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# SYNTHESIS OF THE TRANSFORMING MECHANISM OF THE ROCKING MACHINE

This article discusses the synthesis of a six-link transforming mechanism of a rocking machine. First, the problem of synthesizing a four-link articulated-lever mechanism for reproducing a vertical line was solved. For this purpose, the problem of synthesizing a rectilinear-guiding mechanism of the Evans type, which is a hinged-lever four-link mechanism with a straight vertical line drawing point, is considered. The task of synthesis is to implement the constraint equation. The geometric meaning of the constraint equation is to determine the hinge, the positions of which in the absolute coordinate system are equidistant from the origin of the $OXY$ coordinate system.
The problem of synthesis is formulated as a problem of quadratic approximation. According to the found dimensions of the articulated four-link, performing the position analysis, the true positions of the suspension point of the rod column were determined. After that, the found parameters were refined using the output criterion directly, that is, the deviation from the given rectilinear trajectory.
After the synthesis of a straight-line guiding mechanism, a drive kinematic chain was synthesized, which consists of a crank and a connecting rod.
Thus, a rocking machine drive mechanism was obtained, containing a base, a crank pair connected to the main hinged four-link mechanism. The technical result is achieved by the fact that a two-link group is attached to the main four-link mechanism, forming a class III mechanism. The attached two-drive group is the leading crank connected to the rack and connecting rod.
Based on the obtained dimensions of the six-link converting mechanism, an experimental model was developed, which fully confirmed the efficiency of the transforming mechanism.
**Key words**: Synthesis, rocking machine, drive, connecting rod, four-link articulated-lever mechanism, converting mechanism.

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**Сорғыш қондырғының түрлендіруші механизмінің синтезі**

Бұл мақалада сорғыш қондырғының алты буынды түрлендіруші механизмінің синтезі талқыланады. Біріншіден, тік сызықты жаңғыртуға арналған төртбуынды топсалы-иінтіректі механизмнің синтез мәселесі қарастырылады. Осы мақсатта түзу тік сызықты сызу нүктесі бар топсалы иінтіректі төрт буынды механизм болып табылатын Эванс типті түзу сызықты бағыттаушы механизмді синтездеу мәселесі қарастырылған. Синтездің міндеті - шектеу теңдеуін жүзеге асыру. Шектеу теңдеуінің геометриялық мағынасы абсолютті координаталар жүйесіндегі орындары $OXY$ координаталар жүйесінің басынан бірдей қашықтықта орналасқан топсаны анықтау болып табылады.

Синтез мәселесі квадраттық жуықтау есебі ретінде тұжырымдалған. Топсалы төрт буынның табылған өлшемдеріне сәйкес позициялық талдауды орындау арқылы өзек бағанының ілу нүктесінің шынайы позициялары анықталды. Осыдан кейін табылған параметрлер тікелей шығыс критерийін, яғни берілген түзу сызықты траекториядан ауытқуды пайдаланып нақтыланды.
Түзу сызықты бағыттаушы механизм синтезделгеннен кейін иінді және шатуннан тұратын жетекті кинематикалық тізбек синтезделді.
Осылайша, негізгі топсалы төрт буынды механизмге айналшақ-бұлғақты жұбы негізі қосылған, бар сорғыш қондырғының жетек механизмі алынды. Техникалық нәтижеге екі буынды топ негізгі төртбуынды механизмге бекітіліп, III класты механизмді құрайды. Бекітілген екі жетекті топ тірекке және шатунға қосылған жетекші айналшақ.
Алты буынды түрлендіру механизмінің алынған өлшемдері негізінде түрлендіру механизмінің тиімділігін толық растайтын тәжірибелік үлгі әзірленді.
**Түйін сөздер**: Синтез, тербелгіш машина, жетек, шатун, төртбуынды топсалы-иінтіректі механизм, түрлендіруші механизм.

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**Синтез преобразующего механизма станка качалки**

В данной статье рассматривается синтез шестизвенного преобразующего механизма станка качалки. Сначало решена задача синтеза четырехзвенного шарнирно-рычажного механизма для воспроизведения вертикальной прямой. Для чего рассмотрена задача синтеза прямолинейно-направляющего механизма типа Эванса, который представляет собой шарнирно-рычажный четырехзвенный механизм с чертящей точкой прямую вертикальную линию. Задача синтеза заключается в реализации уравнения связей. Геометрический смысл уравнения связей заключается в определении шарнира, положения которых в абсолютной системе координат является равноудалеными от начала системы координат $OXY$.
Сформулирована задача синтеза в виде задачи квадратического приближения. По найденным размерам шарнирного четырехзвенника, выполняя анализ положений определен истинные положения точки подвеса колонны штанг. После этого произведен уточнение найденных параметров используя непосредственно выходной критерий, то есть отклонение от заданной прямолинейной траектории.
После синтеза прямолинейно - направляющего механизма, синтезирован приводная кинематическая цепь, которая состоит из кривошипа и шатуна.
Тем самым получен механизм привода станка качалки, содержащий основание, кривошипно-шатунную пару соединенный к основному шарнирно четырехзвенному механизму. Технический результат достигается тем, что на основной четырехзвенный механизм присоединяется двухповодковая группа, образуя механизм III класса. Присоединенная двухповодковая группа является ведущим кривошипом, связанное с стойкой и шатуном.
На основе полученных размеров шестизвенного преобразующего механизма разработан экспериментальный образец, который полностью подтвердил работоспособность преобразующего механизма.
**Ключевые слова**: Синтез, станок качалка, привод, шатун, четырехзвенный шарнирно-рычажный механизм, преобразующий механизм.

## 1 Introduction

Of the existing mechanized methods of oil production, the most common is the sucker-rod deep-pumping method with balancing individual drives of mechanical action.

As a converting mechanism connecting the gearbox with the balancer, a four-link crank mechanism is used, which converts the uniform rotation of the crank into the reciprocating movement of the plunger. At the same time, according to the location of the balancing load, the designs of pumping units with crank, rocker (balance) and combined balancing are distinguished. The most commonly used and well-studied are converting mechanisms with a two-arm load balancer and crank balancing, less common are designs with a single-arm load balancer with heavy loads on the balancer and traverse. An important advantage of such installations is the ability to control the pumping mode by changing the stroke of the plunger, for which the crank pin connecting the lower head of the connecting rod with the crank is put on different holes on the crank.

Meanwhile, the use of lever mechanisms, whose connecting rod point describes a straight path with high accuracy, could eliminate the arc head, and the rod string can be hung directly from the drawing point. Such a design could also solve another problem - reducing the metal consumption of the installation due to the possibility of reducing the height of the support frame. The fact is that a significant drawback of the existing design is the high location of the so-called "upper rack", on which the hinge of the balancer is located - the most loaded link. The large height of the balancer attachment point, which is affected by large support reactions, creates a significant swinging force on the rack, which makes it necessary to manufacture a massive foundation from high-quality concrete. This factor is largely due to the high metal consumption of the structure. This drawback can be overcome by synthesizing a lever system with a reduced mounting height of the rack hinges.

## 2 Analysis of literature data and problem statement

Displacement analysis for four-link linkages has been extensively covered in the technical literature [1, 2].

In terms of optimization, bioinspired techniques have expanded significantly over the past two decades. One of the earliest work on an evolutionary algorithm applied to the optimal synthesis of a four-link linkages generator [4]. The authors developed a genetic algorithm to solve three research cases with and without given time and considering different target points. In [5], a procedure for synthesizing the path to the generator connections using a neural network is proposed, it consists of a training stage, at which a large number of kinematic simulations with random dimensions are generated, and at the second stage, the neural network is used to approximate the synthesis of the solution to the problem. The article [6] describes the process of optimal synthesis of a four-link by the method of controlled deviations of variables using the differential evolution algorithm. In [7], the authors consider the Pareto optimal synthesis of four-link mechanisms for generating a path, taking into account the tracking error and the transmission angle error, it is solved using a multicriteria hybrid genetic algorithm. A hybrid evolutionary algorithm for synthesizing a four-link link path is presented in the study [8], where a hybridization between the genetic algorithm and the differential evolution algorithm is proposed. The authors state that the main advantages of this algorithm are the simplicity and ease of implementation and solving of complex optimization problems without the need for deep knowledge of the search space. In the article [9], the authors present a new approach to the multicriteria synthesis of the optimal four-link path and its application to the traditional problem with one, two, and three objective functions. A new algorithm called "Mechanism synthesis algorithm of the University of Malaga" for the synthesis of mechanism paths has been successfully applied to six cases of synthesis of paths and functions of four-bar and six-bar mechanisms [10].

Similar topics can be found in the titles of articles [11, 12, 13]. In the literature, the kinematic and optimization formulation of the four-rod generator is very similar. The kinematic setting in these papers is based on the traditional closed loop condition, and the objective function is the sum of squared Euclidean distances, where the main difficulty is the need for a penalty when the kinematics has no solution in two-dimensional real space.

For this reason, the formulation proposed here is based on the use of natural coordinates and the Hermitian Conjugate of an Operator to construct an objective function whose output is always a positive real number. It should also be noted that the statement proposed here can be extended to any problem of the synthesis of planar mechanisms with a closed solution.

## 3 Solution of the problem

Consider, for the synthesis of a four-link hinged-lever mechanism for reproducing a vertical straight line, the problem of synthesizing a straight-line-guiding mechanism of the Evans type, which is a four-link hinged-lever mechanism $ABCO$ with a drawing point $D$. We consider given $N$ finitely distant positions of the point $D$ along a vertical straight line in a section of length $S$ ($S$ is the stroke of the rod string, for example $S = 2500$ mm), given by the absolute coordinates $X_i^*$, $Y_i^*$ :

$$X_i^* = X_k \tag{1}$$

$$Y_i^* = Y_k + S * \frac{i-1}{N-1}, i = 1, \ldots, \; N \tag{2}$$

We also assume that the parameters of the dyad $ABD$ given by the values $X_A$, $Y_A$, $a$, $b$ are also given. Where $X_A$, $Y_A$ are the absolute coordinates of the hinge $A$ relative to the fixed coordinate system $OXY$ (Figure 1).

Behind each given position $D_i^*$ of the point D along the straight line from the analysis of the dyad ABD with given dimensions, we determine the absolute coordinates $X_{Bi}$, $Y_{Bi}$ of the hinge $B_i$. (Figure 2).

By solving a system of two equations (3) and (4) using the Maple program.

$$\begin{cases} (X_i^* - X)^2 + (Y_i^* - Y)^2 - b^2 = 0 \\ (X_A - X)^2 + (Y_A - Y)^2 - a^2 = 0 \end{cases} \tag{3}$$

$$X = X_{Bi}, \quad Y = Y_{Bi}$$

$$\alpha_i = \arctan(Y_i^* - Y_{B_i}, X_i^* - X_{B_i})$$

$$\begin{cases} X_{Ci} = X_{Bi} + x_C^{loc} \cos\alpha_i - y_C^{loc} \sin\alpha_i \\ Y_{Ci} = Y_{Bi} + x_C^{loc} \sin\alpha_i + y_C^{loc} \sin\alpha_i \end{cases} \tag{4}$$

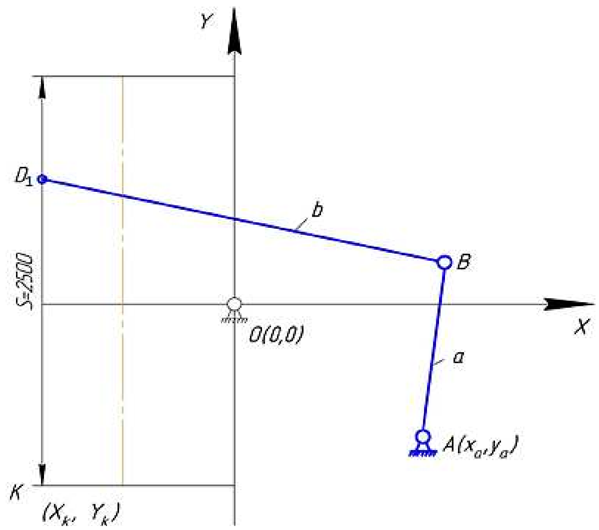


Figure 1: Geometric interpretation of the equation of connection of the synthesis problem.

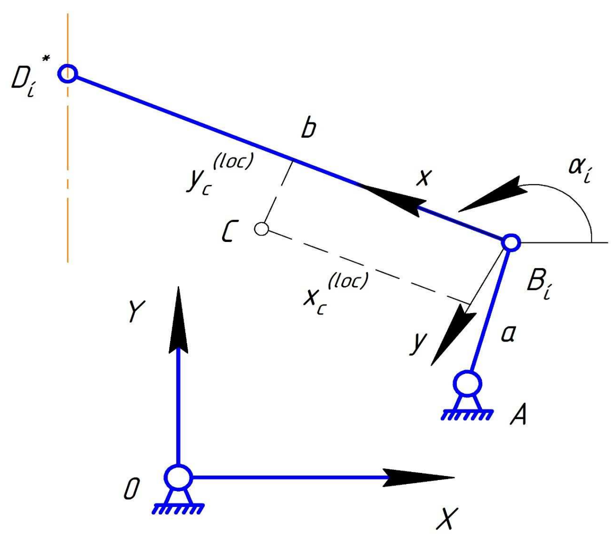


Figure 2: Geometric interpretation of the equation of connection of the problem of synthesis of a four-link mechanism.

According to the given absolute coordinates of the hinges $B_i$ and $D_i^*$, it is possible to determine the angular positions $\alpha_i$ of the links $BD$. The $Bxy$ coordinate system is rigidly connected with the $BD$ link, while the Bx axis is directed along the vector $\overrightarrow{B_iD_i^*}$. Then the absolute coordinates of the hinge $C$ with local coordinates $x_C^{loc}$, ${}_C^{loc}$ are determined from formula (4).

The task of synthesis is to implement the constraint equation of the form (5).

$$(X_{Ci} - X_0)^2 + (Y_{Ci} - Y_O)^2 - l_{OC}^2 = 0, \quad i = 1, \ldots, N. \tag{5}$$

The geometric meaning of this equation is to determine the hinge , the positions of which in the absolute coordinate system $C_i,\ i = 1, \ldots, N$ are equidistant from the origin of the $OXY$ coordinate system. Thus, the positions of the hinge $C$ must approximately realize a circle centered at the point $O$ and with a radius $l_{OC}$.

Substituting from (6) the absolute coordinates $X_{Ci}$, $Y_{Ci}$ into (5), we obtain the equation of relations in the following form

$$[(X_{Bi} - X_O)\cos\alpha_i + (Y_{Bi} - Y_O)\sin\alpha_i]\, x_C^{loc} + [-(X_{Bi} - X_O)\sin\alpha_i + (Y_{Bi} - Y_O)\cos\alpha_i]\, y_C^{loc} + \\ + x_C^{(loc)^2} + y_C^{(loc)^2} - \frac{1}{2}l_{OC}^2 + \frac{1}{2}(X_{Bi} - X_O)^2 + \frac{1}{2}(Y_{Bi} - Y_O)^2 = 0, \quad i = 1, \ldots, N \tag{6}$$

Introduce the notation

$$a_i = (X_{Bi} - X_O)\cos\alpha_i + (Y_{Bi} - Y_O)\sin\alpha_i,$$

$$b_i = -(X_{Bi} - X_O)\sin\alpha_i + (Y_{Bi} - Y_O)\cos\alpha_i$$

$$c_i = 1$$

$$d_i = \frac{1}{2}(X_{Bi} - X_O)^2 + \frac{1}{2}(Y_{Bi} - Y_O)^2$$

Then in new variables $x_1 = x_C^{loc}$, $x_2 = y_C^{loc}$, $x_3 = x_C^{(loc)^2} + y_C^{(loc)^2} - \frac{1}{2}l_{OC}^2$ constraint equations in the form

$$\Delta_i \equiv a_i x_1 + b_i x_2 + c_i x_3 + d_i = 0, \quad i = 1, \ldots, N \tag{7}$$

Here $\Delta_i$ is called a deviation from the implementation of the given equation of relations, then the synthesis problem will consist of approximate implementations of equation (7) for all, $i = 1, \ldots, N$ given positions of points.

In the general case, when $N > 3$, that is, when more than 3 positions of the points $D_i^*$ are given, the exact implementation of equation (7) is not possible, and for their approximate implementations it is necessary to find the minimum of the function

$$S(x_1,\ x_2,\ x_3) = \sum_{i=1}^{N} \Delta_i^2 \to \min_{x_1,\, x_2,\, x_3} \tag{8}$$

Thus, the synthesis problem is formulated as a quadratic approximation problem. Equating the partial derivatives with respect to $x_i$ to zero,

$$\frac{\partial S}{\partial x_i} = 0$$

obtain a system of 3 linear equations for determining the variables $x_1, x_2, x_3$.

$$A\overrightarrow{X} = \overrightarrow{b} \tag{9}$$

where

$$A = \begin{bmatrix} \frac{1}{N}\sum a_i^2 & \frac{1}{N}\sum a_i b_i & \frac{1}{N}\sum a_i c_i \\ \frac{1}{N}\sum a_i b_i & \frac{1}{N}\sum b_i^2 & \frac{1}{N}\sum c_i b_i \\ \frac{1}{N}\sum a_i c_i & \frac{1}{N}\sum b_i c_i & \frac{1}{N}\sum c_i^2 \end{bmatrix}, \quad \overrightarrow{X} = \begin{bmatrix} x_1 \\ x_2 \\ x_3 \end{bmatrix}, \quad \overrightarrow{b} = \begin{bmatrix} -\frac{1}{N}\sum a_i d_i \\ -\frac{1}{N}\sum b_i d_i \\ -\frac{1}{N}\sum c_i d_i \end{bmatrix}$$

The solution of this equation for $\det A \neq 0$ is written as

$$\overrightarrow{X} = A^{-1}\overrightarrow{b} \tag{10}$$

It can be proved that the case $\det A = 0$ corresponds to the case of degeneracy of the system of linear equations (9). The geometric meaning of which is to replace the rotational kinematic pair with a translational one. In view of obtaining an infinite value of the radius of the circle. Based on the found values $x_1, x_2, x_3$, we determine the variables $x_C^{loc}$, ${}_C^{loc}$, and also

$$l_{OC} = \sqrt{\left(x_C^{loc}\right)^2 + \left(y_C^{loc}\right)^2 - 2x_3} \tag{11}$$

Based on the found dimensions of the $ABCO$ articulated four-link, performing the analysis of the positions, we determine the true positions of the point D of the suspension of the column of rods. After that, it is possible to refine the found parameters using the output criterion directly, that is, the deviation from the given rectilinear trajectory.

Let us introduce the local coordinate system $_{55}$ by directing the abscissa axis $_5$ along the link , the angular positions of the $_5$ axis relative to the absolute coordinate system will be denoted by $\alpha_{CD}$ (Figure 3). Absolute coordinates of the new suspension point of the rod column $_{D1}$, $Y_{D1}$ according to the formula (12, 13).

After defining variables

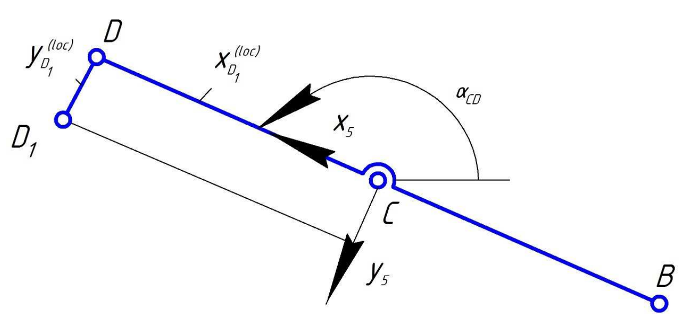


Figure 3: Local coordinate system $_{55}$.

$$X_{D_1} = X_C + x_{D_1}^{(loc)}\cos\alpha_{CD} - y_{D_1}^{(loc)}\sin\alpha_{CD} \tag{12}$$

$$Y_{D_1} = Y_C + x_{D_1}^{(loc)}\sin\alpha_{CD} + y_{D_1}^{(loc)}\cos\alpha_{CD} \tag{13}$$

Synthesis: Refinement

Then the constraint equation is written as

$$X_{D_{i1}} = X_O, \quad i = 1, \ldots, N \tag{14}$$

which means the requirement for the constancy of the $X$ coordinate of the points $D_{1i}$ (Figure 4).

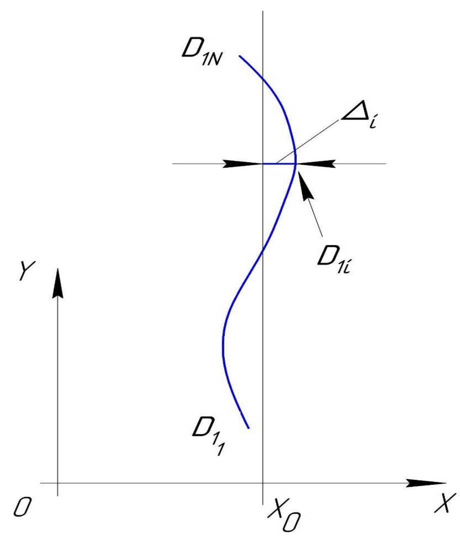


Figure 4: Approximation error – deviation of the true from the given vertical line.

Substituting the value of the absolute coordinates $_{D1}$ from formula (12) we obtain the relation equation in the form

$$-X_0 + x_{D_1}^{(loc)} \cos\alpha_{CDi} - y_{D_1}^{(loc)} sin\alpha_{CDi} = -X_{C_i} \tag{15}$$

Then introducing the notation

$$\begin{array}{l} x_1 = X_0, \quad x_2 = x_{D_1}^{(loc)}, \; x_3 = y_{D_1}^{(loc)} \\ a_i = -1, \; b_i = \cos\alpha_{CDi}, \quad c_i = -\sin\alpha_{CDi}, \quad d_i = X_{C_i} \end{array} \tag{16}$$

We obtain the synthesis equation in the form

$$\Delta_i = a_i x_1 + b_i x_2 + c_i x_3 + d_i = 0 \tag{17}$$

Here $\Delta_i$ is the approximation error. The task of synthesis in the general case for $N > 3$ will be in the approximate implementation of these synthesis equations. To do this, it is necessary to solve the problem of quadratic approximation, which consists in minimizing the function $S$, of the form: problem (8).

The solution of this problem can be obtained by analogy with the solution of the previous problem of quadratic approximation in the form (9). This solution is the only solution to the system of linear equations (11), with $\det A \neq 0$.

## 4 Synthesis of a drive kinematic chain

After the synthesis of a straight-line guide mechanism, it is necessary to synthesize the drive kinematic chain $GFE$, which consists of a crank $GF$ and a connecting rod $FE$ connected to the connecting rod $BC$ (Figure 5).

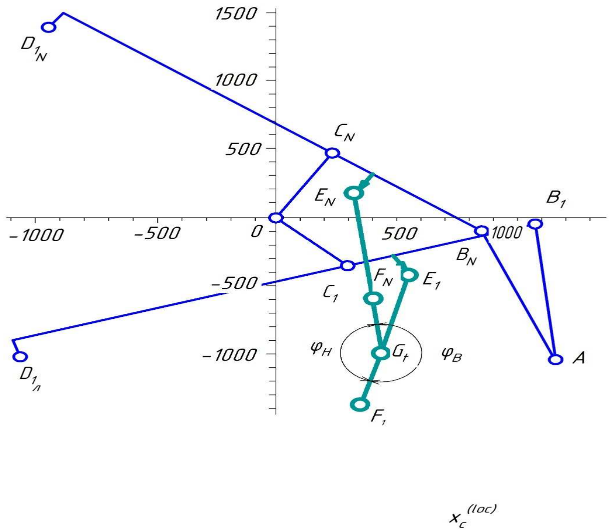


Figure 5: Synthesis of the crank group $GFE$.

Let us introduce the local coordinate system $Bxy$ rigidly connected with the connecting rod $BC$ by directing the $Bx$ axis along the link $BC$. Let us introduce a hinge $E$ on the connecting rod $BC$ with local coordinates $x_E^{loc}$, ${}_E^{loc}$. Then, when the $OC$ link moves from the lowest position $OC_1$ to the extreme upper position $OC_N$, the hinge occupies the positions $E_1, \ldots, E_N$. It is believed that a kinematic analysis of the four-link $ABCO$ has been performed and the angular positions of the connecting rod BC determined by the angle of rotation $\alpha_{BC}$ are known.

Then the absolute coordinates of the hinge $E$ is determined through the absolute coordinates of the hinge $B$ and the rotation angles $\alpha_{BC}$ according to the formulas

$$\begin{aligned} X_{E_i} &= X_{B_i} + x_E^{(loc)} \cos\alpha_{BC_i} - y_E^{(loc)} \sin\alpha_{BC_i} \\ Y_{E_i} &= Y_{B_i} + x_E^{(loc)} \sin\alpha_{BC_i} + y_E^{(loc)} \cos\alpha_{BC_i} \end{aligned} \tag{18}$$

Let's set the position of the hinge $G$ relative to the fixed coordinate system $OXY$ through the coordinates $X_G$, $Y_G$. Denote by $\rho_i$ the distance between the hinges $Gi$ and $Ei$ and determine the minimum and maximum values of $\rho$:

$$\begin{aligned} \rho &= |GE_i| \\ \rho_{\min} &= \min\nolimits_{i=1,\ldots,N} \rho_i \\ \rho_{\max} &= \max\nolimits_{i=1,\ldots,N} \rho_i \end{aligned} \tag{19}$$

Then the required lengths $l_1$, $l_2$ of the crank $GF$ and connecting rod $FE$ are determined from the ratio

$$\begin{cases} l_1 + l_2 = \rho_{max} \\ l_2 - l_1 = \rho_{min} \end{cases} \tag{20}$$

From here we determine $l_1$, $l_2$ by the following formulas

$$\begin{aligned} l_1 &= (\rho_{\max} - \rho_{\min})/2 \\ l_2 &= (\rho_{\min} + \rho_{\max})/2 \end{aligned} \tag{21}$$

Points $F_1$, $G$, $F_N$ define two angles $\varphi_{\text{в}}$ and $\varphi_{\text{н}}$ and $\varphi_{\text{в}} > \varphi_{\text{н}}$, $\varphi_{\text{в}} + \varphi_{\text{н}} = 2\pi$, where $\varphi_{\text{в}}$ corresponds to the angle of rotation of the crank when the rod string goes up, $\varphi_n$ corresponds to the lowering of the plunger down.

The drive mechanism of the rocking machine, containing a base, a crank pair connected to the main articulated four-link mechanism, a balancer support, a two-arm balancer with a front arm and a rear arm, characterized in that it has a connecting rod consisting of two triangular contours, which is pivotally connected to the rear arm a double-arm balancer and with a rocker, and the front triangular contour, which serves as the front shoulder of the connecting rod, is connected to the suspension point of the column rods, and the counterweight is fixed on the front shoulder of the two-arm rocker.

The technical result is achieved by the fact that a two-link group is attached to the main four-link mechanism, forming a III class mechanism. The attached two-drive group is the leading crank connected to the rack and connecting rod.

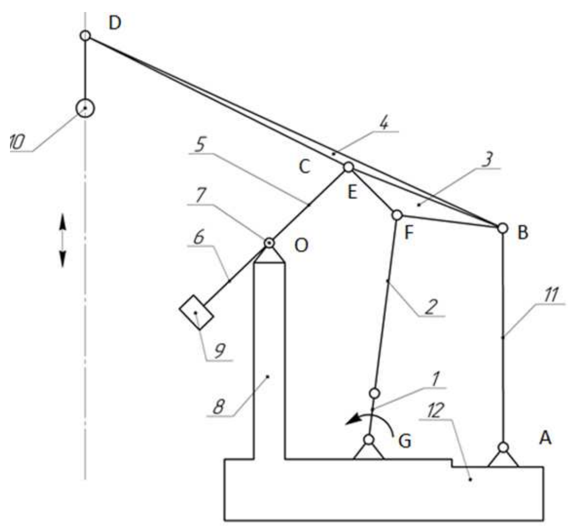


Figure 6: Scheme of the drive mechanism of sucker-rod pumping units in the upper position.

The sucker-rod pumping drive mechanism contains a crank 1 (Figure 6), a connecting rod 2 hinged on one side to the crank 1, and on the other side to the connecting rod, which consists of two triangular contours 3 and 4. The balancer 6 on the rear arm 5 is connected to the connecting rod 3, the middle hinge 7 is connected to the support 8, and the counterweight 9 is fixed on the front arm of the balancer-6. The connecting rod 3 is connected to the rocker arm 11, and the head 10 is fixed on the front arm 4 of the connecting rod 3. The rocker arm 11 and the crank 1 are pivotally connected to the rack 12.

Dimensions: $L_{AB}$ = 1115 mm, $L_{BD}$ = 2360.35 mm, $L_{BD}$ = 1019.205 mm, $L_{BC}$ = 868.28 mm, $L_{CD}$ = 1494.10 mm, $L_{BC}$ = 868.28 mm, $L_{OC}$ = 548.95 mm, $L_{CE}$ = 533.729 mm, $L_{EF}$ = 1163.4655 mm, $L_{FG}$ = 454.879 mm.

## 5 Discussion of experimental results

The analysis showed the possibility of using this mechanism as a converting mechanism for driving sucker-rod pumping units. Based on the obtained dimensions of the six-link converting mechanism, an experimental model was developed, which fully confirmed the efficiency of the transforming mechanism. For the manufacture of an experimental model of the design of a six-link rectilinearly guiding converting mechanism for the drive of sucker-rod pumping units, a geometric model of all structural components of the mechanism was designed in Kompas 3D as part of the work.

An experimental model of the converting mechanism for the drive of sucker-rod pumping units is shown in Figure 7.

Figure 7: Experimental model of a six-link rectilinearly guiding converting drive mechanism.

In addition to a significant gain in dimensions, the use of the mechanism leads to a significant simplification of the design, since the arc head is removed, the connecting rod point is directly connected to the stuffing box without the use of an intermediate flexible link. Reducing the hinges of the mechanism to the foundation leads to a significant reduction in the metal consumption of the foundation, since the rocking forces on the frame of the mechanism are reduced.

## 6 Conclusion

The problem of synthesis of a straight-line guiding mechanism has been solved, a drive kinematic chain has been synthesized, which consists of a crank and a connecting rod. The dimensions of the articulated four-bar linkage are found, by performing the position analysis, the true positions of the suspension point of the rod column are determined. The found parameters were refined using the output criterion directly, that is, the deviation from the given straight-line trajectory.

A rocking machine drive mechanism has been obtained, containing a base, a crank pair connected to the main hinged four-link mechanism. A four-link mechanism is joined by a two-link group, forming a class III mechanism. The attached two-drive group is the leading crank connected to the rack and connecting rod.

Based on the obtained dimensions of the six-link converting mechanism, an experimental model was developed, which fully confirmed the efficiency of the transforming mechanism.

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