

# Geometric modeling of controlled third-class hinged mechanisms with a stand in one extreme position for cyclic automatic machines

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**Abstract.** The geometric model for the synthesis of third-class lever mechanisms is proposed, which allows, by changing the length of the auxiliary link and the position of its fixed hinge, to rearrange the movement of the working organ onto the cyclograms with different predetermined dwell times. It is noted that with the help of the proposed model, at the expense of the corresponding geometric constructions, the best uniform Chebyshev approximation can be achieved at the interval of the standstill.

## 1. Introduction

Cycle machines are widely used in a number of industries, in particular, machine-building, printing, food, textile and others [1, 2]. As a rule, such automatic machines have high productivity, provided by the speed of their executive mechanisms. The automatic machine composition usually includes a sufficiently large number of executive mechanisms, which requires strict coordination of the movements of their working organs in time and space.

## 2. Formulation of the problem

One of the most promising executive mechanisms for the implementation of fast-flowing technological processes by cyclic automatic machines are lever mechanisms that have a high load capacity and the possibility of a purposeful conversion of motion and transfer it to sufficient distances from the leading link to the working body within the machine. Lever mechanisms in such industries as printing and food can be considered as an alternative to cam mechanisms, which until now are quite common actuators of automatic machines with an unchanging cyclogram.

One of the most significant problems of cyclic automatic machines is their relative "rigidity" in the transition to a slightly different process from the previous process, associated with changes in the size of the manufactured product, the duration of individual operations, etc., and, consequently, with the corresponding change cycle of operation of individual actuators.

In the works [3-5], the methods of synthesis of hinged lever mechanisms were considered that ensure a given duration of the working element's standing in one and at its two extreme positions. There are also articles that deal with the issues of adjusting the length of stay and the amount of movement of the working body under certain conditions [6, 7].



It should be noted that geometric modeling is widely used in solving general problems of analysis and synthesis in the theory of mechanisms and machines [8-12], especially for lever (rod) mechanisms [9, 11].

In this paper, we propose a constructive method for geometric modeling of third-class hinged link mechanisms according to the Assur-Artobolevsky classification, oriented to obtaining such a displacement of the working part of a cyclic automatic machine, which has an approximate stopping of the output link in one of the extreme positions. In this case, such a stop (delay) occurs at the time points specified by the cycle and has a predetermined duration. In addition, the proposed method makes it possible to synthesize such mechanisms for the realization of a family of cyclograms formed by a number of fixed preset values for the duration of an organ working in one of its extreme positions.

### 3. Methods of solution

The geometric design of the lever mechanism of the third class for cyclograms with several fixed dwell times should be carried out in two stages.

The initial basic stage should be considered as the synthesis of the initial mechanism corresponding to some average duration of the standstill (to some intermediate specified value of the crank angle for the interval of the standstill). During this stage, all the necessary kinematic parameters of the lever mechanism are determined, as well as the initial length of the auxiliary link and the position of its fixed hinge on the rack of the machine.

The second stage consists in graphically constructing some additional points of the mechanism with other (smaller and larger) fixed values of the crank angle for the intervals of stability and allowing to determine the corresponding length and position of the fixed hinge of the auxiliary link.

The geometrical construction of the mechanism will be carried out in relative units of length, assuming the crank length equal to one. This will allow us to obtain more general solutions of the problem of synthesis of the mechanism and by scaling to integrate the obtained kinematic scheme into the design of the machine.

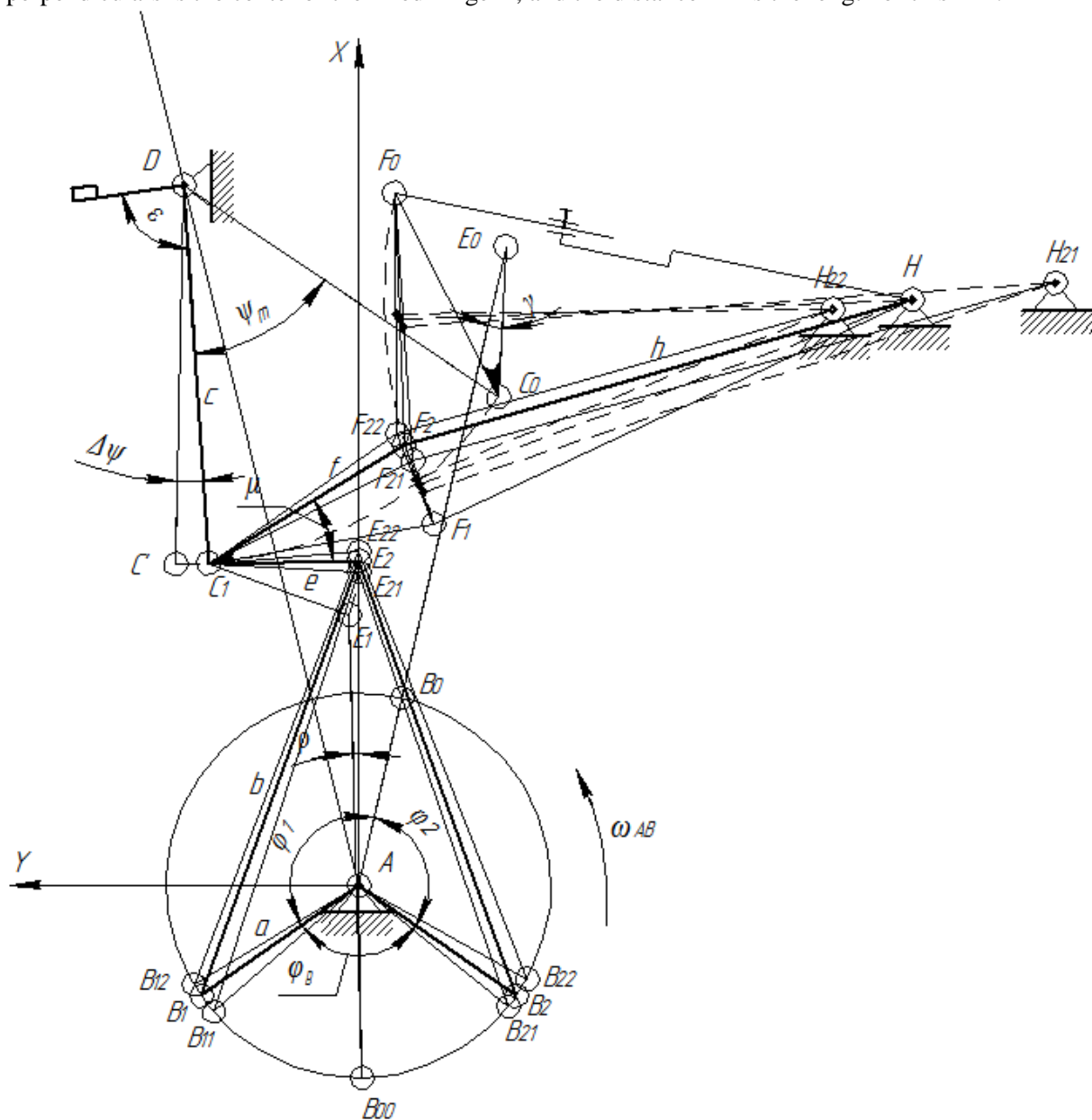
*First phase.* From the center A of the crank AB rotate the circle with the radius  $a = l$  (figure 1), draw the crank in the initial position  $AB_0$  and in the positions  $AB_1$  and  $AB_2$  respectively at the angles  $\varphi_1$  and  $\varphi_B$ , laid counterclockwise (here  $\varphi_1$  and  $\varphi_B$  respectively the crank rotation angles for the interval working stroke and for the interval of standstill). Recall that in the first step, one of the prescribed intermediate elevation angles is adopted as the angle  $\varphi_B$ . We arbitrarily assign the length  $b$  of the connecting rod BE and, making the notches of radius  $b$  from the points  $B_1$  and  $B_2$ , we find the position of the point  $E_1$  at the beginning and at the end of the interval of finite stability. The position of the point  $E_0$  with instantaneous growth is defined as the extension of the crank in the initial position  $AB_0$ . As a result, we obtained the positions of the links AB and BE at the beginning of the working stroke and at the beginning and end of the interval of stability. In the interval of extinction, the point E will approach the point A, reaching a minimum distance from it equal to  $b - l$ . To determine the position  $E_p$  of the point E at the minimum distance from the point A, draw from this point a circle of radius  $b - l$  and a straight line  $AE_p$  under with some small angle  $\rho$ . The intersection of the named circle and the straight line will be taken as the position where the point  $E_p$  will have momentary exponentials at the interval of finite stability.

The foregoing construction corresponds to the definition of the characteristic positions of the so-called two-link drive [4].

We determine the position of the  $C_1$  point of the output link of the CD mechanism at the beginning and end of the interval of the gap  $\varphi_B$ . It is determined from the condition of ensuring a uniform Chebyshev approximation, in accordance with which the point  $C_1$  must occupy the same position both at the beginning and the end of the interval of extinction, and at the moment of instantaneous stability of the point  $E_p$ . Therefore, the point  $E_p$  can be graphically found as the point of intersection of circles drawn from the points  $E_1$  and  $E_p$  by the preset length  $e$  of the EU side of the triangle link ECF.

We find on the triangular link ECF the point F in position  $F_1$  along the previously assigned length  $f$  of the side CF of this link and the angle  $\mu$ .

Let us now turn to the construction of the positions of the links of the mechanism at the instant of instantaneous standing of the working organ. From the point  $E_0$  under the preselected angle  $\gamma$  we draw the side CF of the triangular link ECF and finish this link in the position  $E_0C_0F_0$ . To complete the construction of the kinematic scheme of the mechanism, it is now sufficient to find the positions of the rotation centers D and H of the fixed hinges, respectively, of the output link CD and the auxiliary link FH. The position of the point D and, therefore, the length of the CD are at a given angle of the span  $\psi^m$  with the use of the chord  $C_1C_0$ . To determine the point H and the length FH, it is necessary to draw perpendiculars from the middle of the segments  $E_1E_p$  and  $F_1F_0$ . The intersection of these perpendiculars is the center of the fixed hinge H, and the distance FH is the length of this link.



**Figure 1.** Geometric model of the kinematic scheme of the hinged mechanism of the third class with a stand in one extreme position.

As a result of the geometric constructions carried out at the first stage, a kinematic scheme of the hinged mechanism is realized, realizing the cyclogram with the adopted intermediate prescribed angle of the gap  $\varphi_B$ . The sequence of the presented constructions is the content of the first stage of synthesis. *Second phase.* In the second stage of the synthesis, it is necessary to find the position of the point H for all other (smaller and larger) values of the corners of the stability. The points  $B_1$  and  $B_2$  for other angles of stability will be laid symmetrically to the bisector of the angle  $B_1AB_2$  and from these points we will determine the positions of the points  $E_1$  for each of the corners of the dwell, and then the position of the point  $F_1$ . In accordance with the new positions of the point  $F_1$ , as before, the positions of the center H of the fixed hinge of the auxiliary link FH are determined on the perpendiculars drawn from the middle of the segments  $E_1E_p$  and  $F_1F_0$ .

As an example, in the figure 1, the constructions for angles of stability equal to 100, 110 and 120 angular degrees were made (in the first stage,  $110^\circ$  was assumed as the intermediate value of the standoff angle). The positions of the characteristic points  $B_1$ ,  $B_2$ ,  $E_2$ ,  $F_2$  and H of the mechanism obtained in the second stage of the synthesis for the angles of stability equal to 100 and 120 angular degrees are marked with an additional lower index 2.

To determine the error of the working element's standing, it is sufficient to make a notch on the circle of radius CD with the center at point D equal to the sum of the lengths of the links FH and CF. The angle  $C_1DC_p$  is the deviation of the working element from the standing position. With a suitable selection of the parameters of the mechanism, it is possible to ensure a sufficiently small value of this angle [1].

To switch the mechanism from one cycle to another, it is sufficient to release the fastener on the telescopic link FH and to re-attach the hinge H to the position corresponding to the required stop angle (in the figure 1 of position  $H_{21}$   $H_{22}$ ).

It is important to note that when switching from one cycle to another, the angular working stroke of the output link  $\psi_m$  remains unchanged and is equal to the originally set value. The required position of the operating element in the automatic machine relative to the output link of the mechanism can be set by changing the angle  $\varepsilon$ .

#### 4. Results

A geometric model for the synthesis of lever mechanisms of the third class is obtained, allowing in two stages to design the hinge mechanisms for a number of fixed, preset values of the corners of the stability.

#### 5. The discussion of the results

The proposed geometric model contains all the necessary data for the transition, if necessary, to the analytical method of geometric modeling in order to obtain the corresponding mathematical dependencies.

#### 6. Conclusion

The proposed method of geometric design can be used when developing the appropriate nomograms or tables for setting up the considered mechanisms for reproducing the cyclogram with the required length of the standby due to the change in the length of the auxiliary lead and changing the position of its fixed hinge on the body of the machine with the remaining lengths of the remaining links. The method has the property of visibility, which makes it suitable for use in engineering practice.

#### References

- [1] Wilhelm S R, Sullivan T and Van de Ven J D 2017 Solution rectification of slider-crank mechanisms with transmission angle control *Mechanism and Machine Theory* **107** pp 37–45
- [2] Nadezhdin I V 2010 Designing of level mechanisms of cyclic automatic machines (Moscow: *Mechanical Engineering*) 232

- [3] Chaudhary K and Chaudhary H 2015 Optimal dynamic balancing and shape synthesis of links in planar mechanisms *Mechanism and Machine Theory* **93** pp 127–146
- [4] Gogate G R 2016 Inverse kinematic and dynamic analysis of planar path generating adjustable mechanism *Mechanism and Machine Theory* **102** pp 103–122
- [5] Chanekar P V and Ghosal A 2013 Optimal synthesis of adjustable planar four-bar crank-rocker type mechanisms for approximate multi-path generation *Mechanism and Machine Theory* **69** pp 263–277
- [6] Khomchenko V G, Bakcheev V A, Khorunzhin V S and Skabkin N G 2002 Synthesis of level mechanisms of the third class with an adjustable stand from the condition for minimizing the deviation of the asymmetry angle *Mechanics of processes and machines: collection of scientific papers*, ed V V Evstifeeva (Omsk: OmSTU) pp 8–17
- [7] Qu X and Kim D 2012 Kinematic design and analysis of a four-bar linkage-type continuously variable valve actuation mechanism *Mechanism and Machine Theory* **57** pp 111–125
- [8] Suna J, Chena L and Chu J 2016 Motion generation of spherical four-bar mechanism using harmonic characteristic parameters *Mechanism and Machine Theory* **95** pp 76–92
- [9] Tiana Y, Shirinzadeha B, Zhangb D, Liuc X and Chetwynd D 2009 Design and forward kinematics of the compliant micro-manipulator with lever mechanisms *Precision Engineering* **33(4)** pp 466–475
- [10] Volkert van der Wijk 2017 Design and analysis of closed-chain principal vector linkages for dynamic balance with a new method for mass equivalent modeling *Mechanism and Machine Theory* **107** pp 283–304
- [11] Balchanowski J 2016 General method of structural synthesis of parallel mechanisms *Archives of Civil and Mechanical Engineering* **16(3)** pp 256–268
- [12] Bronsvort W, Gravesen J and Keyser J 2011 Theory and practice of geometric and physical modeling *Computer-Aided Design* **43(7)** pp 739–740
- [13] Zhaoa H, Hana D, Zhangb L and Bi S 2017 Design of a stiffness-adjustable compliant linear-motion mechanism *Precision Engineering* **48** pp 305–314
- [14] Imbach R, Schreck P and Mathis P 2014 Leading a continuation method by geometry for solving geometric constraints *Computer-Aided Design* **46** pp 138–147