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Optimal selection of the parameters of the kinematic chain of the main drive of the cold rolling mill

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Abstract. The article is devoted to the choice of optimal parameters of the system of dynamic balance of loads in the mechanism of the cyclic action of the drive of the working stand of the cold rolling mill.

Introduction

The balancing of dynamic loads in the main drive line of cold pilgrim rolling mills that arise as a result of large accelerations in the extreme positions of the work stand during its reciprocating motion is the main reason for limiting their speed [1-4]. The share of the main drive mechanism accounts for up to 65% of the total number of failures of cold rolling mills (CPT) [5]. The solution of the problem of increasing the structural and operational reliability requires the systematization of general patterns of changes in internal loads, the establishment of functional dependences of the conversion of power flows, and the parametric control of their values. The resulting loading of the main drive system of the mill is determined by the technological and inertial loads arising during the high-frequency reciprocating motion of the work stand [1]. When using the mechanisms of dynamic balancing, the solution of this problem is to determine their rational parameters, taking into account the phase shift of the technological and inertial components.

Depending on the size of the mills, various variants of balancing of inertial loads are used [1]: the type of the main moment balancing by placing the rotating counterweight on the drive shaft of the crank-slider mechanism; the balancing system of the stand with a mass m1, complemented by a synchronous rotating mass m_3, which allows to reduce the values of the rotating counterweights; the balancing system of the stand drive by an additional load, whose crank is offset relative to the crank of the stand by 90^0.

Let us consider the solution of the optimal choice of balancing parameters for the lever-type mechanisms on the drive crank in the vertical plane used to convert the rotational motion of the main drive shaft into reciprocating motion of the work stand (Figure 1).

The lengths of connecting rods are the same 1.75 meters. The deaxial of the balancing chain is absent, and in the chain of the stand it is 0.15 meters. The radius of the drive of the OA stand is 0.25 meters. The radius of the balancing circuit is more than 0.3 meters for design reasons.

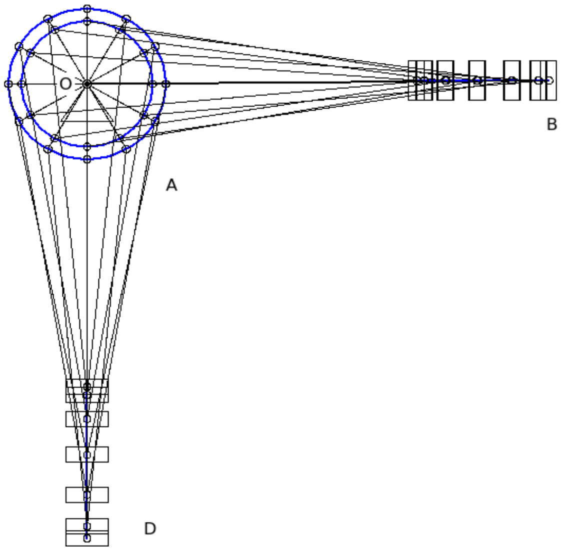

Figure 1. Kinematic chain of the main drive of the cold rolling mill

The equation of motion of the machine aggregate is represented by the equation

$$J\ddot{\varphi} + \frac{1}{2}J'_{\varphi}\dot{\varphi}^2 = M_d - M_C - M_S,$$

where

Jи J'_{φ} - The reduced moment of inertia and its derivative with respect to the angle of rotation;

$\ddot{\varphi}, \dot{\varphi}, \varphi$ - angular acceleration, angular velocity and crank angle;

M_d - motor torque;

M_C - The given moment of technological loading;

M_S - reduced moment of the weight of the balance weight.

Equalizing the torque on the crank will reduce load and speed fluctuations. Ideally, the acceleration of the crank will be assumed to be zero, and then

$$\Im = \int_0^{2\pi} (M_d)^2 d\varphi,$$

$$M_d = M_I + M_C + M_S,$$

It means $$\Im \to \min_p ,$$

where p = (m_g, beta, e, l)^T.

Here $J'_{\varphi} = \vartheta^T \Theta \vartheta_{\varphi}$ is a quadratic form in the coordinates

$$\vartheta, \vartheta^T = (\vartheta_{Bx}, \vartheta_{Dy})^T.$$

Coordinate of the stand $\vartheta_{Bx} = Rcos(\varphi) + \{L^2 - [Rsin(\varphi) - E]^2\}^{0.5}$,

Coordinate of cargo $\vartheta_{Dy} = rsin(\varphi + \beta) - \{l^2 - [rcos(\varphi + \beta) - e]^2\}^{0.5}$,

m_g– balancing weight,

beta– the difference of the turns of the cranks,

e–dexial of the movement of the counterbalancing load,

l–length of connecting rod of counterbalancing load.

Vector $\vartheta_\varphi = \dfrac{d\vartheta}{d\varphi}$ - is the derivative ϑ of the coordinate of the crank angle

The matrix of a quadratic form is diagonal $\Theta = \begin{bmatrix} m_k & 0 \\ 0 & m_g \end{bmatrix}$.

Let us consider the limiting case for cold rolling. For such a stroke length, the frequency ω=25 radians per second. The weight of the stand is 1450 kg. The given moment of the technological load will be considered as the maximum.
The interval boundaries for all parameters are chosen from the design features of the equipment.
The moment of inertia forces of the second kind M_I= has a constant phase and amplitude at a given speed and mass of the counterbalancing load, but the moment of inertia changes the phase due to the angle β and the difference between the stand mass and the counterbalance load. The moments M_C and M_S on the interval [0:2π] remain at fixed positions. Parameters are selected taking into account the previously assigned values that ensure maximum loads.

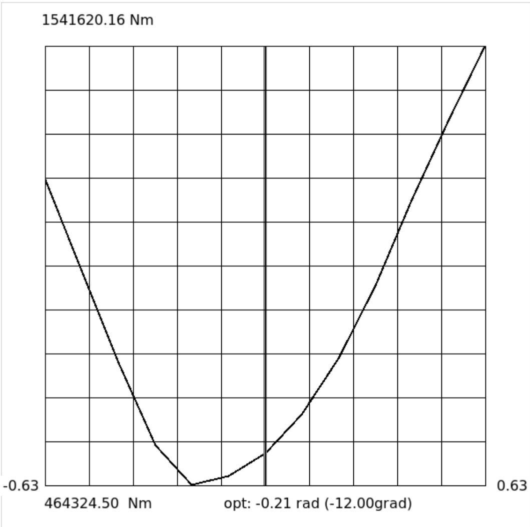

Figure 2. The dependence of the moment on the difference in the angles of rotation of the cranks of the crane drive and the balancing weight

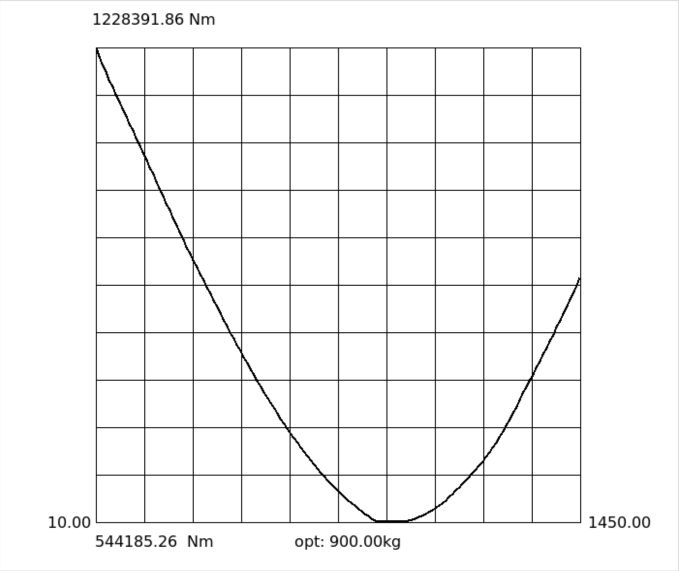

Figure 3. The dependence of the moment on the mass of the balanced cargo All the parameters of the balanced circuit are related to each other. For the values obtained, the best connecting rod length of the counterbalancing load is 1.1 meters.

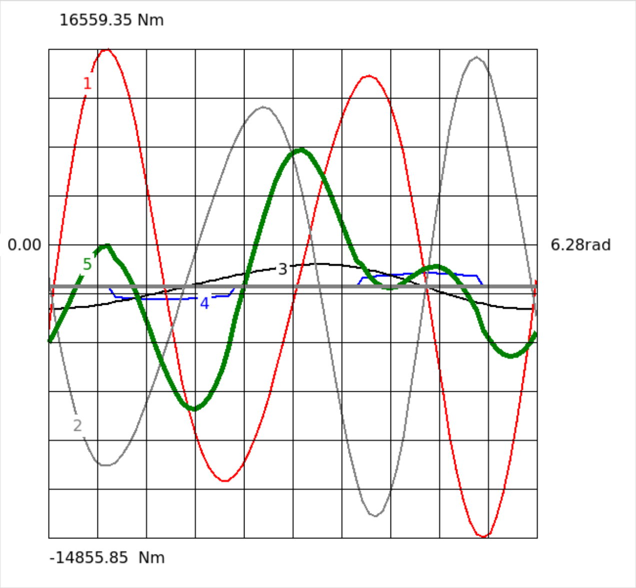

Figure 4. Kinematic diagram with the selected parameters and graphs of the moment dependencies on the selected parameters (1 is the moment of inertia of the second kind of stand, 2 is the moment of inertia of the second kind of the counterbalancing load, 3 is the static moment of the counterbalancing load, 4 is the technological moment, and 5 is the total moment).

Conclusion

All points are brought to the OA crank. Depending on the speed of the mill and the magnitude of the technological load moment, the values of the parameters change. Obviously, the weight of the counterbalancing load and the angle β had the greatest influence on the change of the engine torque.
Less other parameters are affected by the length of the connecting rod and the de-axial chain of the counterbalancing load.

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