Kinematic analysis of crank -cam mechanism of process equipment

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Abstract. This article discusses how to define the kinematic parameters of a crank-cam mechanism. Using the mechanism design, the authors have developed a calculation model and a calculation algorithm that allowed the definition of kinematic parameters of the mechanism, including crank displacements, angular velocities and acceleration, as well as driven link (rocker arm) angular speeds and acceleration. All calculations were performed using the Mathcad mathematical package. The results of the calculations are reported as numerical values.

1. Introduction

Domestic enterprises in various industries have many items of process equipment, including a lot of high-complexity machines, such as looms.

There are two ways to continuously feed the weft to looms: shuttle and shuttle-less. In the first case, the weft threads are fed to the shed from the weft package which is placed in the shuttle and moves with it. In shuttle-less looms the weft is fed from a package outside shed.

The most progressive way of feeding the weft shuttle-less looms is the way to weft the reels of the cross-rewind with the metering device (weft detaining device).

The change of color on shuttle-less looms occurs with the help of a special weft changing mechanism. On the STB shuttle-less looms this mechanism is based on the principle of transferring the motion from the maltese-cross mechanism to the cam.

It is one of the main mechanisms limiting the productivity of the loom and belongs to the step-by-step rotation mechanism (SRM), which form one of the most advanced groups of automatic machines cyclic mechanisms.

Improving the existing and creating new high-performance reliable and durable equipment is one of the major challenges in the development of modern machinery. Such equipment includes shuttle-less looms. The widespread introduction of these into the textile industry has identified a number of requirements for improvement: firstly, there is a need to review the requirements for the individual mechanisms of the machines, both with regard to their functionality and reliable performance; second, the machines must be versatile, capable of producing a wide range of fabrics. Mechanisms that limit the performance of this machinery include the color changing mechanism. The end effector of this mechanism should ensure that the mechanism is rotated to its original position in the fourth part of the machine main shaft revolution which ranges from 0.075 to 0.0375 seconds. This creates an increased dynamic mode for the mechanism. In addition, the end effector closure is looped by the spring, whose
behavior requires further investigation. These mechanisms are designed to transform the uniform rotary movement of the driving link into the intermittent rotation of the driven link. The step-by-step rotation mechanism should ensure that the driven link is surgelessly rotated to a specified angle with sufficient precision and according to the cyclical graph of the automatic machine.

One of the most important requirements for modern machines is that the driven gears are required to perform displacements that are in accurate correspondence with the law. This requirement is sometimes not feasible if simple part connections are being used. Therefore, links with different contour surfaces are implemented in these mechanisms. These are cams, i.e. irreversible or high kinematic pairs.

The aim of this research is to investigate the kinematic structures of the STB loom color changing mechanism.

2. Materials and methods

The authors created a design model whose sequence starts with a crank to perform a kinematic analysis of the color changing mechanism (Figure 1). It is connected by two pinion gears (with a transmission ratio equal to 1.5) with a cam having contact with a roller located on the rocker arm. The rotary motion from the loom's stackable shaft (1) is transmitted by chain transmission to the star wheel (2) where rollers (3) are placed. The roller interacting with the grooves transmits the rotary motion of the maltese cross (4) from which the toothed gear and the star wheel (2) transmit it to the Carton route. Carton is a set of different profile plates connected by bushings in one common sequence. The thread holders with different colors thread weft are situated on the axis (5). At a certain point of time, the profile of the cam on the sequence link interfacing with the roller deflects the thread holder arm (5) at a specific angle and thus takes up a new position, and with it, the thread holder bracket takes a new position with the other weft.

![Figure 1. Design of the two colors changing mechanism.](image)

A number of sources [1-5] state that the analysis of such mechanisms is performed sequentially, from the drive unit to the closing link. The first step is to define the kinematic parameters of the driving mechanism (crank) then, moving sequentially, the kinematic properties for the cam are defined, and only then those of the driven link (rocker arm). The authors propose studying the mechanism starting with defining the coordinate position at the sequence beginning (maltese mechanism) to the end effector which represents the mechanism of the cam type with the force closure. At the beginning of the research, let us assume that the angular velocity of the driving link is a constant value equal to 1. In this case (see Figure 2), for the mechanism kinematic analysis, one finds the coordinates of characteristic points for the entire mechanism, and then process the received values with splines to find their velocity and speed. The X axis is directed along the calculation diagram by connecting the crank centers and the rotation axes of the pinion gears, the cam, and the rocker arm. Moving along the calculation diagram, let us define the coordinates of point B (see Figure 2) (the center of the crank roller).
Figure 2. A kinematic scheme of the roller cam mechanism

\[ BX(\varphi) = l_1 \cdot \cos(\varphi) \]
\[ BY(\varphi) = l_1 \cdot \sin(\varphi) \]
\[ BX_1(\varphi) = l_2 - BX(\varphi) \]
\[ BY_1(\varphi) = BY(\varphi) \]
\[ \alpha(\varphi) = \arctan \left( \frac{BY_1(\varphi)}{BX_1(\varphi)} \right) \]

The first two expressions (1) allow defining the coordinates of the roller for the crank, depending on its rotation angle, and the third and fourth expressions define the crank coordinates. The last expression defines crank rotation angles depending on the rotation angle of the crank. If the crank is connected to the cam by toothed gear, the rotation angle of the cam equals to the crank rotation angle divided by the transmission ratio of that pair (I).

\[ \beta(\varphi) = \frac{\alpha(\varphi)}{I}, \]

The rocker arm rotation angle is determined by the expression (see Figure 2)

\[ \gamma(\varphi) = \arctan \left( \frac{EY_1(\varphi)}{EX_1(\varphi)} \right), \]

where

\[ EX_1(\varphi) = l_3 - EX(\varphi) \]
\[ EY_1(\varphi) = EY(\varphi) \]
\[ EX(\varphi) = \rho_1(\varphi) \cdot \cos \beta(\varphi) \]
\[ EY(\varphi) = \rho_1(\varphi) \cdot \sin \beta(\varphi) \]

The derivative of the expression (2) defines the angular velocity and acceleration of the cam.

Knowing the of rotation angles of the link that acts as a cam, the crank rotation angle is found in accordance with the expressions (3) and (4). Since the computations are voluminous, a mathematical package of Mathcad is used to determine the kinematic characteristics.

The methodology for the kinematic study is as follows: - the authors received the E point acceleration by double differentiating expression (3); - the authors created a matrix of values consisting of two columns consisting of the cam rotation angles and the corresponding angular acceleration values.

As a result of sorting the matrix in ascending and descending order, a new array of values was obtained for the acceleration of the driven link (rocker arm). It is important that sorting and scaling the angular acceleration values matrix is performed prior to integration. As a result, the array of values for
angular acceleration is received. The array integration yields the speeds equivalents [6-13].

3. Results and discussion

The previous section provides a general methodology for defining the kinematic characteristics of a crank-cam mechanism. To consider defining the kinematic characteristics of the color changing mechanism of the STB shuttle-less looms it is necessary to define its design model, i.e. a crank-cam mechanism presented in Figure 2 which includes: crank AB, roller BC, pinion gears with the number of teeth \( Z_1 = 28 \) and \( Z_2 = 42 \). On the pinion gear \( Z_2 \) axis there is a rigidly anchored star wheel on which the Carton route is placed, the connecting links of which being executed as cams. Design data are: \( l_1 = 71 \) mm; \( l_2 = 100 \) mm; \( l_3 = 57 \) mm.

The radius of the cam vector is taken from the manufacturer's technical documentation and is not listed in the work [14-16]. Figure 3a shows the graph for changing vector radii based on the rotation angle of the main shaft of the machine.

The kinematic properties definition begins by defining the coordinates of the mechanism's characteristic points. The origin of the coordinates is positioned on the rotation axis of the star wheel. The coordinates for the center of the roller are marked by B; the center coordinates for the pinion gear and the wheels are placed on the X axis and marked by C and D. The center of the crank roller is marked by E. The crank axis of rotation is set at point F and is placed on the X axis.

Previously mentioned formula (2) is used to define the kinematic characteristics of the crank. Angular velocity and acceleration of the crank are shown in Figure 3b.

![Figure 3](image.png)

**Figure 3.** Source data: a - change of the cam vector radius for the Carton route; b - graphs of crank kinematic characteristics equivalents

The kinematic diagram presented in Figure 2 assumes that the cam, which is the link of the Carton route, rotates with variable angular velocity and acceleration. The angular velocity value for the Carton route is based on the dependency (2). The curve of the graph is the same as in Figure 3b, however, the amplitude value is 1.5 times smaller. Expression (3) is differentiated twice to define the rocker arm angular acceleration.

Figure 4a shows the equivalents of the rocker arm angular acceleration which is of uneven character and is not suitable for further processing. Therefore, it requires processing using a matrix of values. It includes 41 lines and two columns; the rotation angles are represented by integers from 0 to 40, and the ordinates are filled with the amplitude values of the acceleration curve. The graph is then scaled so that the positive and negative areas are equal or comparable in size. The following graph layout along the abscissa (in degrees) is proposed: 0 -10; 10 – 20; 20 - 30; 30 – 40. The result is a new, red-colored graph that has been scaled according to two criteria. This creates a new matrix with a smooth acceleration graph. The scale factor for the first interval was \((0 -10; 10-20) \mu = 1\); for the second interval \((20 -30; 30-40) \mu = 0.998\).
The graph presented in Figure 4b is the result of the integration. It can be observed that it is symmetric on the abscissa axis with positive and negative values correlating in amplitude.

Figure 4. Numerical simulation results: a - angular acceleration equivalents graphs; b - angular velocity equivalents

4. Conclusion
Using the industrial design of the STB shuttle-less loom's color changing mechanism, the authors have developed a design model which comprises a crank-cam mechanism including a crank that replaces a maltese cross. Toothed gear is provided for by the contact ratio of this pair, and the elements of the Carton route are represented by the cam. A rocker arm is used as a driven link.

Conclusions
1. Using mechanism's geometric parameters and its design, the authors proposed the algorithm for kinematic analysis of the crank-cam mechanism in the Mathcad mathematical package.
2. The authors defined angular displacement, equivalents of rocker angular velocities and accelerations depending on the rotation angle of the crank: the maximum angular velocity is 2.44 s\(^{-1}\) and the maximum angular acceleration is 5.5 s\(^{-2}\).
3. The displacement of the driven link (rocker arm) was 4 mm, equivalent to the maximum angular velocity of 0.24 s\(^{-1}\); angular acceleration equivalent -1.5 s\(^{-2}\).
   The angular velocity and acceleration of driven link (rocker arm) should be taken into account for further dynamic research.

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