Improvement of the construction of vibration-centrifugal unit

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Abstract. The article presents the scientific and technical studies to create an effective system for balancing the lever mechanism of the centrifugal grinding unit with the possibility of automatic adjustment during the technological grinding process for changing the size of the grinding load in the working chambers.

1. Introduction
Grinding unit [1,2] is equipped with counterweights located at the ends of the eccentric shaft which is the input link of the crank-slider mechanism. Counterweights have the ability to manually move and install in the desired position which is determined depending on the mass of the moving parts of the unit including the grinding load in the working chambers. This principle of balancing allows static balancing of the lever mechanism which is ineffective since during operation of the unit the moving parts move with sufficiently high speeds, varying in size and direction. There are significant accelerations and inertial loads, the consequence of which is the vibration of the structure which negatively affects the strength characteristics of the unit (fatigue, additional stresses, etc.).

In addition, it may be necessary to change the parameters of the operating modes of the unit, for example, changing the load factors of the working chambers in the process of grinding materials with different physical and mechanical properties. Consequently, the magnitude of the mass and the position of the center of mass of the grinding unit. All this leads to the appearance of additional vibration which entails a reduction in the service life of the components and parts of the grinding unit and adversely affects its operation in industrial conditions.

2. Calculation of the dynamic characteristics of the grinding unit
Earlier studies of the dynamics of the mechanical system of the unit [3, 4] allow quantifying the dynamic characteristics of the structure.

To assess the impact on the power characteristics of CGU of the movement of the grinding load inside the grinding chambers, the operation of the unit was considered without taking into account the impact of the grinding load on the links of the lever mechanism. The design scheme of the lever mechanism is presented on Figure 1.

To solve the problem, we used the method of determining the reduced moments which consists in the equality of the capacities developed by the reduced moment (applied to the link of the reduction - link 1) and replaced by forces and moments applied to the links of the mechanism, i.e.

\[ P_{II}(\varphi) = \sum_{i=1}^{N} P_{i}. \]
We apply this method to the lever mechanism of the CGU for the steady motion mode (angular velocity of rotation of the link 1 \( \omega = \text{const} \)).

Represent the power required to overcome the forces of resistance as an expression

\[
P_{H}(\varphi) = M_{c}(\varphi) \omega,
\]

where \( M_{c} \) is the reduced moment of resistance forces, \( N \cdot m \); \( \omega \) is the angular velocity of the link (eccentric shaft), \( \text{rad/s} \).

![Design scheme of the lever mechanism without regard to the action of the grinding load](image)

**Figure 1.** Design scheme of the lever mechanism without regard to the action of the grinding load

The magnitude of the given moment of \( M_{c} \) can be represented as follows

\[
M_{c}(\varphi) = \frac{\sum_{i} P_{i}}{\omega},
\]

where \( \sum_{i} P_{i} \) can be found from expression

\[
\sum_{i} P_{i} = \sum_{i} F_{i} v_{i} \cos \alpha_{i} + \sum_{i} M_{i} \omega_{i},
\]

where \( F_{i} \) is force applied to the link \( i \), \( H \); \( M_{i} \) is the moment applied to the link \( i \), \( N \cdot m \); \( v_{i} \) is the velocity of the point of application of force \( F_{i} \), \( m / s \); \( \omega_{i} \) is the angular velocity of the link \( i \), \( \text{rad} / s \); \( \alpha_{i} \) is the angle between the force and velocity vectors, degrees.

As a result, the expression for determining the reduced moment of force resistance \( M_{c} \) for various positions of the mechanism depending on the angle of rotation of the eccentric shaft was obtained. The origin of the angle is the positive direction of the X-axis (Figure 1). The mass of the sliders (link 3) is not taken into account since it is small compared to the masses of the eccentric shaft and the movable frame with grinding chambers fixed on it (link 2).
where \( M_{Ci} \) is the reduced moment of resistance forces for the \( i \)th position of the mechanism, Nm; \( G_1 \) is gravity of the eccentric shaft, N; \( G_1 \) is gravity of the movable frame with grinding chambers fixed on it, N; \( F_{E2i} \) is the inertia force of the grinding unit for the \( i \)th position of the mechanism, N; \( M_{E2i} \) is moment of inertia forces of the grinding unit for the \( i \)th position of the mechanism, Nm; \( G_2 \) is gravity of the eccentric shaft, N; \( F_n \) is the inertia force of the counterweight, N; \( V_{S1i} \) is velocity of the center of mass of the eccentric shaft, \( m / s \); \( V_{Sn} \) is speed of the center of mass of the grinding unit for the \( i \)th position of the mechanism, m / s; \( \omega \) is the angular velocity of the eccentric shaft, rad / s; \( \alpha_2i \) is the angle between the vectors \( \vec{G}_2 \) and \( \vec{V}_{S2i} \), degrees; \( \alpha_3i \) is the angle between the vectors \( \vec{F}_{E1i} \) and \( \vec{V}_{S2i} \), degrees; \( \alpha_4i \) is the angle between the vectors \( \vec{F}_n \) and \( \vec{V}_{Sn} \), degrees.

For the experimental model CGU with an eccentricity value of \( e = 0.02 \) m, the maximum possible value of the moment \( M_C = 19.55 \) Nm was determined.

According to the results of the study of the movement of grinding bodies in the CGU chambers a design diagram which is shown in Figure 2 was made.

![Figure 2. Design diagram of the lever mechanism, taking into account the action of the grinding load](image)

For this case, the expression (3) has the form:

\[
M_{Ci} = (G_1 \cdot V_{S1i} \cdot \cos \alpha_{1i} + G_2 \cdot V_{S2i} \cdot \cos \alpha_{2i} + F_{H2i} \cdot V_{S2i} \cdot \cos \alpha_{3i} + M_{H2i} \cdot \omega_{2i} + \\
+ 2 \cdot G_n \cdot V_{Sn} \cdot \cos \alpha_{4i} + 2 \cdot F_{Hn} \cdot V_{Sn} \cdot \cos \alpha_{5} + G_{M1} \cdot V_{SM1} \cdot \cos \alpha_{6i} + \\
+ F_{H1} \cdot V_{SM1} \cdot \cos \alpha_{7i} + G_{M2} \cdot V_{SM2} \cdot \cos \alpha_{8i} + F_{H2} \cdot V_{SM2} \cdot \cos \alpha_{9i} + \\
+ G_{M3} \cdot V_{SM3} \cdot \cos \alpha_{10i} + F_{H3} \cdot V_{SM3} \cdot \cos \alpha_{11i}) / \omega_1, \tag{6}
\]
where \( G_{M1} \) is the gravity of the grinding load of the lower chamber, N; \( F_{IM1} \) is force of inertia of the grinding load of the lower chamber, N; \( G_{M2} \) is gravity of the grinding load of the middle chamber, N; \( F_{IM2} \) is the inertia force of the grinding load of the middle chamber, N; \( G_{M3} \) is gravity of grinding load of the upper chamber, N; \( F_{IM3} \) is the inertia force of the grinding load of the upper chamber, N; \( \alpha_6 \) is the angle between the vectors \( \vec{G}_{M1} \) and \( \vec{V}_{SM1} \), degrees; \( \alpha_7 \) is the angle between the vectors \( \vec{F}_{IM1} \) and \( \vec{V}_{SM1} \), degrees; \( \alpha_8 \) is the angle between the vectors \( \vec{G}_{M2} \) and \( \vec{V}_{SM2} \), degrees; \( \alpha_9 \) is the angle between the vectors \( \vec{F}_{IM2} \) and \( \vec{V}_{SM2} \), degrees; \( \alpha_{10} \) is the angle between the vectors \( \vec{G}_{M3} \) and \( \vec{V}_{SM3} \), degrees; \( \alpha_{11} \) is the angle between vectors \( \vec{F}_{LE3} \) and \( \vec{V}_{SM3} \), degrees.

The maximum possible value of the moment is \( MS = 28.90 \) Nm. The increase in the reduced moment of resistance forces by 47.8% indicates a significant effect of the grinding load on the dynamic properties of the mechanical system.

As a result, the need to create two balancing systems for units of this type becomes obvious. One system, the main one, balances directly the lever mechanism; the second, additional, balances the action of the grinding load in the grinding unit. Moreover, the additional system should be able to automatically control depending on changes in the mass and location of the center of mass of the grinding unit.

3. Design of the modular balancing device

Based on the task a modular balancing device which provides additional balancing of the mechanism and reduction of the vibration level during the operation of the unit [9, 10] was developed. The drive diagrams of the central control unit without an additional device and with a modular balancing device are shown on Figure 3.

\[ \text{Figure 3. Driving diagram of CGU: a – without modular balancing device; b – with modular balancing device} \]

The parts of the unit are mounted on the frame 1. The rotation from the electric motor 2 is transmitted through the V-belt transmission 3 to the eccentric shaft 4. To prevent elastic deformations of the eccentric shaft due to its considerable length, gears 5 and intermediate shaft 6 are used. The frame 7 is a connecting rod in a crank-slider mechanism; it is pivotally connected with the eccentric shaft 4 and the sliders 8. On the eccentric shaft 4 there are counterweights 9, balancing the lever mechanism. The modular balancing device which is a differential gear mechanism contains gears 10
for transmitting torque from the eccentric shaft 4 to an additional hollow shaft 13, which is connected to the carrier 14 of the differential mechanism. Inside the shaft 13 there are axle shafts 15 connected to the bevel gears of the half-axes of the differential mechanism and electromagnetic clutches 11. The counterweight 12 through the spindle and the screw-and-gear transmission is connected to the satellite of the differential mechanism and performs translational movement along the guides mounted on the carrier in one or the other side when changing the rotational speed of one of the semi-axes. Electromagnetic couplings are controlled by the control system, which includes a programmable controller, a counterweight position sensor and a power supply.

Modular balancing device works as follows. When the permissible level of vibration of the unit rack (set point) is set, the clutches are disconnected and the satellites are stationary relative to the cogwheels of the semi-axes 15. The semi-axes 15 and the intermediate shaft 13 rotate at the same angular velocity. When the vibration level changes, the control system includes one of the couplings, the angular velocity of the semi-axis decreases, and as a result, the relative rotation of the satellite of the differential mechanism and the associated lead screw are obtained. The counterweight 12 moves along the axis of the spindle until the vibration level reaches the set value. It should be noted that the mass of the counterweight 12 is substantially less than the mass of the main counterbalances 9, because calculated on the basis of the nominal mass of the grinding load in the working chambers.

The control circuit of the modular balancing device is shown on Figure 4.

**Figure 4.** Modular balancing device: 1– frame; 11– shaft; 12, 26 – supports; 13 – gears; 14 – carrier; 15 – guides; 16 – transverse rod; 17 – counterweight; 18 – “screw-nut” transmission; 19 – spindle; 20 – satellite; 21 – stroke limiters; 22 – semi-axes; 23, 24 – gears; 25 – electromagnetic clutches; 27 – programmable controller; 28 – counterweight position sensor; 29 – power supply

4. Conclusion

Thus, the developed system of automatic vibration reduction, containing a modular balancing device, can maintain the specified vibration amplitude or change it periodically depending on the process conditions without stopping the unit.
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