Modeling of the converting mechanism of rocking machines

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Abstract. The transforming mechanism of rocking machines is an articulated four-link mechanism made following the symmetrical and asymmetrical kinematic schemes. On the straight line passing through the extreme positions of the base of the bolt and the balancing lever, there is a center of rotation of the crank in symmetrical mechanisms. In different variants they are asymmetrical mechanisms. The dimensions of the converting mechanism of Russian symmetrical rocking machines are smaller than foreign asymmetrical rocking machines. This method can be used to show the contrast between technical and operational indicators of the rocking machines manufactured using various kinematic schemes. A contrast of the data multiplied by $S_0K/K_1$ with the real data of rocking machines revealed their compliance, which confirms the correctness of the method.

1. Introduction
The transforming mechanism of rocking machines is an articulated four-link mechanism made following the symmetrical and asymmetrical kinematic schemes (see Figure 1) [1-3]. On the straight line passing through the extreme positions of the base of the bolt and the balancing lever, there is a center of rotation of the crank in symmetrical mechanisms. In different variants they correspond to the asymmetrical scheme.

Figure 1. The balancing rocking machine (combined balancing): 1 - electric motor; 2 - V-belt transmission; 3 - reducer; 4 - cranks; 5 - balancing loads; 6 - base of the bolts; 7 - traverse; 8 - balancing load; 9 - balancing lever; 10 - balancer head; 11 - a rope suspension bracket; 12 - the rod is polished; 13 - wellhead fittings; 14 - foundation; 15 - frame; 16 – stand
2. Materials and methods
Based on the already known method of designing a transforming symmetrical mechanism, we use kinematic relations and the ratio of the crank radius to the length of the rear arm of the balancer and base of the bolts as initial data [4-6]. When designing the asymmetrical mechanism, in addition to these parameters, the deaxial angle are used. Deaxial angle is the angle which formed by the positions of the base of the bolts corresponding to the beginning and end of the movements of the balancer head.

The practical method is to design the mechanism according to the preset output parameters. It is better to use parameters that directly determine all types of a kinematic scheme of the converting mechanism and its dimensions: the balancing angle $\delta_0$ and the deaxial angle $\theta$ [7]. Figure 2 shows a converting mechanism of the rocking machine. A circle with center at point $O_1$ of arbitrary radius $R$ is drawn through points $B_1$ and $B_2$. Any point of this circle can be considered as the point of rotation of the crank. Connecting point $C$ with points $B_1$ and $B_2$, we have $B_1C = l + r$ and $B_2C = l - r$, and the angle between these lines ($\angle B_1CB_2 = \theta$) is the deaxial angle.

![Figure 2. The calculation scheme of the converting mechanism of a rocking machine](image)

It follows from this that if you connect any point of a circle with radius $R$ with points $B_1$ and $B_2$, you get a transforming mechanism with the same degree of unevenness and the same balancing angle. At the same time, the absolute lengths of the base of the bolts and crank and overall dimensions are different. It can be seen from Figure 1

$$\left(\angle B_1CB_2 = \angle B_1DB_2 = \frac{1}{2}B_1B_2 = \theta \right)$$ (1)

Therefore, the angle $\theta$ and the angle $\delta_0$, moves along a circle passing through points $B1$ and $B2$. They correspond to the extreme positions of the base of the bolts to the balancing lever.

Let us call this circle a circle of equal deaxials. The center of this circle is on the bisector of the balancer swing angle: for rocking machines with a positive deaxial, it is on the opposite side relative to the line $B_1B_2$ from the balancer support; for machines with a negative deaxial, it is on the balancer support.

Let’s call this circle a circle of equal parts. The center of the circle is located on the bisector of the rotation angle of the balancer lever. It is located on the opposite side relative to the line $B_1B_2$ from the balancer support. For swinging machines with a positive deviation from the axis and for machines with a negative deviation from the axis, it is located on the balancer support.

According to the initial data, it is also possible to determine the parameters of a circle of equal desaxials [8]. Since the angle $\angle B_1O_1B_2$ is central, using equality (1), we have
\[ \angle B_1O_1B_2 = 2\theta \]  

Then for a circle with equal parts, you can determine the radius of the triangles OB_1B_2 and O_1B_1B_2 by formula:

\[ R = K \frac{\sin \delta_0}{\sin \theta} \]  

Then the coordinates of the center of the circle are

\[ x_0 = K \cos \frac{\delta_0}{2} + R \cos \theta = K \frac{\sin \left( \theta + \frac{\delta_0}{2} \right)}{\sin \theta}; \]
\[ \phi_0 = 0 \]  

In parametric form, the center of rotation coordinates of the cranks with the same angle of deviation can be expressed as follows:

\[ x_c = x_0 - R \cos (\theta + \psi); \]
\[ y_c = R \sin (\theta + \psi) \]  

where \( \psi \) is the angle between radius \( R \) and its real position \( O_1B_1 \).

The length is a horizontal line from the center of rotation of the crank to the support of the balancer. The vertical line is the height of the converter. Figure 1 shows that the abscissa and the ordinal of the center of rotation (5) are the length and height. From the isosceles triangle \( B_1O_1C \) and \( B_2O_1C \) we have

\[ l = 2R \cos \frac{\theta}{2} \sin \frac{\theta + \psi}{2}; \]
\[ r = 2R \sin \frac{\theta}{2} \cos \frac{\theta + \psi}{2}. \]  

From equations (3) and (6) we can obtain expressions for determining the kinematic relations:

\[ \frac{r}{K} = \frac{\sin \frac{\delta_0}{2} \cos \frac{\theta + \psi}{2}}{\cos \frac{\theta}{2}}; \]
\[ \frac{r}{l} = \frac{\sin \frac{\theta}{2} \cos \frac{\theta + \psi}{2}}{\sin \frac{\theta}{2}}; \]
\[ \frac{r}{P} = \frac{r}{P_K}. \]  

The pole distance can be determined from triangle \( OO_C \):

\[ \frac{P}{K} = \frac{\sin \left( \theta + \frac{\delta_0}{2} \right)}{\sin \theta} \sqrt{1 + \frac{\sin^2 \frac{\delta_0}{2}}{\sin^2 \left( \theta + \frac{\delta_0}{2} \right)} - 2 \frac{\sin \delta_0 \cos (\theta + \psi)}{\sin \left( \theta + \frac{\delta_0}{2} \right)}. \]  

Using the values of the kinematic relations (7) and using formulas [9], it is possible to know all the angles between the links of the converting mechanism, as well as movement, speed and acceleration of the suspension point of the rocking machine bars.

As can be seen from expressions (6) - (8), the length of the links and the length ratios depend on the source data and \( \psi \). To get the mechanisms, it is necessary to determine the limit values of \( \psi \).

\[ \frac{P}{K} < 1 + \frac{1 - \frac{r}{T}}{r}; \]
\[ \frac{P}{K} < 1 + \frac{1 - \frac{r}{T} \frac{r}{K}}{K}. \]  

Taking into account the system of equations (7), we have
From equations (8) and (10), we can obtain the condition for the existence of a mechanism:

\[
1 + \frac{\sin \frac{\delta_0}{2} \sin \frac{\psi}{2}}{\sin \theta} > \frac{\sin \left( \theta + \frac{\delta_0}{2} \right)}{\sin \theta} \sqrt{1 + \frac{\sin^2 \frac{\delta_0}{2}}{\sin^2 \left( \theta + \frac{\delta_0}{2} \right)} - \frac{2 \sin \frac{\delta_0}{2} \cos(\theta + \psi)}{\sin \left( \theta + \frac{\delta_0}{2} \right)}}.
\]

By solving this inequality, we can determine the area of angles \( \psi \) caused by the rotability of the mechanism:

\[
0 < \psi < 180^\circ - (2 \theta + \delta_0)
\]

This ratio can be obtained from triangle \( O_1DB_2 \) (Fig. 2), since \( \angle O_1B_2D = \angle O_1DB_2 = \theta + \delta/2 \).

Not all theoretically real mechanisms can be practically feasible [11]. Therefore, the real area of angles \( \psi \) is theoretically real (3) and should be determined by design features of the mechanism [20-22].

Given that, we present this area of angles \( \psi \) (8)

\[
\arctg \left( \frac{tg \frac{\theta}{2}}{\left( \frac{T}{L} \right)_{max}} \right) < \frac{\psi + \theta}{2} < \arctg \left( \frac{tg \frac{\theta}{2}}{\left( \frac{T}{L} \right)_{min}} \right)
\]

The limit values of \( \psi \) are determined by the maximum and minimum exponents of the equation \( r/l \).

Knowing the design features of the characteristics and design experience, it is recommended to use kinematic and dynamic characteristics and design experience, it is recommended to use

\[
\left( \frac{T}{L} \right)_{max} \text{ and } \left( \frac{T}{L} \right)_{min}
\]

3. Results and Discussion

Thus, the real range of parameter values is based under the accepted design constraints by the deaxial angle

\[
\arctg \left( 2.5tg \frac{\theta}{2} \right) < \frac{\theta + \psi}{2} < \arctg \left( 5tg \frac{\theta}{2} \right)
\]

The analysis showed that the boundaries of the angle area (Figure 3) can be calculated by the following linear dependence

\[
1.5 < \frac{\psi}{\theta} < 3.75
\]

from which \( \psi_{min} / \psi_{max} \).

Given different values of \( \psi \) within the specified limits (15) and using formulas (7) and (8), we can calculate values of the converting mechanism having the same balancer angles and degree of unevenness, but different link lengths, dimensions and masses. Having compared the characteristics, we can choose the most compact mechanism.
Figure 3. The dependence of angle $\psi$ on angle of deaxial $\theta$ of the rocking machine

To use this method, it is better to represent the required values in stroke fractions [12]. The rear arm of balancer $K$, crank radius $r_0$, rod length $l_0$, pole distance $P_0$, length $L_0$ and height $H_0$ of the converter can be calculated as following:

\[
\begin{align*}
K &= \frac{1}{\delta_0}; \\
r_0 &= \frac{\sin \frac{\delta_0}{2} \cos \theta + \psi}{\delta_0 \cos \frac{\theta}{2}}; \\
l_0 &= \frac{\sin \frac{\delta_0}{2} \sin \theta + \psi}{\delta_0 \sin \frac{\theta}{2}}; \\
P_0 &= \frac{\sin \left(\theta + \frac{\delta_0}{2}\right)}{\delta_0 \sin \theta} \sqrt{1 + \frac{\sin^2 \frac{\delta_0}{2}}{\sin^2 \left(\theta + \frac{\delta_0}{2}\right)} - 2 \frac{\sin \frac{\delta_0}{2} \cos (\theta + \psi)}{\sin \left(\theta + \frac{\delta_0}{2}\right)}}; \\
L_0 &= \frac{\sin \left(\theta + \frac{\delta_0}{2}\right) - \sin \frac{\delta_0}{2} \cos (\theta + \psi)}{\delta_0 \sin \theta}; \\
H_0 &= \frac{\sin \frac{\delta_0}{2} \sin (\theta + \psi)}{\delta_0 \sin \theta}.
\end{align*}
\] (17)

To find out the absolute length of the links for any stroke length of the swinging machine, the data obtained by formulas (17) should be used by $\frac{S_0}{K_1}$ (S$_0$ is the stroke length of the suspension point of the rods, m; $K_1$ is the front arm of the balancer).
Using these formulas and data from catalogs, the resulted values of the kinematic parameters of transforming mechanisms can be determined (see Table 1).

**Table 1.** Kinematic parameters for various kinematic schemes of transforming mechanisms of rocking machines

| Kinematic parameters | Symmetrical scheme | Asymmetrical scheme * |
|----------------------|--------------------|----------------------|
| $\delta$ | 57.3 | 45/50 |
| $\theta$ | 0.00 | 2.5/11 |
| $\psi$ | 0.00 | 5.46/23.56 |
| $r/K$ | 0.4794 | 0.38/0.405 |
| $r/l$ | 0.4 | 0.31/0.31 |
| $K$ | 1.0 | 1.273/1.146 |
| $r_0$ | 0.4794 | 0.486/0.486 |
| $L_0$ | 1.1985 | 1.55/1.50 |
| $H_0$ | 1.1985 | 1.546/1.44 |
| $L_0$ | 0.8776 | 1.273/1.44 |

Note: * numerator data provided by Lufkin company, in the denominator - by Bethlehem company (USA)

4. Conclusion

As can be seen, the overall dimensions of the converting mechanism of Russian symmetrical rocking machines are smaller than those of foreign asymmetrical rocking machines (length by 45–60% and height by 25–30%). The method based on formulas (17) can be used to compare technical and operational indicators of two types of rocking machines. The comparison of the data, multiplied by $S_0 K / K_1$, with the actual data of rocking machines showed their complete compliance, which confirms the correctness of the method.

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