings), which provides energy-efficient destruction of materials, a shock-vibratory cone crusher with spatial vibrations of active members was developed [4]. This machine, due to its quasi-resonant operation, has such advantages as high degree of crushing and minimal content of small classes in the finished product.

## 2. Analysis of the dynamics of the shock-vibration cone crusher

The crusher consists of a softly cushioned body (supporting body) and a crushing cone screwed to it with special packages of helical springs. In this operation mode, the cone has one translational and one rotational degree of freedom in relation to the body. The machine is driven by two self-synchronizing unbalance vibro-exciter mounted on the body. With such a placement, the self-synchronization margin turns out to be the highest and weakly dependent on the operation mode. The absence of rigid kinematic links between two self-synchronizing exciters makes the crusher exceptionally easy to maintain and reliable in operation; in addition, the machine is completely balanced.

The design scheme of the shock-vibration cone crusher is shown in Figure 1. The body of the machine 1, and also the crushing cone 3 attached to it by means of special packages of helical springs 2 can be considered as absolutely solid bodies, which perform rectilinear vibrations along the vertical axis  $z$  and rotational vibrations around the same axis in the operation mode. In this case, the body is connected to the fixed base by flexible shock absorbers 4, the rigidity of which is negligibly small. The unbalanced vibro-exciter 5 located in the body (the carrying body) are unbalanced rotors driven by rotation from two independent asynchronous motors and rotating in intersecting planes inclined at an angle  $\beta$  to the horizontal plane in opposite directions.



**Figure 1.** The design scheme of the vibration cone crusher with the spatial motions of the active members.

The dynamics of the considered machine was investigated in the paper [5], in which the laws of forced helical vibrations of the body and the cone of the crusher were obtained and analyzed, as well as the expression for the stability factor of the synchronous-in-phase rotation of the exciters. In this case, resistances to the vibrations of the body and the cone of the machine proportional to the first

degrees of velocities of their centers of gravity were taken into account; thus taking into account the presence of material in the crushing chamber. To determine the conditions for the existence and stability of the synchronous-anti-phase motion of the body and the crushing cone in the operation mode, it is extremely important to take into account the influence of the material being destroyed on the stability of this mode. Therefore, to study this issue, we used a model of an oscillating system with a linearly-viscous damper  $\delta$  between crushing bodies [6-8].

In the operation mode of the crusher, when the vibration exciters rotate synchronously and in phase, the laws of forced helical vibrations of the body and the crushing cone looks as follows:

for vertical vibrations

$$\begin{aligned} z_1 &= \frac{s \cos \beta}{m_1 + m_2} \left[ \left( \frac{m_2 \omega^2}{m_1} \frac{k^2 - \omega^2}{\Delta} - 1 \right) \sin \omega t - \frac{m_2 \omega^2}{m_1} \frac{2n\omega}{\Delta} \cos \omega t \right], \\ z_2 &= -\frac{s \cos \beta}{m_1 + m_2} \left\{ \left[ \frac{\omega^2 (k^2 - \omega^2)}{\Delta} + 1 \right] \sin \omega t - \frac{2n\omega^3}{\Delta} \cos \omega t \right\}, \end{aligned} \quad (1)$$

for rotary oscillation

$$\begin{aligned} \theta_1 &= \frac{s_e \sin \beta}{I_1 + I_2} \left[ \left( 1 - \frac{I_2 \omega^2}{I_1} \frac{k_1^2 - \omega^2}{\Delta_1} \right) \sin \omega t + \frac{I_2 \omega^2}{I_1} \frac{2n_1 \omega}{\Delta_1} \cos \omega t \right], \\ \theta_2 &= \frac{s_e \sin \beta}{I_1 + I_2} \left\{ \left[ 1 + \frac{\omega^2 (k_1^2 - \omega^2)}{\Delta_1} \right] \sin \omega t - \frac{2n_1 \omega^3}{\Delta_1} \cos \omega t \right\}. \end{aligned} \quad (2)$$

Here the designations are introduced:

$$\begin{aligned} \Delta &= 4n^2 \omega^2 + (k^2 - \omega^2)^2, \\ \Delta_1 &= 4n_1^2 \omega^2 + (k_1^2 - \omega^2)^2. \end{aligned}$$

Here  $z_1, z_2$  are the vertical centre-of-mass displacements of the body and the crushing cone;  $\theta_1, \theta_2$  – rotations of the body and cone with respect to the  $z$  axis;  $m_1, m_2$  – the masses of the body and the crushing cone;  $I_1, I_2$  – the central moments of inertia of the body and cone;  $s_e$  – the static moment of the inertial vibration exciter;  $\omega$  – synchronous angular velocity of vibro-exciter;  $k, k_1$  – own frequencies of free vertical and rotational vibrations of the system under consideration;  $n, n_1$  – specific coefficients of viscous resistance;  $c_1, c_2$  – the total stiffnesses of the packages of helical springs connecting the body and the cone for compression-tension and torsion, respectively;  $\beta_1, \beta_2$  – coefficients of equivalent viscous resistance to vertical (rectilinear) and horizontal (rotary) vibrations of the body and cone, respectively;  $m$  – the reduced mass of the two-mass system under consideration;  $I$  – the reduced central moment of inertia of the system.

For the stability of the synchronous-in-phase rotation mode of the exciters according to the conclusions of the synchronization theory [9], the stability factor should be positive:

$$\begin{aligned} \chi &= 2(m_1 + m_2 + 2m_e) \left\{ \frac{\cos^2 \beta}{m_1 + m_2 + 2m_e} \left[ \frac{m_2 \omega^2}{m_1 \Delta} (k^2 - \omega^2) - 2 \right] + \frac{a^2 \sin^2 \beta}{I_1 + I_2 + 2m_e a^2} \cdot \right. \\ &\cdot \left. \left[ \frac{I_2 \omega^2}{I_1 \Delta_1} (k_1^2 - \omega^2) - 1 \right] + \frac{2}{m_1 + m_2 + 2m_e} + \frac{a^2 \cos^2 \beta}{I + 2m_e a^2} \right\} > 0. \end{aligned} \quad (3)$$

In addition, the larger its value, the greater the margin for stability.

In order to assess the impact of the material to be destroyed on the dynamics and stability of the operating conditions of the shock-vibration cone crusher, it is necessary to determine the numerical values of the coefficients of viscous resistance for rectilinear and rotational vibrations of the body and crushing cone along and around the vertical axis, respectively, according to the formulas [10]:

$$\beta_1 = \frac{m^2 s^2 \omega^4 \cos^2 \beta (1 + \zeta)}{m_1^2 \zeta N} + \sqrt{\frac{m^4 s^4 \omega^8 \cos^4 \beta (1 + \zeta)^2}{m_1^4 \zeta^2 N^2} - \frac{m^2 (k^2 - \omega^2)^2}{\omega^2}},$$

$$\beta_2 = \frac{I'^2 s^2 b^2 \omega^4 \sin^2 \beta (1 + \zeta)}{I_1^2 N} + \sqrt{\frac{I'^4 s^4 b^4 \omega^8 \sin^4 \beta (1 + \zeta)^2}{I_1^4 N^2} - \frac{I'^2 (k_1^2 - \omega^2)^2}{\omega^2}}. \quad (4)$$

Here,  $N$ – the averaged power, which characterizes the useful energy consumption for destruction of the material in the working chamber of the machine;  $\zeta$  is a coefficient of proportionality between energy consumption for vertical and rotational vibrations of the body and cone of the crusher, respectively. In this case, the value of the power  $N$ , as well as the synchronous angular velocity, should be determined experimentally for this machine operation mode.

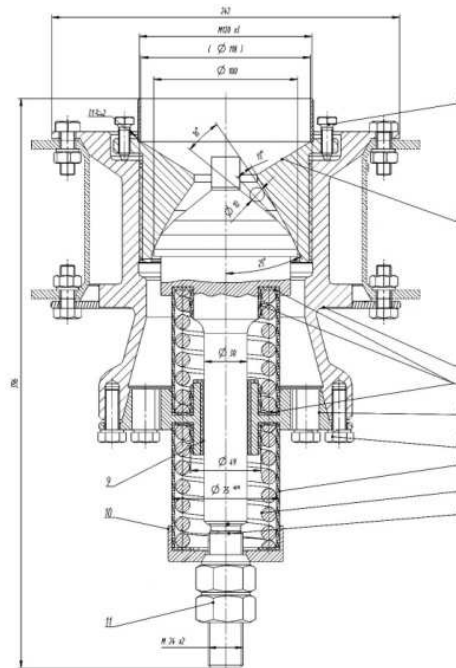
### 3. Laboratory prototype of the crusher

In order to determine the mechanical and technological parameters of the process of destruction of solid materials in the quasi-resonant mode, a laboratory prototype of a shock-vibration crusher with spatial movements of the active members was developed in accordance with Figure 2. The main elements of the laboratory prototype are the crushing module (Figure 3) and the vibration stand (Figure 4). The crushing module (Figure 3) consists of a body 1, inside which a crushing cone 4 is supported on the flange 2 by means of springs 3. Flange 2 is fixed to the body 1 with bolts 13. To



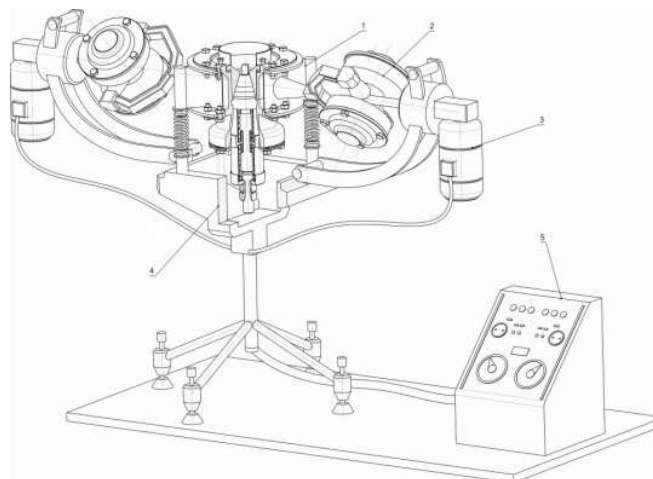
**Figure 2.** The laboratory prototype of the shock-vibration crusher with spatial motions of the active members.

adjust the unloading slot, the bowl 5 is used that is fixed to prevent the rotation using bolts 6. The springs should be protected against contact with the steel parts of the module by the gaskets 7, and by the sleeve 8 against the spilling of the crushed material inside the mechanism. In order to exclude the jamming of the shaft of the crushing cone 4 in the flange 2, a fluoroplastic bushing 9 is inside the flange 2. The cup 10 ensures the compression of the springs 3 by means of the nuts 11.



**Figure 3.** Crushing module.

The vibration stand (Figure 4) includes a vibration-proof active body (stand table) 1, to which two unbalanced exciter 2 are rigidly connected, each of which is driven by a separate electric motor 3; the synchronization of rotation of two unbalanced vibro-exciter in frequency and phase is implemented due to self-synchronization phenomenon [9-11]. Stepless speed adjustment is carried out using the control panel 5. The working table and drive of vibration exciters are located on the support frame 4.



**Figure 4.** Vibration stand.

The use of the self-synchronization phenomenon to synchronize the rotation of the exciter rotors makes it possible to arbitrarily locate the exciter axes relatively the stand table. This makes it possible to obtain various forms of vibrations - rectilinear, circular, angular elliptical, segments of arcs or hellical lines, trajectories in the form of a figure eight, etc., and thanks to a smooth adjustment of the speed of the drive, it is possible to provide a quasi-resonant mode of operation of the vibration stand.

#### 4. Results and discussion

The initial data for calculation of the dynamics of the crusher are presented in Table 1.

**Table 1.** Calculation of the dynamics of the crusher

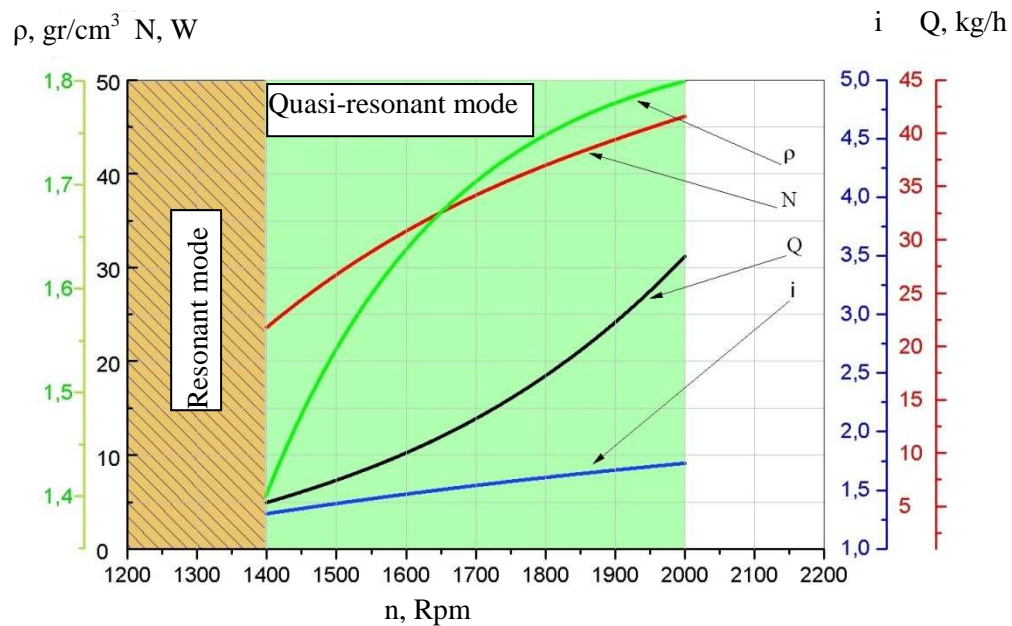
Name	Designation	Value
Distance from the axis of the machine to the axis of the exciter	$a$	0,385 m
Stiffness of packages of helical springs connecting the body and cone for compression-tension	$c_1$	29,430 N/m
Stiffness of packages of helical springs connecting the body and cone for compression-tension	$c_2$	9,810 N/m
Angle of inclination of axes of rotation of vibro-exciter to the horizon	$\beta$	$45^0$
Body weight	$m_1$	58,8 kg
Mass of crushing cone	$m_2$	4,8 kg
Mass of exciter	$m_b$	5,22 kg
Centre moment of inertia of the body	$I_1$	$6,1 \text{ kgm}^2$
Central moment of inertia of the crushing cone	$I_2$	$0,7 \text{ kgm}^2$
Eccentricity of the vibration exciter	$e$	0,0123 m

After substitution of the initial data, we obtain the following values of the natural frequencies of free vertical and rotational vibrations of the system under consideration:  $k = 81,5 \text{ s}^{-1}$ ;  $k_1 = 125,0 \text{ s}^{-1}$ .

In order to effectively destroy the material, the flexible system of the crusher was set for machine operation in the resonance frequency range. This adjustment allows the active members of the machine to perform high-frequency synchronous-antiphase vibrations.

For conduction of experimental studies on the developed laboratory sample of the crusher, a particularly strong material, fused alumina, was used. The fused alumina is an artificial mineral, characterized by a high content of aluminum oxide (more than 99%). It has a high hardness, and on the Mohs hardness scale it occupies second place after the diamond. The main technological task during the tests was to obtain the performance of the crusher by the initial feed  $-5 + 0 \text{ mm}$  more than  $3 \text{ kg/h}$  with the content in the finished product of small classes of grains with a sharp edge.

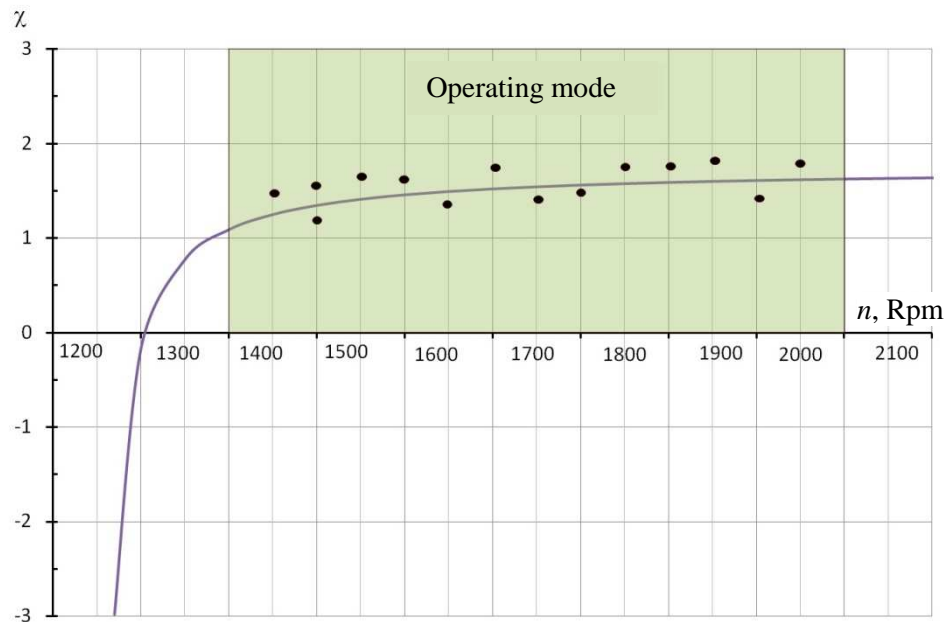
Figure 5 shows the graphs of the dependences of the net power  $N$ , the productivity  $Q$  and the degree of crushing  $i$  on the synchronous angular velocity of rotation of the exciters with the width of the discharge gap equal to  $\delta = 5 \text{ mm}$ .



**Figure 5.** The graphs of dependencies of the technological indicators of the crusher from the angular rotation speed of vibration exciters.

In addition, it was necessary to confirm the stability of the synchronous-anti-phase movement of the body and the crushing cone in the operation mode. For this purpose, the average power of motors corresponding to the energy consumption in the crusher operation mode, as well as the power corresponding to the energy losses in the bearings, was previously determined experimentally. Then, according to equations (4), the values of the coefficients of viscous resistance for vertical and rotation vibrations of the body and cone  $\beta_1$  and  $\beta_2$  were calculated. Then, using formula (3), the dimensionless coefficient of the operation mode stability was determined, whose graph of dependences on the synchronous angular velocity  $\omega$ , is shown in Figure 6. Here, checkpoints corresponding to the values of the coefficient obtained as a result of experimental studies are shown in addition to the theoretical graph.

As you can see from the graph, the dimensionless coefficient valued  $\chi$  is positive in the considered area of angular velocities, which in turn indicates the stability of the required mode. Also, the synchronous and anti-phase movement of the body and crushing cone, as well as the passage of the unbreakable piece of rock in case of its entry into the working chamber of crushing, was confirmed in the operation mode of the machine (when crushing material) using a high-speed camera (baumer VEXG-25c).



**Figure 6.** The graph of the dependence of the crusher operation mode stability coefficient on the angular velocity of rotation of exciters.

## 5. Conclusion

Based on the theoretical and experimental studies of the shock-vibration cone crusher with spatial movements of the active members, it can be concluded that the energy-efficient destruction of the material is ensured when the machine is operated in the high-frequency range (after the second resonance).

The possibility of disintegration of the material in the quasi-resonant mode will significantly improve the technological parameters of the crusher: improve the quality of the finished product and reduce the specific energy consumption of crushing.

In addition, stable synchronous anti-phase movements of the body and crushing cone are implemented in the operation mode of the machine, which are necessary for effective disintegration of the material.

## 6. Acknowledgments

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