Analytical Synthesis and Study of the Functional Capabilities of the Stephenson II Mechanism for Crank Presses

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***Abstract*—This paper is proposed a new analytical method for studying the functionality and synthesis of the Stephenson II crank-slider mechanism according to a given slider change coefficient in average speed and the optimal pressure angle to ensure the highest force transfer from the input link to the functional body (output link). The formulas obtained in the work and their graphs reflect the operational capabilities of the kinematic structure of the Stephenson II crank-slider mechanism. These formulas and graphs authors used when designing this type of crank press. As a result, it is proposed to include them in the technical reference books on mechanisms (or the database on mechanisms). Such reference books about mechanisms that provide a structural diagram and formulas between parameters and their purpose are actively used in engineering and design practice. Disseminating the new mechanism through technical reference books or a mechanisms database contributes to the more significant practical application of the proposed mechanism structure. The application program in the Delphi 7 visual system for determining the parameters of the synthesized crank-slider mechanism by the optimal pressure angle and conducting a kinematic movement analysis of the links in the dialog mode is developed based on the above method.**

***Index Terms*—crank press, Stephenson II crank-slider mechanism, lever mechanisms synthesis, change coefficient in average speed, motion transmission angle**

I. Introduction

Implementing the crank presses technological process must ensure a given cyclogram of the functional slider movement: fast-rising, dwelling, and slow lowering. For this reason, it is necessary to carry out the crank-slider mechanism synthesis according to the change coefficient in the average speed of the output link. It also ensures optimal pressure or transmission angles to provide the most significant transfer of force from the input to the functional body (output link), satisfying simple dynamic requirements.

Investigation and expansion of the functional capabilities of crank-slider operating mechanisms based on four-bar structural groups are attractive for new design development of crank presses and other stamping and forging machines [1], [2]. They eliminate such disadvantages of existing crank presses as skewing of the functional slider in the forging compression and the limited design for implementing the functional slider dwells [1], [3], [4].

One of the press's most important characteristics is its structural elements' rigidity [1], [5]. The rigidity of the press significantly affects the duration of the loading and unloading phases: more significant links rigidity, shorter contact time of stamp with forging. It is crucial for increasing the durability of endurance in hot stamping processes [3], [5]. However, an increase in the rigidity of the press structure leads to a rise in its metal consumption, which is not always economically feasible. For example, improving the forging quality during cold stamping is desirable to increase the duration of the contact of the stamp with the workpiece [4].

In this regard, solving the synthesis of the crank press operating mechanisms based on the Stephenson II, which provides the required functional body dwell, and change coefficient in the average speed of both the crank and the slider has relevance. Moreover, all this is necessary for strictly implementing the cyclogram of the crank press technological process.

The proposed method allows designing a kinematic mechanism diagram to implement a given cyclogram of slider movement. The specified cyclogram implementation accuracy is ensured by selecting constant parameters based on a numerical experiment in the Delphi 7 visual system. So, the cyclogram nature of a crank press is fast rising, dwelling, and slow lowering (workflow).

Should note that all this is true within the framework of this mechanism's functionality (design) structure.

II. Literature Review and Problem Statement

A literature review shows that a crank-slider mechanism based on two-link groups [1] underlies the key functional body of many crank machines of forging and stamping equipment [2]. A press's most straightforward functional operating mechanism is a four-link slider-crank mechanism with one degree of freedom (1 DOF). In work

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of S. Matthews, P. Tanninen, A.Toghyani, H. Eskelinen, S.-S. Ovaska, Ville Leminen, J. Varis the choice of the crank press drive motor has the central part, especially paying attention to the influence of the material (cardboard) thickness on the pressing force [3]. Research on functional mechanisms with one degree of freedom (1 DOF) is currently related to the dynamics of crank presses. The study of the dynamic characteristics of the crank-slider mechanism [4] for a high-speed press system, taking into account the gap in the joints, refers to the regulation of dynamic modes, which assumes the presence of design solutions with optimal geometric dimensions and the transfer function of the functional mechanism. The work compares the efficiency of technological schemes for hot stamping on crank presses [5]. In work of the authors presented a great comparison between types of presses and classifications of the servo press according to mechanisms and drivers [6]. A method of indirect measurements is proposed based on analyzing the size and shape of forgings to assess the manufacture and correct operation of the main elements of a crank-slider press. In single-rod crank-slider mechanisms, the skew of the functional slider is the main reason for the violation of the dynamic modes of the technological process [7]. The analysis of these studies shows the limited functionality of the crank-slider mechanisms based on two-link structural groups with one degree of freedom [7]. To study servo press for metal-forming applications previously studied design and kinematic analysis of mechanisms for servo presses [8] and derived dynamic model using platforms such as MATLAB/Simulink, SimulationX, etc. [9], [10].

The designs of many operating linkages of crank presses are designed by layering various two-link structural groups on the initial RRT group and increasing the number of degrees of freedom of the mechanism. The crank-slider mechanism is used as a feed mechanism for cutting rubber using a circular saw [11]. The synthesis of two independent, autonomous movements can ensure efficient cutting of the rubber block. In work a crank press was investigated from a single-circuit five-link lever mechanism (2 DOF), the drive of which includes an AC electric motor (CV) and a speed regulator, a brushless AC servo motor, and a servo amplifier with a gear drive, a gear shift encoder, a flywheel, and a belt [12]. Double-circuit crank presses with two degrees of freedom (2 DOF) are designed based on seven-level lever mechanisms [13], [14]. This so-called hybrid machine has two inputs. Hybrid machines use two motors, one of which "imitates" a CV motor by prescribing a path profile at a constant speed; the other allows programming of the process sequence by controlling the servomotor. The disadvantage of these designs of the crank presses' functional mechanisms is that the functional sliders skew and often jam the mechanism in particular positions at certain input link positions.

In press-forging machines, mechanisms with higher pairs are used, for example, a wedge-shaped mechanism [15]. This work investigates the kinematics and dynamics of a wedge-shaped press based on its replacement by a conditional two-slider lever mechanism. The requirements for the geometric and kinematic parameters of the press are established based on analysis, and the synthesis problem, as it is accepted in the theory of mechanisms, is not solved in work. Also, the mechanism scheme functionality is not studied.

The authors conducted research on the synthesis of hybrid mechanisms with five rods using genetic algorithms [16]. A study was performed [17] on modeling and kinematic analysis of a hybrid drive of a seven-stage mechanism with an adjustable crank. In work a study was conducted using a hybrid machine (HM) system with a five-girder mechanism [18]. In work a configuration of seven rods was applied using kinematic analysis and the optimal design of a hybrid system [19]. Presented [20] a seven-stage mechanism later used to study stamping efficiency and energy distribution between a servo motor and a flywheel with different motion inputs. In addition, [21] has also developed a control system for a seven-stage mechanism using iterative learning management and feedback control methods. Undoubtedly, these approaches expand the functionality of the executive mechanism, but only at the expense of additional freedom and management functions.

In the works of the works, the Stephenson II crank-slider, which, in comparison with the existing operating mechanism of a crank press, has more comprehensive functional capabilities [7,22]. In this case, the slider skew excludes, which is usually characteristic of single-rod crank-slider mechanisms. The work carried out kinetostatic analysis, manufacturing, and experimental application of a press machine tool based on the Stephenson II mechanism [23]. According to the results of this work Stephenson II mechanism has better characteristics than the conventional press machine. In addition, the Stephenson II mechanism makes it possible to implement the functional link dwell [24] accurately.

The Stephenson II crank-slider mechanism considered in this article is formed based on a more complex assur group of a high class [1]. The presence in the mechanism of two parallel connecting rods between the input and functional bodies eliminates the skew and jam of the slider. The problem of kinematic synthesis of the Stephenson II mechanism, especially by the analytical method, is a complex task. Moreover, there is no work where the kinematic synthesis of the Stephenson II mechanism is considered in terms of the change coefficient in average speed, taking into account the restrictions on the motion transmission angle. The latter determines the coefficient of forces transmission in the mechanism, the assessment of which has a favorable effect on the dynamics of the designed machine (crank press).

K.V. Frolov, S.A.Popov, A.K. Musatov solved the synthesis of the linkage mechanisms in terms of the change coefficient in average speed for "conventional" four-link mechanisms based on two-link Assur groups [25]. The synthesis of four-link mechanisms according to the given link positions or according to the change coefficient in average speed is a classical problem, which was considered by graphical [25], analytical [1], [25], and optimization methods [26], [27]. In works the synthesis method is considered by the change coefficient in the average speed as applied to the four-link hinge and rocker mechanism [1], [25]. The synthesis of the crank-slider mechanism is considered for links in two positions [1], [25] and is not considered separately in terms of the change coefficient in the average speed of the output link. The synthesis task for a given structure and the rational structure choice of linkage mechanisms are posed more broadly [27]. They are reduced to a preliminary assessment of the mechanisms' functionality and various structural charts as generators of functions, trajectories of points, or movements of functional links to create reference material for a reasonable choice of a scheme when solving a specific design problem. In works solutions to many other synthesis tasks of four- and six-link flat lever mechanisms were obtained based on two-link assur groups by the blocked zone method [26],[27]. It ensures the functional links dwell significant contribution in technical reference books for solving engineering problems.

In their works, the authors did not consider the synthesis of the Stephenson II crank-slider mechanism by the coefficient of change in the average speed [24]-[27]. And the speed control problem formulation and the kinematics analytical formulas of the mechanism obtained in the above works have based the development of a new method of kinematic synthesis of the Stephenson II mechanism in terms of the change coefficient in the average speed.

III. Aim and Objects

The study aims to develop a method of kinematic synthesis of the Stephenson II crank-slider mechanism according to the change coefficient in the average speed from the pressure angle extremum conditions to ensure a given cyclogram of crank press functional slider movement: fast-rising, dwelling, slow lowering.

To achieve the aim were set the following tasks:

- output of the calculation scheme and the main kinematic equations of the mechanism;
- task statement of mechanism kinematic synthesis taking into account the cyclogram of functional link movement and the optimization conditions formulation;
- development of a method for solving the kinematic synthesis of the Stephenson II mechanism by the change coefficient in the average speed from the pressure angle extremum conditions.

IV. Materials and Research Methods

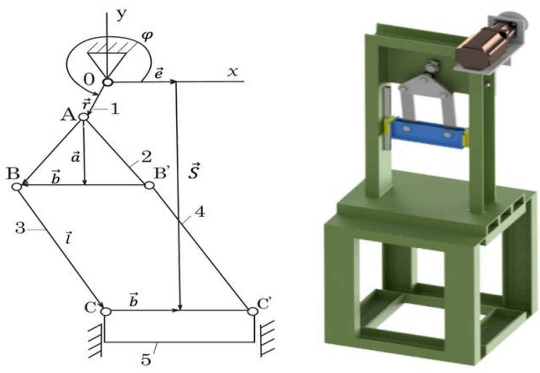
The Stephenson II mechanism has more complex position functions than the simple crank-slider mechanism of the crank press. The research is carried out by the analytical method of lever mechanisms optimization synthesis used in crank presses, in this case, of Stephenson II. The method allows determining the link lengths and the mechanism's eccentricity by a given maximum slider stroke and the change coefficient in the average speed, taking into account the extreme limit on the pressure angle (transmission angle). Based on the necessary extremum condition in the pressure angle, a bicubic equation is obtained, and are proposed this equation analytical solutions to the mechanism's desired parameters.

In this work, change graphs of constant parameters (crank length, connecting rod, eccentricity) are plotted from different values of the coefficient of change in the average crank speed by simulation in the Delphi 7. The crank press Stephenson II mechanism's functional capabilities are evaluated using analytical formulas of the mechanisms' parameters and graphical simulation results. Authors propose to include these formulas and graphs in technical reference books for a reasonable choice of the Stephenson II mechanism when solving a specific design problem.

V. Results

A. Output of the Calculation Scheme and the Main Kinematic Equations of the Mechanism

Based on a literature review and a comparative schemes analysis for the technological process implemented in a crank press, the kinematic scheme of the Stephenson II mechanism is selected. Fig. 1a shows the Stephenson II mechanism kinematic diagram [5]. Fig. 1b shows a 3D model of the crank press experimental sample based on the Stephenson II mechanism. According to the diagram analysis, the mechanism feature is that than variable contour BB′C′C - a parallelogram, and the ABB′ – isosceles triangle. The following designations of coordinates and links sizes were introduced: r – crank 1 length; a – ABB′ triangle height; l – length of parallel connecting rods BC=B$^{/}$C$^{/}$; φ – crank 1 angular coordinate; ψ – angular coordinate of two connecting rods 3 and 4; S – slider 5 linear coordinate; e - slider eccentricity, i.e., trajectory deviation of the slider gravity center from Oy-axis; b - the distance between hinge C and slider center along Ox-axis.



a b

Figure 1. Stephenson II lever mechanism of the crank press: a– kinematic diagram [24]; b– Crank press 3D model [6].

Fig. 2 shows the kinematic mechanism diagram taking into account the above feature, i.e., the presence of an equilateral parallelogram, using a simplified scheme. This diagram was obtained in [24] based on the vector equation $\vec{S}+\vec{e}=\vec{r}+\vec{a}+\vec{l}$.

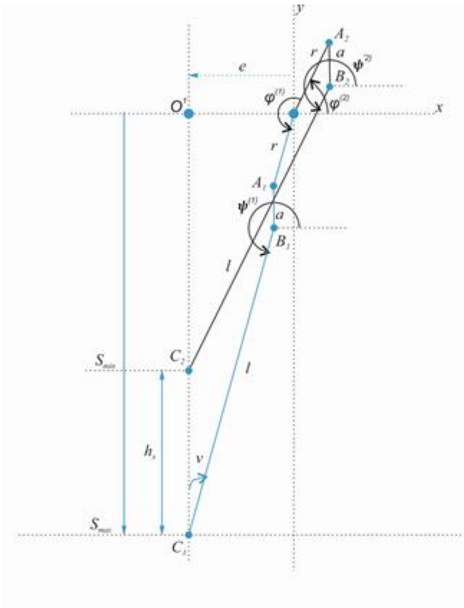


Figure 2. The given diagram of the mechanism is in the two extreme positions of the slide [24].

The kinematics equations of the crank press mechanism [7], [24]:

$$r\cos\varphi + l\cos\psi = e,$$
$$rsin\varphi - a + lsin\psi = -S. \quad (1)$$

The solutions of equations (2) are obtained in the explicit form concerning $S=S(\varphi)$, $\psi=\psi(\varphi)$:

$$\psi = \pm\arccos\left[\frac{1}{l}(e - r\cos\varphi)\right],$$

$$S = a - r\sin\varphi \pm \sqrt{l^2 - (e - r\cos\varphi)^2}, \quad (2)$$

The signs $\pm$ are the two different mechanism assemblies. This corresponds to the fact that when the crank position is fixed, point C occupies a position on the slider guide below the Ox axis (+) or above the Ox axis (-).

B. Cyclogram Analysis of the Crank Press Stephenson Mechanism

The cyclogram of the functional link movement can be represented as follows: let the crank swing angle φ_m and the slider stroke S_m be given; φ_m - the crank positions angular interval corresponding to the slider slow lowering (reverse stroke), (2π- φ_m) - reaching the slider rapid rising (straight stroke). The x, y coordinate system origin is O at the crank rotation center. The slider stroke S_m is given.

The basic ratios between parameters of the motion cyclogram are established. According to geometry based on second system equations (1):

$$S_{\max} = -r\sin\varphi^{(1)} - l\sin\psi + a, \quad S_{\min} = -r\sin\varphi^{(2)} - l\sin\psi^{(2)} + a, (3)$$

Subtracting S_{min} from S_{max} slider stroke is gotten:

$$S_m = S_{max} - S_{min} = r(\sin\varphi^{(2)} - \sin\varphi^{(1)}) + l(\sin\psi^{(2)} - \sin\psi^{(1)}), (4)$$

here $S_m=S_{max}–S_{min}$ - slider stroke, $\varphi^{(1)}$ and $\varphi^{(2)}$ – crank angular positions corresponding to the cyclogram characteristic points, $\psi^{(1)}$ and $\psi^{(2)}$ - connecting rods angular positions, which correspond to the crank angular positions $\varphi^{(1)}$ and $\varphi^{(2)}$.

Based on the second equation of (1), similarly:

$$r(cos\varphi^{(2)} - \cos\varphi^{(1)}) + l(\cos\psi^{(2)} - \cos\psi^{(1)}) = 0. \quad (5)$$

Based on the addition and subtraction formula of trigonometric functions [26], (4) and (5) are written as:

$$rsin\frac{\varphi_m}{2}sin\frac{\varphi^{(2)} + \varphi^{(1)}}{2} + lsin\frac{\theta}{2}sin\frac{\psi^{(2)} + \psi^{(1)}}{2} = 0$$

$$rsin\frac{\varphi_m}{2}\cos\frac{\varphi^{(2)} + \varphi^{(1)}}{2} + lsin\frac{\theta}{2}\cos\frac{\psi^{(2)} + \psi^{(1)}}{2} = \frac{S_m}{2}, \quad (6)$$

where $\varphi_m = \varphi^{(2)}–\varphi^{(1)}$ - crank swing angle; $\theta=\psi^{(2)}–\psi^{(1)}$ – motion transmission angle.

Taking into account the above, (6) is written in the following form

$$a_1sin\alpha_1 + a_2sin\alpha_2 = 0, \quad (7)$$
$$a_1cos\alpha_1 + a_2cos\alpha_2 = x,$$

where

$a_1 = rsin\frac{\varphi_m}{2}$, $a_2 = lsin\frac{\theta}{2}$, $\alpha_1 = \frac{\varphi^{(1)}+\varphi^{(2)}}{2}$, $\alpha_2 = \frac{\psi^{(2)}+\psi^{(1)}}{2}$, $x = \frac{S_m}{2}$.

The solutions of the system (6) satisfy the equations:

$$\cos\alpha_1 = \frac{x^2 + a_1^2 - a_2^2}{2xa_1}, \quad \cos\alpha_2 = \frac{x^2 + a_2^2 - a_1^2}{2xa_2}.$$

Next, ratios between the angles $\varphi^{(1)}$, $\psi^{(1)}$ and $\varphi^{(2)}$, $\psi^{(2)}$ are established. According to the above:

$$\varphi^{(1)} = \alpha_1 - \frac{\varphi_m}{2} \quad \psi^{(1)} = \alpha_2 - \frac{\theta}{2}. \quad (8)$$

Taking into account that $\theta=\varphi_m–\pi$, $\varphi^{(1)}=\psi^{(1)}$ and $\varphi^{(2)}+\pi = \psi^{(2)}$ are gotten based on “(8)”. This means the slider takes extreme positions when the connecting rod and crank are on parallel lines (Fig. 2).

C. Development of Mechanism Synthesis Method from Pressure Angle Extremum Point Conditions

The task of mechanism kinematic synthesis determines mechanism dimensions *a*, *b*, and eccentricity e according to given parameters φ_m and S_m, which ensure the optimal pressure angle v_{ext}.

In synthesis, according to a given angle φ_m, the change coefficient in the average speed of the slider *K* can be equal to

$$K = \frac{\varphi_m}{2\pi - \varphi_m} > 1 \quad (9)$$

Firstly, the formula for eccentricity *e* is defined. Allow $\varphi^{(1)}=\psi^{(1)}$ and $\varphi^{(2)}+\pi = \psi^{(2)}$ from the reduced triangle OC_1C_2

(Fig. 2) is written following ratio, according to the cosine theorem:

$$S_m^2 = (l-r)^2 + (l+r)^2 - 2(l-r)(l+r)\cos\theta. \quad (10)$$

After transforming equation (10) is gotten

$$S_m^2 = 2\left(l^2 - r^2\right)\left(1-\cos\theta\right) + 4r^2. \quad (11)$$

Considering that $\theta=\varphi_m-\pi$: $(1-\cos\theta) = (1+\cos\varphi_m)$.

Based on the formula for degree lowering of a trigonometric function:

$$\left(1+\cos\varphi_m\right) = 2\cos^2\frac{\varphi_m}{2}$$

taking account (11) that

$$\sin^2\frac{\varphi_m}{2} + \cos^2\frac{\varphi_m}{2} = 1$$

$$x^2 = r^2 sin^2 z + l^2 cos^2 z \quad (12)$$

where $x= S_m/2$, $z= \varphi_m /2$.

On the other hand (Fig. 2), for x:

$$2x = \sqrt{(r+l)^2 - e^2} - \sqrt{(r-l)^2 - e^2}, \quad (13)$$

further from (13) e^2 is found:

$$e^2 = r^2 + l^2 - x^2 - \frac{r^2 l^2}{x^2}. \quad (14)$$

Substituting r^2 from (14) into (12), eccentricity is obtained:

$$e = \pm\frac{\cot z}{x}\left(l^2 - x^2\right). \quad (15)$$

In (15), " $\pm$ " means that, due to symmetry, the eccentricity can be chosen, both negative, i.e., the guide can run both to the left of the vertical axis and its right.

"Fig. 2" shows the pressure angle ν. The condition for the optimization synthesis is the maximum pressure angle ν_{ext}. If the right and left sides of (12) are divided by x^2 the dimensionless equation:

$$1 = a^2\sin^2 z + b^2\cos^2 z, \quad (16)$$

where $a=r/x$, $e=e/x$ *and* $b=l/x$ dimensionless parameters with scale factors $x=S_m/2$.

(15) is also represented for dimensionless quantities by dividing by x. Then, concerning dimensionless quantities, (16) and (15) consist of the following system:

$$a^2\sin^2 z = 1 - b^2\cos^2 z,$$

$$e = \pm cotz(b^2-1), b < 1/|cosz|. \quad (17)$$

From the first equation of system "(1)" concerning $\psi=3\pi/2-\nu$:

$$\sin\nu = \frac{a\cos\varphi + e}{b}. \quad (18)$$

From (18) it can be seen that the maximum pressure angle is achieved at $\varphi = 0$ in the initial crank position:

$$\sin\nu_{ext} = \frac{a+|e|}{b},\ b > a + |e|, \quad (19)$$

in this case, the inequality in (19) is the crank existence' condition.

Substituting (17) into (19), the dependence of the maximum pressure angle as a function of the desired parameter b, the dimensionless length of the connecting rod, is obtained:

$$\sin\nu_{ext} = \frac{\sqrt{1-b^2\cos^2 z} + \left|\left(b^2-1\right)\cos z\right|}{b\sin z}. \quad (20)$$

The necessary condition for the function extremum (20) equals zero of the b right-hand side derivative leads to the following bicubic equation:

$$b^6\cos^2 z - b^4\cos 2z + b^2\left(\cos^2 z - 2\right) + \tan^2 z = 0. \quad (21)$$

After the appropriate transformation, (12) can be represented as

$$b^2\left(b^2+1\right)\left[\cos^2 z\left(b^2+1\right)-1\right] - \left(b^2 - tan^2 z\right) = 0,$$

or taking into account that

$$\cos^2 z = 1/\left(1+tan^2 z\right),:$$

$$\left[b^2\left(b^2+1\right)\cos^2 z - 1\right]\left(b^2 - tan^2 z\right) = 0. \quad (22)$$

The solution to the equation $b^2 - \tan^2 z=0$ corresponds to the maximum of the pressure angle, and its value is known (19).

According to "(22)", other extreme values ν_{ext} must satisfy the following biquadratic equation:

$$b^4\cos^2 z + b^2\cos^2 z - 1 = 0. \quad (23)$$

Taking into account that concerning b^2, the equation is quadratic, and its solution satisfying constraints (17) and (19) b equals to

$$b = \sqrt{\frac{1}{2}\left(\sqrt{1+\frac{4}{\cos^2 z}} - 1\right)}. \quad (24)$$

By substituting (24) into the first and second equations of (17) for the mechanism parameters' relative values, the functions from $z=\varphi_m/2$ can be written:

$$a = \frac{1}{2\cos z}\left(\sqrt{\cos^2 z + 4} - |\cos z|\right), e = \frac{1}{2\cos z}\left(\sqrt{\cos^2 z + 4} - 3|\cos z|\right). \quad (25)$$

In addition, based on (19), between the mechanism parameters a, e, and b, there are obvious ratios:

$$a = b^2|\cot z|, \quad e = a - |\cot z|, \quad \frac{1}{b^2} = \frac{a-e}{a}. \quad (26)$$

The mechanism kinematics simulation is carried out based on analytical solutions (2), pressure angle ν based on (18) and S', S'', ψ', ψ'' kinematic parameters formulas, which were calculated by the following formulas:

$$S' = -r\cos\varphi - l\psi'\cos\psi, \quad \psi' = -\frac{r\cos\varphi}{l\cos\psi},$$

$$S'' = r\sin\varphi - l\psi''\cos\psi + l\psi'^2\sin\psi,$$

$$\psi'' = -\frac{l\cos\psi\cdot(\psi')^2 + r\cos\varphi}{l\sin\psi}. \qquad (27)$$

Thus, the authors propose a new analytical method for studying the functionality and synthesis of the Stephenson II crank-slider mechanism for a given change coefficient in the average speed of the slider and the optimal pressure angle to ensure the maximum transfer of force from the functional body (output link) to the input link. Analytical formulas of the mechanism parameters, and graphical analysis of the simulation results, which reflect the functional capabilities of the Stephenson II mechanism, can be included in technical reference books for an informed scheme choice when solving a specific design problem. It is broadly believed that such reference data will facilitate the active introduction of new mechanisms into practice.

VI. Discussion of the Results

Fig. 3 shows the graph of the extreme pressure angle $\nu_{ext}=\nu_{ext}(z)$ graph is plotted, according to (20). And Fig. 4 presents graphs of $b=b(\psi_m)$, $a=a(\psi_m)$, $e=e(\psi_m)$ dimensionless parameters dependences, according to (24), (25).

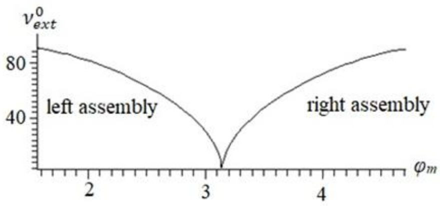


Figure 3. Graph of extreme pressure angle $\nu_{ext}=(\varphi_m)$.

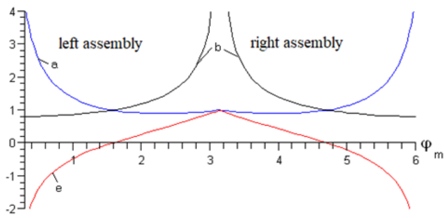


Figure 4. Graphs of functions a, e, and b concerning φ_m.

These graphs analysis shows that there are singular points for φ_m =0; π; 2π in searching for the mechanism parameters when the mechanism dimensions increase indefinitely. All graphs are symmetrical about the vertical φ_m =π. For φ_m =0 and φ_m =3π/2, the links dimensions a=b=1, and the eccentricity e=0, i.e. the central crank-slider mechanism is obtained. On the vertical lines left side (Fig. 3) is obtained the mechanism, which corresponds to eccentricity with "-", and on the right side - eccentricity with "+".

When φ_m = π, the parameter a=1, and connecting rod dimension tends to infinity, the recommendations in [6] are accepted for these particular cases. Based on the previous analysis and graph of pressure angle, the crank swing angle φ_m should belong to the intervals $\varphi_m \in (\pi/2, \pi)$ and $\varphi_m \in (3\pi/2, \pi)$.

These graphs can be used in practical crank presses design based on the Stephenson mechanism as a "map" of the geometric parameters of this mechanism. For that, set the slider speed coefficient (9) and determine φ_m. Next, on graphs (4), restore dimensionless quantities parameters a, b, e. It is advisable to use the right-hand interval when designing because, in this case, the working stroke exceeds the idle speed, which increases this mechanism's efficiency. The results can be incorporated into a design linkage database for various practical applications.

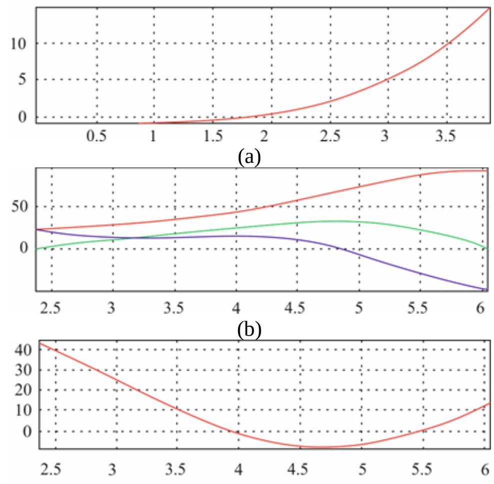


Figure 5. To search for a solution (24) concerning b, kinematic parameters S, S', S'' and pressure angle ν (φ).

Based on the above method, an application program is developed in the Delphi 7 visual system [1, 2], which allows determining the parameters of the synthesized crank-slider mechanism by the optimal pressure angle. Figure 5 shows the simulation process that reflects the search for roots "(21)", kinematic parameters analysis S, S', S'' and pressure angle ν(φ) of the synthesized mechanism. The crank swing angle φ_m or the change coefficient in the average speed of the slider K and the slider maximum stroke S_m is the initial data for design. To simplify calculations and analysis of constraints, all parameters are reduced to dimensionless form according to (24), (25), i.e., it is assumed that the reduced stroke $\bar{S}_m = 2, (x=1)$ is given, and all dimensions are determined depending on this parameter. After pressing the "Synthesis" button, the synthesis parameters a, b, e are derived in a dimensional form depending on the set stroke S_m, the extreme pressure angle ν_{ext}, and the plan of the mechanism position for the angles $\psi \in [\psi_0, \psi_{0+}\psi_m]$.

With a given change coefficient in the average speed K=1.4 and the slider stroke S_m=70cm, the following optimal values of the crank press Stephenson mechanism parameters are determined: φ_m=210^0, r=31,85cm, l=64,5cm, e=22,47cm, a=25cm, b=30cm, v_{ext}=57,36^0 [19].

The research contribution is developing a kinematic synthesis method for the Stephenson II mechanism of a crank press and the scientific substantiation of the choice of press operating geometric parameters of the press operating mechanism from the change coefficient in the average speed and pressure angle. These parameters affect the technological mode of operation and the transmission of forces in crank presses.

VII. Conclusions

1) The paper solves the problem of designing (kinematic synthesis) the crank-slider mechanism Stephenson II. For this, the kinematic scheme of the mechanism for performing the technological process in crank presses is justified, in which the functional body skew is excluded. Furthermore, the mechanism simplified scheme for the derivation and analysis of kinematic equations and calculated dependencies between the variable and constant parameters of the mechanism has been compiled.

2) The analysis of the movement cyclogram of the crank press is carried out, which provides the specified technological process, in particular, the smooth implementation of the working compression and the rapid implementation of the reverse process. This process uses the change coefficient in the average crank rotation speed.

3) An analytical method has been developed for kinematic synthesis of the Stephenson II crank-slider mechanism according to a given change coefficient in the average speed from the conditions of optimizing the pressure angle to ensure the most significant transfer of force from the input link to the functional body. The bicubic equation is obtained from the pressure angle extremum conditions of necessity, which is solved analytically.

As a result, the formulas for the extreme pressure angle on the geometrical parameters of the mechanism are obtained. In addition, are plotted changes in graphs in constant parameters (crank, connecting rod, eccentricity) from different values of the crank swing angle, which corresponds to the crank press working process. Finally, a physical interpretation of the obtained solutions and analytical formulas is carried out. Analytical formulas of the mechanism parameters, and graphical analysis of the simulation results, which reflect the functional capabilities of the Stephenson II mechanism, can be included in technical reference books for an informed scheme choice when solving a specific design problem.

Conflict of Interest

The authors declare no conflict of interest.

Author Contributions

Amandyk K. Tuleshov is the project supervisor and contributed to writing all parts of this work; Balzhan I. Akhmetova conducted the primary research and contributed to the system simulation; Bakhyt M. Merkibayeva contributed to the obtaining experimental results for the paper.

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References

[1] M. Z. Yevgrafov, Y. A. Semenov, and A. V. Sloushch, *Theory of Mechanisms and Machines: Textbook*, Publishing Center "Academy", 2006.- 560 p.

[2] B. Yu, *A. Forging and Stamping Equipment*, M.: Publishing center "*Academy*", 2008, p. 480.

[3] S. Matthews, P. Tanninen, A.Toghyani, H. Eskelinen, S. S. Ovaska, V. Leminen, J. Varis, "Novel method for Selection of drive motor in paperboard forming press utilizing multi-dynamics model based on material thickness," in *Proc. 27th International Conference on Flexible Automation and Intelligent Manufacturing*, FAIM2017, 27-30 June 2017, Sami Matthews et al. /Procedia Manufacturing 11 2017, pp. 2091–2098, Modena, Italy.

[4] Y. Chen, Y. Sun, and D. Yang, "Investigations on the dynamic characteristics of a planar slider-crank mechanism for a high-speed press system that considers joint clearance," *Journal of Mechanical Science and Technology*, vol. 31, no. 1, pp. 75~85, 2017.

[5] I. Telegin and T. Shumilova, "Comparing the efficiency of flowsheets for hot die forging on crank presses," in *Proc. ICMTMTE 2018 MATEC Web of Conferences*, vol. 224, 01092 2018.

[6] R. Halicioglu, L. C. Dulger, and A. T. Bozdana, "Mechanisms, classifications, and applications of servo presses: A review with comparisons," in *Proc. the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture, vol.* 230, no. 7, pp. 1177–1194, 2016.

[7] A. K. Tuleshov, Y. M. Drakunov et al. *Crank Lever Presses Based on the Stephenson II Mechanism: Monograph*, Almaty, 2020.

[8] R. Halicioglu, L. C. Dulger, and A. T. Bozdana, "Modelling and simulation based on Matlab/Simulink: A Press Mechanism," in *Proc. 2014 J. Phys.: Conf.* Ser. 490 012053

[9] R. Halicioglu, L. C. Dulger, and T. Bozdana, "Modeling, design, and implementation of a servo press for metal-forming application," *Int J Adv Manuf Technol*, vol. 91, pp. 2689–2700, 2017.

[10] A. A. Jomartov and A. Tuleshov, et al. "Simulation of the crank press dynamics by SimulationX software," *Journal of Mathematics, Mechanics and Computer Science*, [S.l.], vol. 102, no. 2, pp. 22-33, July 2019.

[11] S. H. Liu and M. Lin, "Design and test of the crank slider mechanism feeding type rubber rotary cutting machine," *Springer Nature Switzerland AG*, vol. 2, p. 917, 2020.

[12] P. R. Ouyang, Q. Li, W. J. Zhang, "Design, modelling and control of a hybrid machine system," *Mechatronics*, vol. 14, pp. 1197–1217, 2004.

[13] Z. Yuan, M.J. Gilmartin, S.S. Douglas, Design of hybrid machines for nonuniform motion production, in *Proc. Inst. Mech. Eng.* C 219 (2005) 491–499.

[14] M. E. Kutuk, L. C. Dulger, "A hybrid press system: Motion design and inverse kinematics issues," *Engineering Science and Technology an International Journal*, pp. 846-856, 2016.

[15] S. G. Karnaukh, N. V. Chosta, O. E. Markov, V. V. Kukhar, "Development and research of the press operating mechanism, made in the form of the wedge-joint mechanism with a curving wedge for separation operations (2021)," in *Proc. International Journal of Advanced Manufacturing Technology*, vol. 116, no. 9-10, pp. 3305-3314.

[16] A. M. Connor, S. S. Douglas, and M. J. Gilmartin, "The synthesis of hybrid five-bar path generating mechanisms using genetic algorithms," in *Proc. 1st International Conference on Genetic Algorithms in Engineering Systems: Innovations and Applications*, pp. 313–318, Galesia, 1995.

[17] L. C. Dülger, A. Kireçci, and M. Topalbekiroglu, "Modeling and simulation of a hybrid actuator," *Mech. Mach. Theory*, vol. 38, pp. 395–407, 2003.

[18] H. Yu, "Modeling and control of hybrid machine systems – a five-bar mechanism case," *International Journal of Automation and Computing*, vol. 3, pp. 235–243, 2006.
[19] H. Li and Y. Zhang, "Seven bar mechanical press with hybrid driven mechanism for deep drawing; part 1: kinematics analysis and optimum design, part 2: dynamic modeling and simulation," *J. Mech. Sci. Technol.*, vol. 11, 2153–2160, 2161–2167, 2010.
[20] C. H. Li and P. L. Tso, "Experimental study on a hybrid driven press using iterative learning control," *International Journal of Machine Tools and Manufacture*, vol. 48, pp. 209–219, 2008.
[21] P. L. Tso, "Optimal design of a hybrid-driven servo press and experimental verification," *J. Mech. Design*, 132, 034503.
[22] Patent of the Republic of Kazakhstan for invention No. 33874. Crank-slide mechanism of the press /Tuleshov A.K., Jamalov N.K., Jomartov A.A., etc.; *applicant and patent holder of U.Joldasbekov IME*. - No.2018/0225.1 of 10.04.2018; publ. 06.09.2019, bul. No. 36. – 4 p. [in Russian])
[23] A. Jomartov, R. Halicioglu, and M. Kuatova, "Kinetostatic analysis, manufacturing, and experimental application of a press machine based on Stephenson II mechanism," in *Proc. the Institution of Mechanical Engineers, Part B: Journal of Engineering Manufacture*, vol. 236, no. 8, pp. 1113–1124, 2022.
[24] A. Tuleshov, R. Halicioglu, A. Shadymanova, and M. Kuatova, "Kinematic synthesis method and eccentricity effects of a Stephenson mechanism," *Mech. Sci.*, vol. 12, pp. 1-8, 2021.
[25] Theory of mechanisms and mechanics of machines: Studies.for vtuzov/K.V.Frolov, S.A.Popov, A.K. Musatov, etc.; *Edited by K.V.Frolov, - M.:Higher School*, 2003.- 496 p.
[26] E. E. Peisakh and V. A. Nesterov, "Design system of flat lever mechanisms," *Moscow: Mashinostroenie*, p. 232, 1988.
[27] E. E. Peisakh, "Optimization-quadratic synthesis of plane lever mechanisms," *Machine Science*, no. 5, pp. 71-78, 1986.

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