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COMPARATIVE ANALYSIS OF THE KINEMATIC PARAMETERS OF THE WEDGING DRIVE MECHANISMS OF THE DIE-CUTTING PRESSES USING THE SOLIDWORKS SOFTWARE

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Die-cutting presses are essential equipment components for producing cardboard packaging blanks. The efficiency and quality of their production depend on the design and operation of the drive mechanism of the pressure plate in the die-cutting presses. This study presents a kinematic analysis of wedging mechanisms in the drive of the pressure plate using the SolidWorks software. To design a 3D model of the mechanism, calculations of the geometric parameters of the mechanisms are performed based on the dimensional parameters of the punching press. The graphical representation of the obtained data during the modeling of the studied mechanisms in SolidWorks Motion is provided. Based on the research on changes in the values of kinematic characteristics, a comparative analysis of the operation of the drive mechanisms of the pressure plate is conducted.

Keywords: *die-cutting press, cardboard blank, pressure plate drive mechanism, double-crank wedging mechanism, CAD/CAE modeling, kinematic analysis.*

Problem Statement. Die-cutting presses with the flat-parallel movement of the pressure plate are widely used in the production of cardboard packaging [1]. In modern equipment, such as BOBST ExpertCut die-cutting presses, a wedging mechanism is utilized for driving the pressure plate during the operation [2]. This mechanism enables vertical movement of the pressure plate from the bottom upwards towards a rigidly fixed stationary plate with the die-cutting form. Within the working zone between the pressure plate and the die-cutting form, cardboard blanks are fed, and the pressure plate transports them toward the form tools. In such equipment, the process of die-cutting blanks involves simultaneous execution of multiple operations, including cutting out the contour of the blanks, creasing of fold lines, embossing of specific areas, and perforation if necessary. To ensure high-quality creasing and embossing, a longer contact time between the cardboard blanks and the tools and die-cutting form is required, positively affecting the formation of crease lines and embossed patterns in the blanks.

Analysis of Recent Research and Publications. In the scientific literature, a limited number of research works are dedicated to analyzing pressure plate drive mechanisms widely used in die-cutting presses. However, it has been found that studies [3, 4] have researched existing mechanisms and performed their comparative analysis. These studies have noted that wedging mechanisms possess several positive qualities associated with overcoming significant technological resistance during cardboard die-cutting (exceeding 600 tons). Mechanisms of this type enable the overcoming of substantial resistance by employing the "wedging" effect during the die-cutting operation. However, the existing mechanism has several operational drawbacks that affect production efficiency. One significant drawback is the presence of oscillatory motion of the pressure plate [2].

To address this drawback, the study [5] proposes reducing the swing angle of the two-armed crank on the drive shaft, while studies [6, 7, 8] suggest replacing the eccentric drive with a cam drive. The study [9] introduces an additional rocker mechanism to the drive system. It minimizes impact loads on the drive by redistributing the duration of the idle and working strokes of the pressure plate. However, the rocker mechanism complicates the press construction.

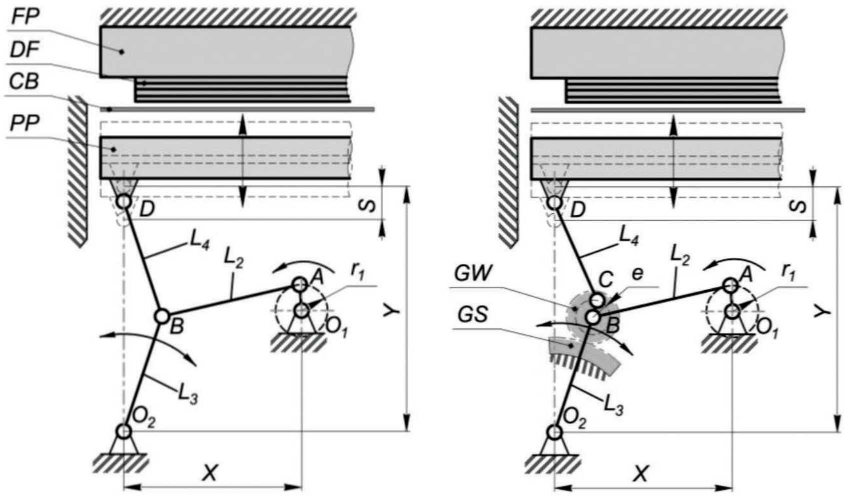
Currently, ongoing research efforts are dedicated to improving the wedging mechanism. However, more studies need to focus on enhancing the wedging mechanism to ensure a longer duration of the pressure plate's presence in the die-cutting zone for forming die-cut patterns.

The article aims to conduct a kinematic analysis of the existing wedging mechanism and the proposed double-slider mechanism with an additional guided slider for the pressure plate drive in a die-cutting press using SolidWorks. The obtained results will be compared, and based on the analysis, the prospects for future research on improving the pressure plate drive mechanism will be justified.

Presentation of the main research material. The existing and proposed combined wedging mechanisms for the pressure plate drive consist of a fixed stationary plate *FP* (Fig. 1a) with a flat die-cutting form *DF* mounted on it, a pressure plate *PP* that moves vertically from bottom to top, a crankshaft O_1A, a connecting rod *AB*, a rocker O_2B, and a driven connecting rod *BD* (*CD* in the proposed double-slider mechanism). The proposed mechanism differs from the existing one by adding the additional driven crank *BC* (Fig. 1b), in which the gear transmission moves. The gear wheel *GW* is fixed on the same shaft as the driven crank *BC* at point *B* and moves along the gear sector *GS*, thereby causing the motion of the crank *BC*, which in turn drives the driven connecting rod *CD*.

To analyze the kinematic characteristics of the investigated mechanisms in the SolidWorks system, it is necessary first to build 3D models of the mechanisms. To construct 3D models of the mechanisms in their accurate dimensions, a synthesis of the investigated mechanisms has been performed to determine their geometric parameters. The input parameters for synthesis are taken as the dimensional sizes of the mechanism for the press pair format SP76 with dimensions of 760×560 mm [1].

– Y = 340 mm – total height of the mechanism;
– X = 250 mm – distances between supports horizontally;
– S = 80 mm – displacement of the pressure plate;
– γ_0 = 5° – the angle of limiting the movement of the links in the uppermost position of the mechanism to avoid jamming the mechanism;
– e = 30 mm – the size of the driven crank (for the double-crank mechanism);
– ω = 60 rpm – rotation frequency of the main shaft.



a b

Fig. 1. Calculation schemes of the left circuit for the synthesis of geometric parameters: a) existing wedging mechanism; b) double-crank wedging mechanism with the additional driven crank

The geometric parameters of the mechanism are calculated based on the derived analytical dependencies.

– the height of the mechanism in the lowest position:

$$L_0 = Y - S; \tag{1}$$

– interbase distance $O_1 O_2$:

$$L_1 = \sqrt{\left(\frac{Y}{2}\right)^2 + X^2}; \tag{2}$$

– rocker length:

$$L_3 = \frac{H \cdot \sin(\gamma_0)}{\sin(\pi - 2 \cdot \gamma_0)}; \tag{3}$$

– driven connecting rod length L_4 in the existing mechanism:

$$L_4 = L_3; \tag{4}$$

– driven connecting rod length L_4 in the double-crank mechanism:

$$L_4 = L_3 - e; \tag{5}$$

– the angle of inclination of the rocker arm O_2B to the vertical axis in the lowest position in the existing mechanism:

$$\gamma_{02} = \arccos\left(\frac{L_0^{\ 2}}{2 \cdot L_3 \cdot L_0}\right); \tag{6}$$

– the angle of inclination of the rocker arm O_2B to the vertical axis in the lowest position in the double-crank mechanism:

$$\gamma_{02} = \arccos\left(\frac{L_0^{\ 2} + L_3^{\ 2} - (L_4 - e)^2}{2 \cdot L_3 \cdot L_0}\right); \tag{7}$$

– angle $O_1O_2B_2$ (kinematic couple B in the uppermost position):

$$\gamma_{\max} = \arcsin\left(\frac{X}{L_1}\right) - \gamma_0; \tag{8}$$

– the distance from support O_1 to point B_2 (leftmost position of kinematic pair B) [10]:

$$L_{O1B2} = \sqrt{L_3^2 + L_1^2 - 2 \cdot L_3 \cdot L_1 \cdot \cos(\gamma_{\max})}; \tag{9}$$

– angle $O_1O_2B_1$ (kinematic couple B in the lowest position):

$$\gamma_{\min} = \arcsin\left(\frac{X}{L_1}\right) - \gamma_{02}; \tag{10}$$

– the distance from support O_1 to point B_1 (rightmost position of kinematic pair B):

$$L_{O1B1} = \sqrt{L_3^2 + L_1^2 - 2 \cdot L_3 \cdot L_1 \cdot \cos(\gamma_{\min})}; \tag{11}$$

– connecting rod length L_2:

$$L_2 = \frac{L_{O1B2} + L_{O1B1}}{2}; \tag{12}$$

– driving crank length r_1:

$$r_1 = \frac{L_{O1B2} - L_{O1B1}}{2}. \tag{13}$$

It has determined the main geometric parameters of the mechanisms based on the analytical dependencies provided above. These parameters were used to construct the 3D models of existing wedging and double-crank mechanisms. The results of calculating the geometric parameters for designing the existing and double-crank mechanisms are presented in Table 1.

Table 1

Geometric Parameters for Designing 3D Models of the Research Mechanisms

Params	Length, mm	
	Exciting mechanism	Double-crank mechanism
r_1 – driving crank O_1A	45,3	29,8
L_2 – connecting rod AB	190,1	205,4
L_3 – rocker O_2B	170,6	170,6
L_4 – driven connecting rod BD	170,6	140,6

The created 3D models of the existing wedging mechanism and the double-crank wedging mechanism are depicted in Figures 2 and 3, respectively. The main elements that form the structure of the existing mechanism are as follows: FP - fixed plate with the die-cutting form; PP - movable press plate; 1 - driving crank; 2 - connecting rod; 3 - lower rocker arm; 4 - driven connecting rod. The structure of the double-crank mechanism slightly differs from the existing mechanism and includes the following additional elements: 4 - gear wheel; 5 - gear sector; 6 - driven crank. That elements drive the driven connecting rod 7.

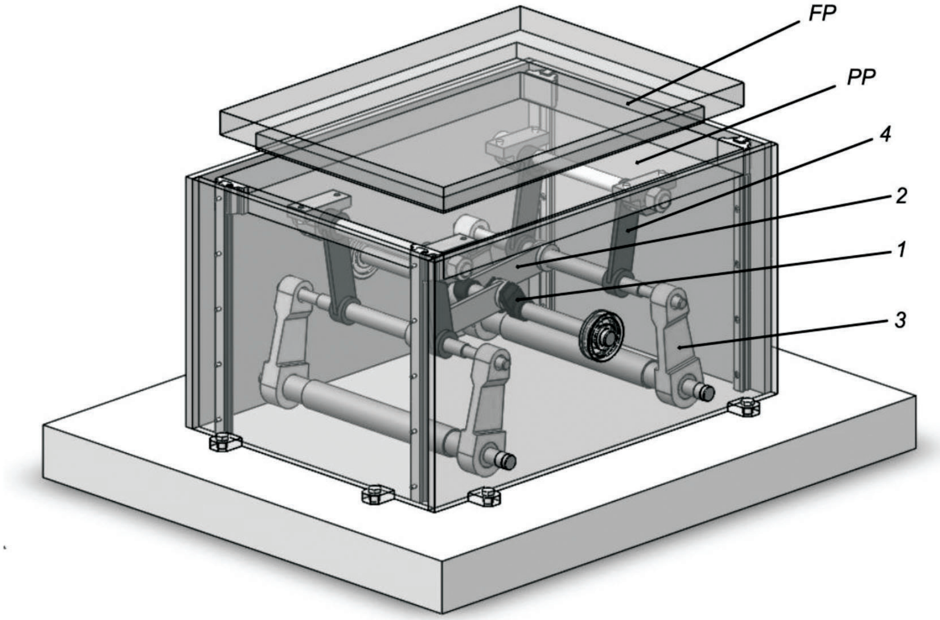


Fig. 2. 3D model of the existing wedging mechanism of the pressure plate drive

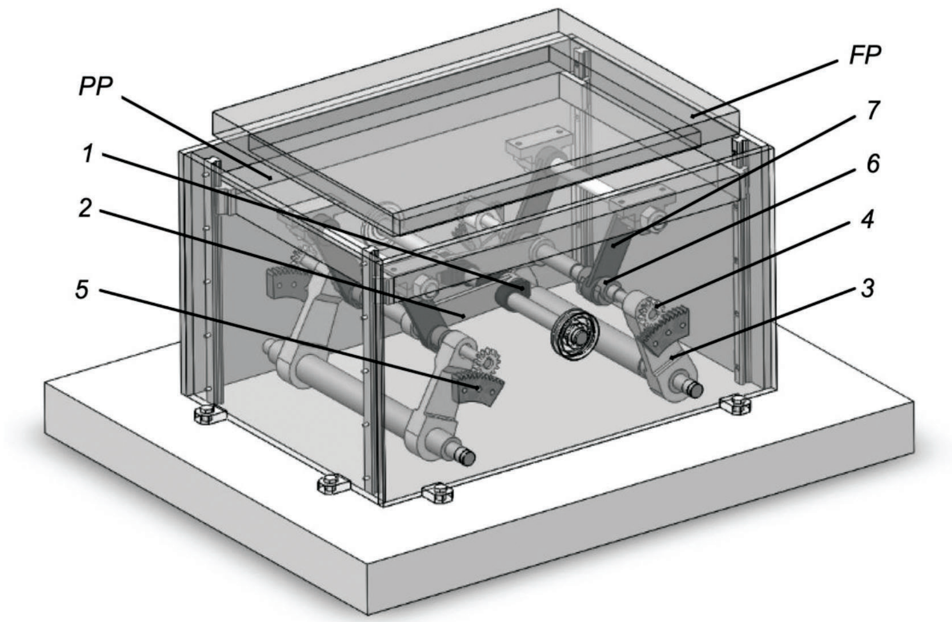


Fig. 3. 3D model of the wedging mechanism with an additional driven crank of the pressure plate drive

Based on the results of the mechanism simulation in SolidWorks Motion, the values of displacement, velocity, and acceleration of the pressure plate were obtained for both the existing wedging mechanism and the proposed double-crank wedging mechanism. As seen in the graphs (Figures 4, 5), the displacement of the pressure plate in the double-crank wedging mechanism remains longer at the end of the working cycle, indicating that the press plate approaches the die-cutting operation zone more rapidly and then slows down. The displacement curve of the existing mechanism is slightly shifted, indicating the asymmetric motion of the right and left sides of the mechanism.

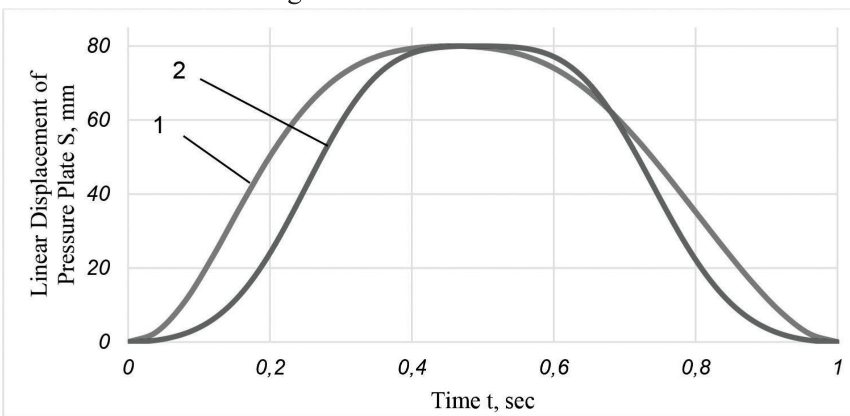


Fig. 4. Graphs of the dependence of the displacement of the pressure plate on time in the wedging mechanisms: the existing (1), the proposed double-crank mechanism (2)

Graphs of the movement of the pressure plate (Fig. 5) confirm that the tools of the die-cutting form in the double-crank wedging mechanism interact with cardboard Δ = 0.5 mm thick for 12% longer than in the existing one.

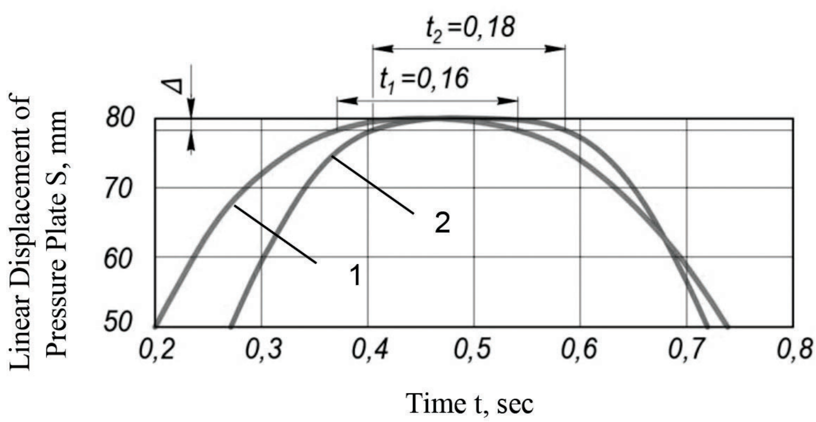


Fig. 5. Graphs of the dependence of the movement of the pressure plate on time in the cardboard die-cutting zone in the wedging mechanisms: the existing (1), the proposed double-crank (2)

In the existing wedging mechanism, the peak velocities of the pressure plate during the working stroke are $V = 0.35$ m/s (Fig. 6) and $V = -0.24$ m/s during the idle stroke. The graphs show that the velocity curve smoothly changes during the transition from working to idle stroke. In the proposed double-crank mechanism, it can be observed that the velocity is closest to zero in the middle of the cycle. The peak velocities during the working and idle strokes of the proposed mechanism are the same in magnitude ($V = 0.37$ m/s and $V = -0.37$ m/s) but slightly higher than in the existing wedging mechanism.

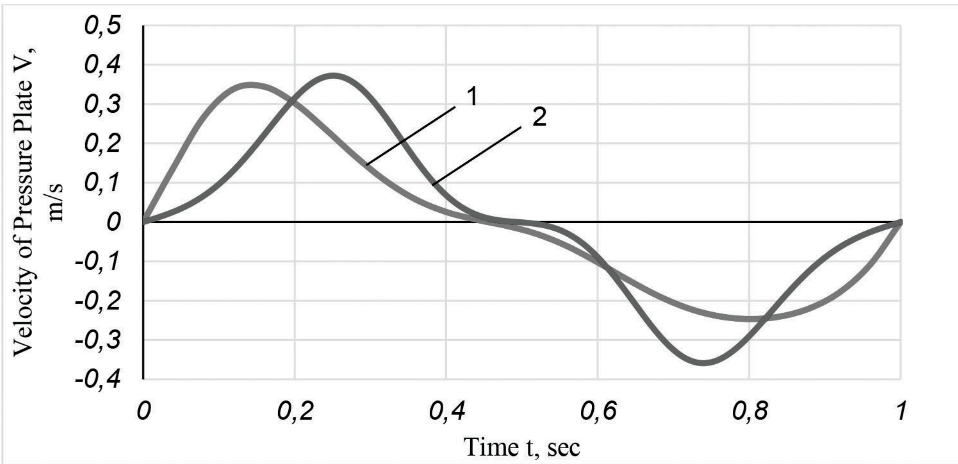


Fig. 6. Graphs of the dependencies of the velocities of the pressure plate on time in the wedging mechanisms: existing (1), proposed double-crank mechanism (2)

The results of the acceleration analysis show that the pressure plate starts with an acceleration of $W = 3.17$ m/s² (Fig. 7), and the peak acceleration during the working stroke is $W = 3.5$ m/s² (negative value $W = -1.75$ m/s²). During the idle stroke, the negative peak acceleration value is $W = -1.1$ m/s², and the peak of positive acceleration occurs at the end of the cycle and is equal to $W = 3.17$ m/s². In the middle of the mechanism's operating cycle (in the zone of performing the technological operation), the acceleration of the pressure plate is $W = -0.55$ m/s². The press plate starts the cycle with $W = 0.58$ m/s² acceleration in the proposed double-crank mechanism. During the working stroke, the peak accelerations have values of $W = 2.46$ m/s² and $W = -2.74$ m/s², while during the idle stroke, they are $W = 2.13$ m/s² and $W = -2.63$ m/s². At the end of the working stroke, the acceleration of the pressure plate approaches zero and becomes $W = -0.16$ m/s².

The acceleration curve of the pressure plate in the double-crank mechanism indicates that the pressure plate starts with an acceleration that is 5.46 times smaller than in the existing mechanism. The positive peak acceleration of the pressure plate in the double-crank mechanism is 1.64 times smaller, while the negative acceleration is 1.5 times larger. During the idle stroke, the positive peak acceleration in the proposed mechanism is 1.48 times smaller, while the negative acceleration is 2.39 times larger. The working phase of the cycle in the double-crank mechanism ends with an acceleration that is 3.43 times smaller than in the existing mechanism.

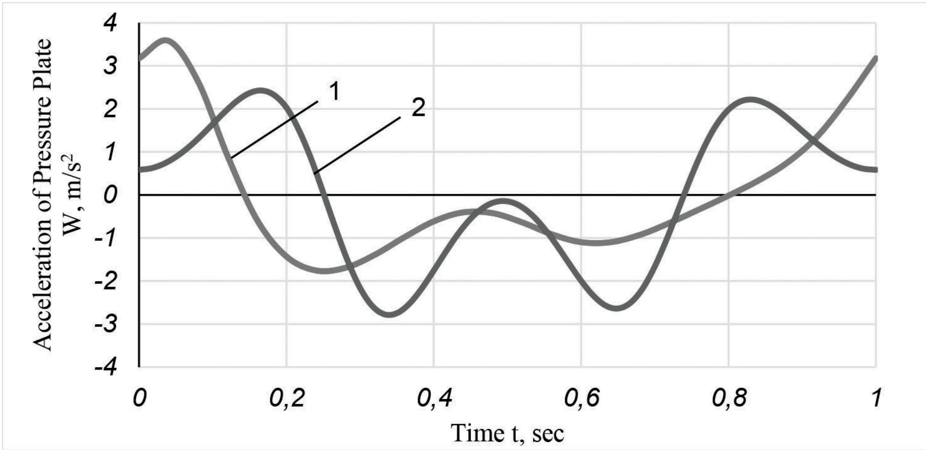


Fig. 7. Graphs of the dependence of the acceleration of the pressure plate on time in the wedging mechanisms: existing (1), proposed double-crank (2)

Conclusions. In this work, a review of works related to the improvement of wedging mechanisms in die-cutting presses has been conducted. It is found that limited attention is given to improving wedging mechanisms that involve cycle redistribution, enhancing the efficiency and productivity of the process.

The research has derived analytical dependencies for calculating the existing wedging mechanism's geometric parameters for the pressure plate's drive and the proposed double-toggle mechanism. Based on the derived analytical dependencies, a geometric synthesis is conducted. The synthesis results give the necessary parameters for designing 3D models of the researched mechanisms.

The research on kinematic characteristics is conducted in the SolidWorks system based on the designed 3D models. The analysis results reveal significant differences in the displacement values of the pressure plate between the existing wedging mechanism and the proposed double-crank mechanism. In the proposed mechanism, the pressure plate moves faster towards the die-cutting zone but slows down at the end of the working stroke. Additionally, in the proposed mechanism, the duration of the pressure plate's presence in the die-cutting zone and the interaction of the cardboard with its forming tools is 12% longer compared to the existing mechanism. This positively affects the creasing process and the relief embossing of cardboard blanks. The acceleration of the pressure plate in the proposed mechanism is 3.43 times lower at the end of the working stroke compared to the existing mechanism.

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ПОРІВНЯЛЬНИЙ АНАЛІЗ КІНЕМАТИЧНИХ ПАРАМЕТРІВ РОЗКЛИНЮВАЛЬНИХ МЕХАНІЗМІВ ПРИВОДА НАТИСКНОЇ ПЛИТИ У ШТАНЦЮВАЛЬНИХ ПРЕСАХ ЗА ДОПОМОГОЮ ПРОГРАМИ SOLIDWORKS

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Штанцювальні преси є основними складовими обладнання для виготовлення розгорток картонного паковання. Ефективність та якість їх виробництва залежать від конструкції та функціонування механізму привода натискної плити штанцювальних пресів. У сучасному обладнанні для привода натискної плити слугує розклинювальний механізм її переміщення. Цей механізм забезпечує вертикальне переміщення натискної плити знизу вверх до жорстко закріпленої нерухомої плити зі штанцювальною формою. На сьогодні існують дослідження, присвячені удосконаленню розклинювального механізму, проте немає праць, які пов'язані з удосконаленням розклинювального механізму для забезпечення тривалішого перебування натискної плити в зоні штанцювання розгорток.

У статті проведено кінематичний аналіз існуючого та пропонованого розклинювальних механізмів привода натискної плити за допомогою програми SolidWorks. Пропонований механізм відрізняється від існуючого застосуванням додаткового веденого кривошипа, який переміщується за допомогою зубчатої передачі. Зубчате колесо закріплене на одному валу з веденим кривошипом та переміщується по зубатому сектору і так приводить у рух ведений шатун, а той своєю чергою натискну плиту.

Для побудови 3D-моделей механізмів проведено розрахунки геометричних параметрів механізмів на основі габаритних параметрів штанцювального преса. Результати отриманих даних в процесі моделювання досліджуваних механізмів у SolidWorks Motion подано графічно. На основі аналізу змін значень кінематичних характеристик виконано порівняльний аналіз функціонування механізмів привода натискної плити.

Ключові слова: *штанцювальний прес, картонна розгортка, механізм привода натискної плити, двокривошипний розклинювальний механізм, CAD/CAE modelling, кінематичний аналіз.*

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