Visualization and analysis of balancing of the slider-crank mechanism on an elastic foundation in the mining tunneling machines

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Abstract. The research rationale aimed at development and use of the slider-crank mechanism in the mining equipment is justified. The advantages, as well as functional, engineering and technological features are determined for the feeder in the tunneling machine. The development ways of the structural design solutions of the slider-crank mechanism in the mining machines are specified. The analysis for balancing of the mechanism on an elastic foundation with special pendulums attached to the crankshaft is done. The calculations to justify balancing using the pendulum are done. Modeling of the slider-crank mechanism is realized.

The park of the tunneling machines in the coal industry is amounted to about 400 units (half of them are located in Kuzbass). The machines of GPKS type by Kopeisk mechanical engineering plant form the basis of this park (e.g. 97% in Kuzbass). The state analysis of the tunneling machines is indicative of the steady decline of a new car park. The machines wearing in the major coal companies is a disturbing factor in terms of possibility to cover the required scope of preparatory work in the coal companies.

The main technological operations of the tunneling mining are as follows:

- Loosening of destructible rocks and mineral deposits.
- Withdrawal of the separated mass from the mine face and its loading on the production board (conveyor or mine cars).
- The production surface preparation for support setting, erection, lagging and backing.
- Auxiliary operations to ensure functioning of the mine face (the vale construction; development of the production transport vehicles (conveyors and rail track); capacity development of the main vent, water and air mains etc.).

The level of mechanical operations and labor costs of the first three technological operations out of four above-mentioned are largely determined by the design and the structure of the tunneling machine. The same technological operations also determine, for the most part, the rate of mining and the costs for the heading construction. The machine capacity, intended for adjusting both these operations in time, allows a significant reduction of the duty cycle duration. And the possibility of the heading construction with the high quality side surfaces and soil allows significant improvement of the performance of the machine by volume reduction of the destructible mass and a significant volume reduction of backing. The feeder has a great impact on the operational efficiency of the machine as a whole. The feeder of the machine “KPU-50” by OOO “Yurginski mashzavod” is configured into removable devices, such as pallet handles or spiders. It has separate hydraulic drives for every shovel item. The hydraulic drives of the stroke and feeder allow operating in the flood places. The chain-and-flight conveyor of the machine with the swinging tail section is capable of loading the rock mass in any mine transportation vehicle. The lift swinging chain-and-flight conveyor with the armed pin chain provides loading of the broken rock mass in any mine transportation vehicle [1].

The feeder in the form of an inclined table equipped with two gathering arms with individual synchronized hydraulic drives is designed for loading of the broken rock mass into the chain-and-flight conveyor of the machine. The feeder is pivotally mounted to the frame and is capable of lifting (450 mm) and lowering (250 mm) to the relative ground level using two hydraulic cylinders. When lowering on the ground, the feeder becomes another support attachment that improves the stability of the machine at the bottom fracture. The feeder consists of a feeder drive and a support frame interconnected with axle shafts. The support frame is the feeder pick. If necessary, the feeder is supplemented with expanders and foot plates, which allow increasing the width of the loading section from 2,400 mm to 4,000 mm. Since the feeder is equipped with the mechanical-hydraulic drive, there are no safety elements provided therein. The feeder drive consists of a welded shell, a bypass head for the conveyer chain, clutches, rockers, power-driven crankshafts, hydraulic motors and cylindrical gears. The rotation of the hydraulic motors is transmitted through the gear to the wheels and via the splined joint to the crankshafts which have the gathering arms mounted thereon. The bearing mount assemblies of the clutches, rockers, crankshafts are mechanically sealed. The housing has counterbores made to fasten the feeder to the frame and to mount the feeder lift hydrocylinders. The assembly unit under analysis consists of the following components: a drive, a clutch, a rocker. Further modeling is done for these assembly units.

The phenomenon of self-synchronization is widely used in the automatic trim systems of rotating rotors [2, 3]. In the analysis, the authors show that the phenomenon of synchronization can be applied also to balance the hinged mechanisms, in particular, of the slider-crank type.

The mechanical model of the system under analysis is shown in Figure.1, where: 3 - crankshaft, 4 – crank rod, 5 - slider, 1, 2 - loose pendulums axle-mounted on the crankshaft, Mк – correction mass. The hinged mechanism is mounted on a horizontal platform, which can do only linear motion. The platform is temporary attached to a fixed base. The hinged mechanism is driven by a motor, which rotates the crankshaft at constant angular velocity $\dot{\varphi} = \Omega$

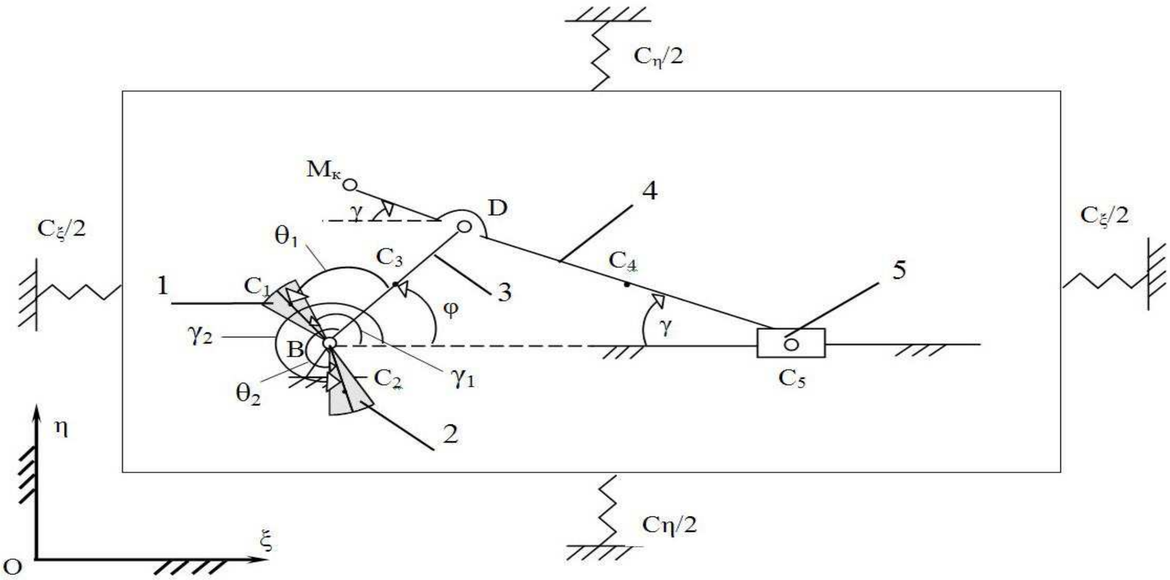

Figure 1. 1, 2 – loose pendulums axle-mounted on the crankshaft, 3 – crankshaft, 4 – crank rod, 5 – slider, Mк – correction mass.

The following coordinates of the crankshaft axle are accepted as generalized coordinates ξ_B, η_B in the system of fixed axles $\xi O\eta$ and angles of rotation γ_1, γ_2 of the pendulums in relation to fixed axle $O\xi$. Then the kinetic energy of the mechanism can be presented as follows:

$$T=\frac{1}{2}m_{\Pi}\left(\dot{\xi}_B^2+\dot{\eta}_B^2\right)+\sum_{i=1}^{2}\left[\frac{1}{2}m_i\left(\dot{\xi}_{C_i}^2+\dot{\eta}_{C_i}^2\right)+\frac{1}{2}J_{C_i}\dot{\gamma}_i^2\right]+\frac{1}{2}m_3\left(\dot{\xi}_{C_3}^2+\dot{\eta}_{C_3}^2\right)+$$
$$+\frac{1}{2}J_{C_3}\dot{\varphi}^2+\frac{1}{2}m_4\left(\dot{\xi}_{C_4}^2+\dot{\eta}_{C_4}^2\right)+\frac{1}{2}J_{C_4}\dot{\gamma}^2+\frac{1}{2}m_5\left(\dot{\xi}_{C_5}^2+\dot{\eta}_{C_5}^2\right)+\frac{1}{2}m_K\left(\dot{\xi}_K^2+\dot{\eta}_K^2\right) \tag{1}$$

Here, m_{Π} and m_K –weight of the platform and correction mass; m_i (i=1..5) – unit mass; J_{Ci} (i=1..4) – inertia moments in relation to the central axles; ξ_{Ci}, η_{Ci} and ξ_{Ki}, η_{Ki} – coordinates of mass centers C_i of the units and correction mass, which are expressed via distance $BC_i=h_i$ (i=1..3), $BD=l_3$, $DC_i=h_i$ (i=4,5), $DM_k=h_k$ and angles γ_i, φ, γ.

The motion equation of the mechanical system under analysis is presented in the form of the Lagrange equations of the second kind [4].

$$\frac{\partial}{\partial t}\frac{\partial T}{\partial \dot{q}_i}-\frac{\partial T}{\partial q_i}=Q_{qi} \qquad (i=1..4) \tag{2}$$

In accordance with equations (1) and (2), they are presented as follows:

$$M\ddot{\xi}_B-\Omega^2A_0\cos\Omega t-A_1\left(\ddot{\gamma}\sin\gamma+\dot{\gamma}^2\cos\gamma\right)-\sum_{i=1}^{2}m_ih_i\left(\ddot{\gamma}_i\sin\gamma_i+\dot{\gamma}_i^2\cos\gamma_i\right)=Q_{\xi_B};$$
$$M\ddot{\eta}_B-\Omega^2A_0\sin\Omega t-A_1\left(\ddot{\gamma}\cos\gamma-\dot{\gamma}^2\sin\gamma\right)+\sum_{i=1}^{2}m_ih_i\left(\ddot{\gamma}_i\cos\gamma_i-\dot{\gamma}_i^2\sin\gamma_i\right)=Q_{\eta_B}; \tag{3}$$
$$J_{Bi}\ddot{\gamma}_i+m_ih_i\left(\ddot{\eta}_B\cos\gamma_i-\ddot{\xi}_B\sin\gamma_i\right)=Q_{\gamma_i} \qquad (i=1,2)$$

with the following notions introduced:

$$M=m_{\Pi}+m_K+\sum_{i=1}^{5}m_i; \qquad A_0=m_3h_3+l_3(m_4+m_5+m_K);$$

$$A_1=m_4h_4+m_5h_5-m_Kh_K; \qquad J_{Bi}=J_{Ci}+m_ih_i^2 \qquad (i=1,2).$$

We believe that the generalized forces are calculated according to the following formulas [5]:

$$Q_{\xi_B}=-C_{\xi}\xi_B; \quad Q_{\eta_B}=-C_{\eta}\eta_B; \quad Q_{\gamma_i}=-h_{\gamma}(\dot{\gamma}_i-\dot{\varphi}) \tag{4}$$

The steady motion of the system is presented as follows:

$$\gamma_i=\Omega t+\theta_i, \quad \xi_B=const=0, \quad \eta_B=const=0 \tag{5}$$

Here, constants θi, specifying the equilibrium position of the pendulum in relation to the crankshaft, are calculated and subject to the condition of the platform's immobility.

At such values of generalized coordinates, the two latter simultaneous equations (3) are satisfied identically [6]. Substituting (5) in the first two simultaneous equations (3), the conditions imposed on the parameters of the system are obtained:

$$m_1 = m_2 = m, \quad h_1 = h_2 = a, \quad A_1 = 0. \tag{6}$$

And equations to determine θ_1 and θ_2 are

$$\begin{aligned} &\cos\,\theta_1 + \cos\,\theta_2 = -\frac{A_0}{ma}; \\ &\sin\,\theta_1 + \sin\,\theta_2 = 0 \end{aligned} \tag{7}$$

and let us calculate the correction mass with (6)

$$m_K = \frac{m_4 h_4 + m_5 h_5}{h_K}. \tag{8}$$

Under condition (8), the mass center of the crank rod with a slider and MK mass is reduced to point D of the swing joint with a crankshaft. This condition is obtained in [7]. In the range of variation θ_1 from 0 to π, the only solution of simultaneous equations (7) is as follows:

$$\begin{aligned} &\theta_1 = \arccos\left(-\frac{A_0}{2ma}\right); \\ &\theta_2 = -\theta_1 \end{aligned} \tag{9}$$

The arrangement of the pendulums corresponding to (9) is shown in Figure 2.

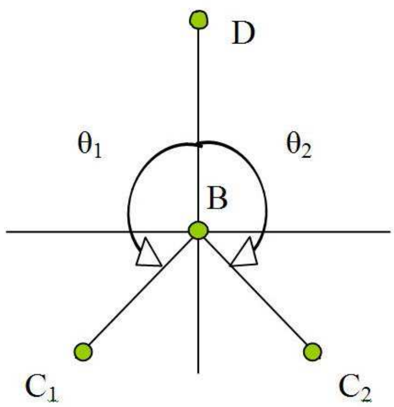

Figure 2. Arrangement of the pendulums.

It is (9) that shows what should be met

$$\frac{A_0}{2ma} \le 1 \tag{10}$$

Equations (6) and (10) are essential for the platform immobility that corresponds to balancing of the mechanism [8].

The solution obtained is realized in practice if it is stable.

The condition of solution stability (5) of simultaneous equations (3) under conditions (6) is similarly deduced as in [5] as follows:

$$\frac{\varpi^2 - \Omega}{\left(\varpi_\xi^2 - \Omega^2\right) \cdot \left(\varpi_\eta^2 - \Omega^2\right)} < 0 \quad , \tag{11}$$

where

$$\varpi^2 = \frac{\left(\varpi_\xi^2 + \varpi_\eta^2\right)}{2}, \quad \varpi_\xi^2 = \frac{C_\xi}{M}, \quad \varpi_\eta^2 = \frac{C_\eta}{M}$$

To be definite, $\omega_\xi < \omega_\eta$ is accepted, then it is easy to show that $\omega_\xi < \omega < \omega_\eta$. The whole variation range of the frequency rotation rate of the crankshaft is divided into four intervals

$$\Omega < \omega_\xi, \quad \omega_\xi < \Omega < \omega, \quad \omega < \Omega < \omega_\eta, \quad \omega_\eta < \Omega$$

(11) is not satisfied in the first and third intervals, while it is met in the second and forth intervals. Thus, there are two stable intervals corresponding to the elimination of the static unbalance of the slider-crank mechanism.

$$\omega_\xi < \Omega < \omega, \quad \omega_\eta < \Omega \tag{12}$$

Modeling of the mechanism under analysis in CAD Autodesk Inventor (Figure 3) allows solving a lot of issues that arise during operation of the tunneling machines. They may include the issues of friction, loosening of the fixtures, the strain experienced by the machine parts, etc.

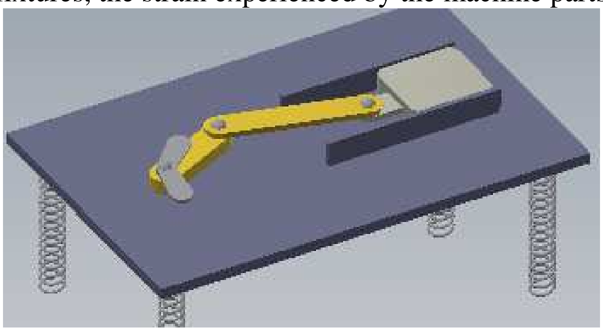

Figure 3. The model of the slider-crank mechanism.

The method, studied here for balancing the slider-crank mechanism using pendulums, is much preferred than that considered in work [7], where the balancing counterweights are used on the crankshaft and the crank rod of the mechanism, so that the mass center of the slider-crank mechanism is not moving. The method is preferred in that it is performed automatically in-process at the mass variation of the slider, the crank rod and crankshaft, as well as the coordinates of their mass centers.

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