An investigation of the influence of cam track manufacturing accuracy on the dynamics of automatic machine mechanisms

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Abstract. The article deals with the application of dynamic analysis methods to study the efficiency of cam mechanisms depending on the accuracy of manufacturing their elements. It is commonly known that the successful choice of the cam track contour law significantly determines the level of dynamic loads in the mechanism and, hence, its reliability and efficiency. However, all efforts to create the “best” cam track contour law can be nullified by such a factor as the error in manufacturing its elements (primarily cams). The proposed paper analyzes the influence of the accuracy of manufacturing cam tracks for the transfer mechanism of the automated cold-upsetting machine contoured according to the sinusoidal, cosinusoidal and uniformly accelerated laws. Research was carried out for various speed modes of the mechanism operation. A technique for estimating the calculation of the error function parameters at which the dynamic processes in the mechanism increase sharply is given.

1. Dynamic processes in high-speed mechanisms and their capability
The links of a mechanism have mass and the ability to deform. Their movement results in oscillatory (dynamic) processes arising in the mechanism, which leads to a violation of the accuracy of positioning its elements, in dynamic forces whose values differ significantly from the estimated ones, and, ultimately, in reliability decrease and the mechanism malfunction [1-4]. The level of dynamic processes increases as the operating speed of the mechanism, the gaps in kinematic pairs and their manufacturing error increase [1, 5, 6].

2. Evaluation of the accuracy of manufacturing cam tracks in automatic machine mechanisms
As a rule, the drives of the mechanisms that perform transport operations in modern high-speed automatic machines include cam mechanisms facilitating the movement of the output link following the instroke – dwell – backstroke – dwell scheme. Interrelation can be traced between the efficiency of the mechanism design and such its parameters as the law of cam track contour and the accuracy of its application [1, 3, 6, 7].

Obviously, the contour distortion function is random in nature and in practice, for a ready-made cam, can be obtained by measurements. Such an approach, however, becomes meaningless in the dynamic research of the mechanism at the design stage. The task of investigating the dynamics of the mechanism, with account of the cam contour manufacturing error, consists in determining such characteristics of the change in the distortion function – within the tolerance for cam manufacture – at which the level of dynamic processes in the mechanism reaches its maximum. This makes it possible to obtain one more criterion for cam rejection (besides the production tolerance), as well as to implement the algorithm for
studying oscillatory processes in a cam mechanism whose contour is manufactured with a specified error.

\[ U(\phi_0) = \overline{U}(\phi_0) + \chi(\phi_0). \]  

In expression (1) \( \phi_0 \) is the angle of cam rotation, \( \overline{U}(\phi_0) \) is the function describing its track ideal contour, \( \chi(\phi_0) \) is the harmonic error variation function:

\[ \chi = \chi_{\text{max}} \sin k\phi_0 + \chi_0, \]  

Figure 1. Sectional drawing of the transfer mechanism instroke cam contour KVA AV1818, where are 1 – theoretical contour, 2 – actual ideal contour, 3 – actual contour with account of error: (a) actual contour for \( k = 5 \), (b) – for \( k = 20 \).

4. Dynamic and mathematical models of the mechanism

The intensity of the dynamic processes in the mechanisms of automatic cold upsetting machines in the time interval \( 0 \leq t \leq T \) can be estimated on the basis of the following criteria [6-8]:

Accuracy and root-mean-square value of the actuating link positioning accuracy:

\[ \delta = \max |x_p(t) - \overline{x}_p(t)|, \]  

Figure 1 shows the possible contour deviations from the ideal one for different values of \( k \) for the instroke cam of the transfer mechanism drive KVA AV1818. The cam contour tolerance field corresponds to \( h7 \) (from 0 to -40 microns), but by way of illustration the figure is enlarged by a factor of 100.

Thus, the research task is to find the values of the coefficient \( k \) at which the dynamic processes in the mechanism increase sharply while it operates with different efficiency and to determine to what extent the error in the cam track manufacture under various laws for its contour affects the dynamics of the mechanism.
\[ \tilde{\delta} = \sqrt{\frac{1}{T} \int_0^T (x_n - \bar{x}_n)^2 dt} . \]  

(4)

Maximum acceleration value of the actuating link:

\[ \ddot{x}_{\text{max}} = \max |\ddot{x}_n(t)| . \]  

(5)

Maximum and root-mean-square deviations of the actuating link acceleration from ideal values:

\[ \nu = \max |\ddot{x}_n(t) - \ddot{\bar{x}}_n(t)|, \]  

\[ \hat{\nu} = \sqrt{\frac{1}{T} \int_0^T (\ddot{x}_n - \ddot{\bar{x}}_n)^2 dt} . \]  

(6)

(7)

Maximum and root-mean-square loads in the mechanism links:

\[ q_i = \max |Q_i(t)|, \]  

\[ \hat{q}_i = \sqrt{\frac{1}{T} \int_0^T Q_{i}^2(t) dt} . \]  

(8)

(9)

Formulas (3) – (7) \( x_n, \ddot{x}_n, \dddot{x}_n \) respectively show the real and ideal (without taking into account the elasticity of the links, the accuracy of their manufacture and the gaps in the kinematic pairs) movement and acceleration of the actuating (\( n \)-th) link of the mechanism. It is assumed that this link is moving forward. \( Q_i(t) \) in expressions (8), (9) is the load in the \( i \)-th link of the mechanism which can be found as the product of its rigidity by deformation.

Figure 2 (a) shows the drive of the workpiece transfer system between the \( KVA \ 1818 \) stamping positions, Figure 2 (b) shows its dynamic model. The principles underlying the dynamic model and its mathematical description (10) are given in [5, 7, 9, 10]. To calculate the elastic-inertial parameters, the solid-state modeling methods are applied [11, 12].

\[
\begin{align*}
J_1 \dddot{\varphi}_1 &= -Q_1 + R_{b1} + Q_{21} - R_{b21} + Q_3 - R_{b3} \\
J_2 \dddot{\varphi}_2 &= -Q_2 + R_{b2} - Q_{21} + R_{b21} \\
J_3 \dddot{\varphi}_3 &= -Q_3 + R_{b3} + Q_4 U_4' - R_{b4} U_4' \\
J_4 \dddot{\varphi}_4 &= -Q_4 + R_{b4} + Q_5 U_5' - R_{b5} U_5' \\
m_5 \dddot{x}_5 &= -Q_5 + R_{b5}
\end{align*}
\]  

(10)

The equations (10) show the loads and forces of energy dissipation in the mechanism links and its kinematic pairs:

\[
\begin{align*}
Q_i &= c_i (\varphi_i - U_i) & Q_3 &= \bar{c}_3 (\varphi_3 - \varphi_1) & Q_i &= c_i (x_i - U_i) \\
R_{bi} &= -b_i (\dot{\varphi}_i - U'_i \dot{x}_{i-1}) & R_{b3} &= -b_3 (\dot{\varphi}_3 - \dot{\varphi}_1) & R_{bi} &= -b_i (\dot{x}_i - U'_i \dot{x}_{i-1}) \\
Q_{21} &= \bar{c}_{21} (\varphi_2 - \varphi_1) & R_{b21} &= -b_{21} (\dot{\varphi}_2 - \dot{\varphi}_1).
\end{align*}
\]  

(11)
Figure 2. Cam-lever drive of the transfer mechanism KVA 1818 (a) and its dynamic model (b): 0 – cam block, 1 – instroke lever, 2 – backstroke lever, 3 – instroke lever upper arm with the gear sector, 4 – crank with toothed wheel, 5 – transfer carriage.

The stiffness coefficients of the links with account of gaps in the kinematic pairs (12) – (14) in equations (10) are:

\[
c_i = \begin{cases} 
\bar{c}_i + \frac{\eta_i}{(\varphi_i - U_i)} & \text{when } \varphi_i - U_i \leq -\eta_i \quad i = 1, 2; \\
0 & \text{when } \varphi_i - U_i > -\eta_i \\
\bar{c}_i + \frac{\eta_i}{(\varphi_i - U_i)} & \text{when } \varphi_i - U_i \leq -\eta_i \quad i = 4; \\
0 & \text{when } -\eta_i < \varphi_i - U_i \leq +\eta_i \\
\bar{c}_i + \frac{\eta_i}{(\varphi_i - U_i)} & \text{when } \varphi_i - U_i > +\eta_i \\
\bar{c}_i + \frac{\eta_i}{(x_i - U_i)} & \text{when } x_i - U_i \leq -\eta_i \\
0 & \text{when } -\eta_i < x_i - U_i \leq +\eta_i \quad i = 5 \\
\bar{c}_i + \frac{\eta_i}{(x_i - U_i)} & \text{when } x_i - U_i > +\eta_i.
\end{cases}
\]

Here, \(\eta_i\) is a one-way gap in the link joint of the mechanism, \(\bar{c}_i\) is the rigidity of the corresponding link, \(b_i\) is the energy dissipation coefficient.

It is not always reasonable to apply all the above criteria simultaneously in the study of the mechanism dynamics. However, speaking about the study of oscillatory processes in the mechanism, their minimum set must include quantities containing the root-mean-square value of a kinematic parameter and, possibly, its maximum value. In this particular case, the variation in the actuating link (transfer carriage) acceleration is chosen as such a parameter.
4. Results of estimating the influence of cam contour parameters on the mechanism dynamics

Dynamic studies of the transfer mechanism drive were conducted in the dam system [4] under the assumption that the dimensions of its elements lie within the manufacturing tolerance, i.e., their wear is minimum and the driving link (cams) rotation speed is constant.

Figure 3 shows what the functions $\ddot{x}_n(t)$ and $\dddot{x}_n(t)$ look like both in the instroke section and in the upper dwell section for different values of $k$ in the contour error expression (2). The cam track on the climbing section is configured according to the symmetric sinusoidal law of acceleration variation. The rotation speed of the cams is 100 rpm.

![Figure 3](image)

**Figure 3.** Acceleration (real and ideal) of KVA 1818 transfer carriage for different values of $k$.

![Figure 4](image)

**Figure 4.** Values of the root-mean-square deviation of the KVA AB1818 transfer carriage real acceleration from the ideal one for various laws of cam contour and their different rotation speeds:
1 – 120 rpm, 2 – 100 rpm, 3 – 80 rpm.

As is seen from the data analysis shown in figure 3, the nature of the error function $\chi$ variation can influence significantly the dynamic processes occurring in the mechanism and at certain frequencies of its harmonics lead to a sharp increase in the latter.

The results of estimating the level of dynamic processes in the drive of the workpiece transfer mechanism using the criterion (6) – the root-mean-square deviation of the real actuating link acceleration value from the ideal one – are shown in figure 4. If we use $\hat{k}$ to designate the value of $k$ at which the function reaches its maximum, and $n$ to designate the cam rotation speed, then the dependence $\hat{k}(n)$ will be close to a linearly decreasing one (figure 5) and practically independent of the cam contour law.
Figure 5. Dependence of the critical frequency of the cam track error function on its rotation speed.

The analysis of this dependence makes it possible to determine the zone of critical values of the $k$ parameter of the cam track error function for a given speed interval of equipment operation. For the cams of the $KVA\ AV\ 1818$ transfer system, the frequency of the error function $\chi$ must not lie in the range from 10 to 35 when the machine operates with passport performance.

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