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RESEARCH ON THE PRESSURE PLATE MOVEMENT BY THE DRIVE WITH WEDGING MECHANISMS IN THE DIE-CUTTING PRESS

V. V. Vlakh, Yu. Yu. Mykhailiv

Ukrainian Academy of Printing,
19, Pid Holoskom St., Lviv, 79020, Ukraine

The increase in the volume of paper and cardboard production in the world and the fastest growth of the paper and cardboard packaging segment in the general market of packaging materials are noted. It is emphasized that to meet the market's needs, container manufacturers are expanding the technological equipment park. Its important component is the die-cutting equipment, intended to produce flat sweeps from cardboard blanks. It is noted that during die-cutting presses, the pressure plate oscillates. It can negatively affect the quality of the product production and the operational capabilities of the equipment. The analysis of the scientific literature did not give results of thorough research on pressure plate movement during working and idling cycles. The scheme of the pressure plate drive using wedging mechanisms is proposed. Their geometric synthesis is performed (on the example of the left contour), and the current relative movement of the pressure plate by the wedging mechanisms of the left and right contours is researched. The oscillating motion of the pressure plate is studied.

Keywords: *die-cutting press, pressure plate, wedging mechanism, crank, connecting rod, cardboard blank, oscillate move.*

Problem Statement. The volume of paper and cardboard production for packaging in the world has increased by almost 30% in the last ten years. Marketing agencies predict that the paper and paperboard segment of the overall packaging market will grow the fastest due to its efficiency, innovation, adaptability, and sustainability, which add value to food and industrial products. The European paper and cardboard packaging market is expected to grow at a rate of 4.8% during 2022-2027 [1].

To meet the market's needs, container manufacturers are expanding their technological equipment park, an important one of which is die-cutting equipment designed to produce flat blanks from cardboard. Die-cutting presses, which are components of such equipment, are equipped with wedging lever mechanisms for driving the pressure plate [2].

During the press operation, the mechanism of the drive's right and left wedging circuits ensures a strictly horizontal position of the pressure plate only at the moment of completion of the die-cutting process. Its raising and lowering are accompanied by oscillation moves [3].

The die-cutting press in the upper part contains a horizontal stationary support plate 1 (Fig. 1) with a fixed flat die-cutting mold 2. In the lower part of the drive of the pressure plate 3 with a cardboard blank CB, the press is equipped with wedging mechanisms of the left LC and right RC contours. The mechanisms consist of cranks 4.1 and 5.1, fixed oppositely on the drive shaft (not marked in the figure); driving connecting rods 4.2, 5.2; driven rods 4.3, 5.3 and driven connecting rods 4.4, 5.4, which are hingedly connected to the pressure plate 3 from the bottom side.

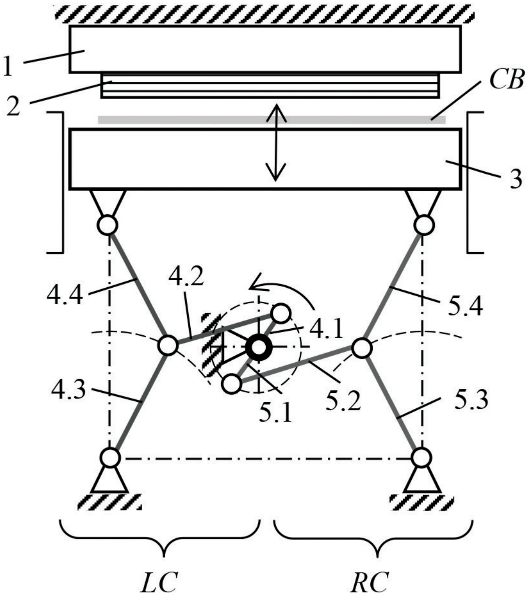


Fig. 1. Kinematic scheme of a die-cutting press equipped with a pressure plate drive with wedging mechanisms of the left and right circuits

The beginning of die-cutting blanks, especially those made of corrugated cardboard with an inclined pressure plate, can lead to angular crumpling of the ejector cushions of the die-cutting mold with subsequent displacement of the blanks relative to its tools and, as a result, to the production of low-quality products.

Analysis of recent research and publications. In work [4], the analysis of the functioning of the wedging mechanisms in the drive of the pressure plate of the die-cutting press was performed, and it was established that it oscillates during movement. It is noted that the non-parallel movement of the pressure plate when overcoming a significant technological resistance (due to die-cutting the blanks) causes wear of the sliders and can cause jamming in the guides. The difficulty of setting up the press is due to the need to equalize the loads over the entire plane of the plate. At the same time, no studies were conducted to evaluate the quantitative parameters of the movement of the pressure plate through different contours of the wedging mechanisms.

The graphs of the relative movement of the sliders of the pressure plate, given in the paper [5], confirm the fact that the right slider lags behind the left one on the return stroke and, when lifting (the working stroke) catches up with it and moves to the working area of the sweep die-cutting almost simultaneously with the left one. At the same time, the planes of the pressure plate and the tools of the die-cutting mold are aligned and occupy a parallel position. The reason for this phenomenon, which consists of an angular arrangement of the interbase axes of the left and right wedging contours, is given. However, the work presents only graphic dependences of the linear movement of the left and right sides of the pressure plate, which does not reveal the absolute values of the angle of its oscillation.

In the work [6], the authors claim that operating die-cutting presses using the pressure plate drive wedging mechanisms proves that its oscillating movement occurs during the working and idle strokes. It is noted that it negatively affects the stability of the die-cutting press and the machine. It is proposed to use a drive using screw-nut gears instead of wedging mechanisms to convert the rotary motion of the screws into the translational motion of the pressure plate to ensure the minimization of negative consequences.

Another technical result, as shown by the work materials [7], is the replacement of wedging mechanisms in the pressure plate drive with an eccentric one. In this case, the axes of the rolling roller bearings are located relative to the axes of symmetry of the movable pressure plate at an angle of 45°, which ensures strict horizontal movement of the pressure plate and uniform pressure over its entire plane.

As can be seen from the analysis of the publications, in one case, the study of the pressure plate movement in the die-cutting presses was not conducted thoroughly; in the other - the studies were directed at replacing the wedging mechanisms of the drive.

The article aims to synthesize the wedging mechanisms of the left and right contours of the pressure plate drive in the die-cutting press to investigate its relative movement and angular oscillation.

Presentation of the main research material. For the synthesis of the wedging mechanisms of the left and right contours of the pressure plate drive in the die-cutting press and the study of its movement, the following is adopted:

designation of relative parameters:

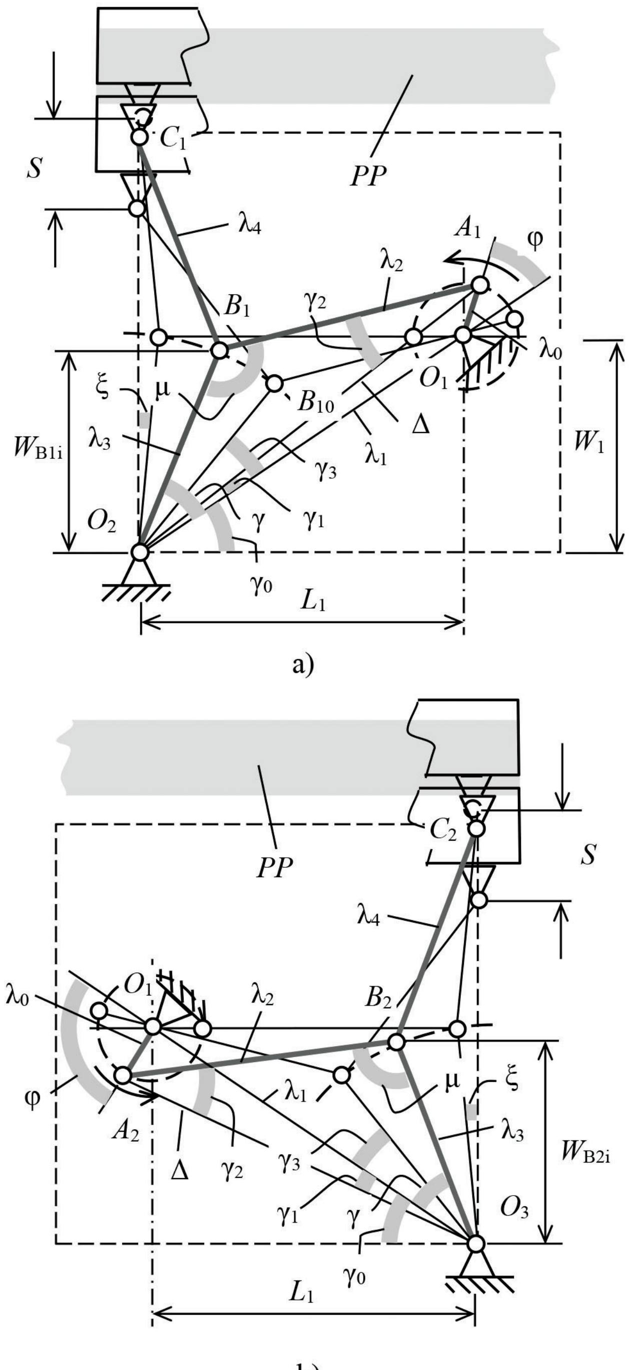
- $S = 1{,}0$ – maximum movement of the pressure plate PP (Fig. 2);
- λ_0 – crank radius;
- λ_1 – interbase distance between axes O_1 i O_2;
- λ_2 – the length of the connecting rod of the driving circuit;
- λ_3 – the length of the driven rod of the driving circuit;
- λ_4 – the length of the connecting rod of the driven circuit;
- $L_1 = 3{,}125$ – horizontal distance between axes O_1 i O_2;
- $W_1 = 3{,}125$ – vertical distance between axes O_1 i O_2;

designation of absolute parameters:

- φ – current crank angle;
- $\xi = 5°$ – the angle between the driven rod of the driving circuit and the vertical axis in the uppermost position of the pressure plate.

Geometric synthesis of the wedging mechanism (on the example of the left contour).

The task of synthesizing wedging mechanisms consists of substantiating the relative sizes of the crank λ_0 and the driving connecting rod λ_2, provided that the other components are specified. A system of equations was obtained to find the needed parameters:



a)

b)

Fig. 2. Schemes for the geometric synthesis and calculation of the wedging mechanisms of the pressure plate drive of the left (a) and right (b) circuits

$$\begin{cases} \lambda_3 \cdot sin\xi + \lambda_2 + \lambda_0 = L_1, \\ \lambda_2 - \lambda_0 = \sqrt{\lambda_3^2 + \lambda_1^2 - 2\lambda_3 \cdot \lambda_1 \cdot cos\gamma_3} \end{cases}. \tag{1}$$

In the second equation of the system, $\gamma_3 = 11.81°$ is the angle between the driven rod λ_3 (in the lowest angular position) and the interbase axis λ_1. It was found according to known geometric parameters: S = 1.0; $W_1 = 3.125$ and $\gamma_0 = 45°$.

The research of the relative dimensions of the rocker arm λ_3 (Fig. 3) and the driven connecting rod λ_4 is based on the consideration that the completion of die-cutting of cardboard blanks is realized if there is an acute angle ξ between the driven rod and the vertical axis O_2C_1; the relative geometric dimensions of the driven rod λ_3 and the driven connecting rod λ_4 are the same. Then $\lambda_3 = W_1/cos\xi = 3{,}125/\cos 5° = 3{,}137$.

The equations system (1) solutions are: $\lambda_0 = 0.679$ and $\lambda_2 = 2.173$.

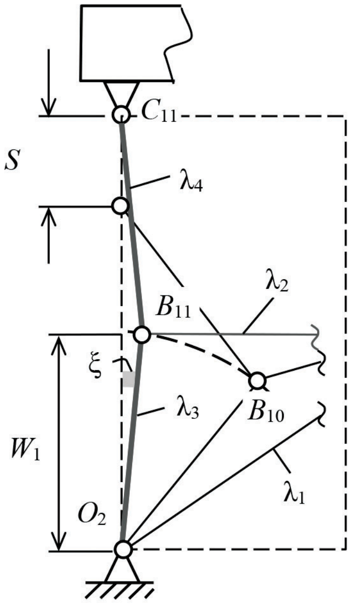


Fig. 3. Scheme for calculating the relative dimensions of the component wedging mechanisms of the pressure plate drive (on the example of the left circuit)

The research of the pressure plate movement.

To study the movement of the pressure plate during the working and idling phases, we will calculate the wedging mechanisms of the left and right circuits. The current values of the relative movement of the left side of the pressure plate (hinged connection C_1 of the driven connecting rod λ_4 with the pressure plate) and the right side (hinged connection C_2 of the driven connecting rod λ_4 with the pressure plate) are determined by the difference:

$$S_{C1i} = 2\left(W_{B1i} - W_{B10}\right), \tag{2}$$

$$S_{C2i} = 2\left(W_{B2i} - W_{B20}\right), \tag{3}$$

where $W_{B10} = W_{B20} = \lambda_3 \cdot sin(\gamma_0 + \gamma_3)$ – the relative initial position of the hinge joints of the driving connecting rods and driven rods relative to the O_1 and O_2 axes; here γ_3 is the angle between the axis λ_1 and the driven rod λ_3 in its lowermost position (corresponds to the initial position of the pressure plate), which is found by the expression:

$$\gamma_3 = arccos\frac{\lambda_1^2 + \lambda_3^2 - (\lambda_2 - \lambda_0)^2}{2\lambda_1 \cdot \lambda_3} = 11{,}81°;$$

W_{B1i}, W_{B2i} – relative vertical current distances of the hinge joints of the driving connecting rods and driven rods relative to the axes O_1 and O_2, which are determined by dependencies:

$$W_{B1i} = \lambda_3 \cdot sin(\gamma_{01} + \gamma), \tag{4}$$

$$W_{B2i} = \lambda_3 \cdot sin(\gamma_{01} + \gamma). \tag{5}$$

To find the current value of the angle γ between the driven rod λ_3 (Fig. 2a) and the interbase axis λ_1 (based on the left wedging contour), we use the dependencies:

$$\gamma = \pi - \mu - \gamma_1 + \gamma_2, \tag{6}$$

$$\gamma = \pi - \mu - \gamma_1 - \gamma_2, \tag{7}$$

(dependence (6) is valid for the case of the location of the crank λ_0 above, and dependence (7) – below the interbase axis λ_1) where:
μ – the angle between the rocker arm λ_3 and the driving connecting rod λ_2. The formula determines its current value:

$$\mu = arccos\frac{\lambda_2^2 + \lambda_3^2 - \Delta^2}{2\lambda_2 \cdot \lambda_1}; \tag{8}$$

γ_1 – the angle between the axis λ_1, which connects the base hinges O_1 and O_2, and the variable axis Δ [8]. The formula determines its current value:

$$\gamma_1 = arccos\frac{\Delta^2 + \lambda_1^2 - \lambda_0^2}{2\Delta \cdot \lambda_1}; \tag{9}$$

γ_2 – the angle between the driving connecting rod λ_3 and the variable axis Δ connecting the hinges A_1 and O_2. The expression finds the value of the angle:

$$\gamma_2 = arccos\frac{\lambda_2^2 + \Delta^2 - \lambda_3^2}{2\lambda_2 \cdot \Delta}. \tag{10}$$

The current values of the Δ axis included in dependencies (2), (3), and (4) can be found using the formula:

$$\Delta = \sqrt{\lambda_1^2 + \lambda_0^2 + 2\lambda_1 \cdot \lambda_0 \cdot cos\varphi}, \tag{11}$$

where φ – the current rotation angle of the crank λ_0 relative to the interbase axis λ_1.

Based on the obtained values of the angles μ, γ_2, and γ_1, the current angles γ of the inclination of the driven rod λ_3 to the interbase axis λ_1 were calculated, which are graphically depicted in Fig. 4.

As it can be seen from the graph, the minimum value of $\gamma = 11.85°$ is recorded for the crank rotation angle of $\varphi = 330°$, and the maximum value of $\gamma = 40°$ for $\varphi = 135°$. The curve is steep during the crank rotation phase of $330° \leq \varphi \leq 135°$ (working stroke of the pressure plate) and sloping during the phase of $135° \leq \varphi \leq 330°$ (idle stroke of the pressure plate).

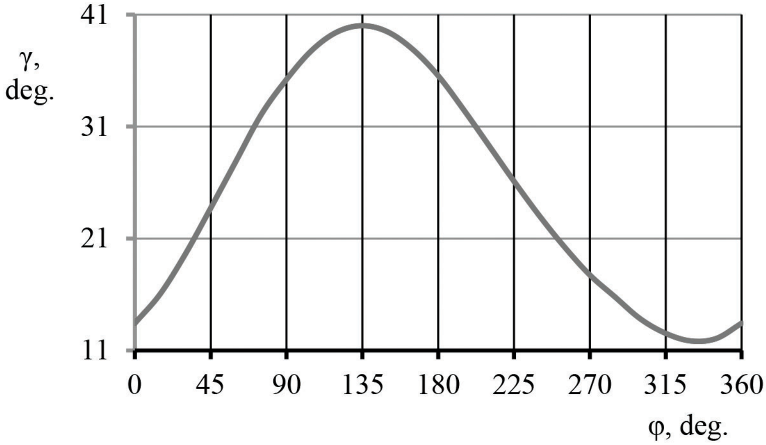


Fig. 4. Graph of the dependence of the angle of inclination of the driven rod to the interbase axis of the wedging mechanism of the left circuit on the crank rotation angle

The obtained current values of the tilt angle of the driven rod to the interbase axis are used to calculate the relative movement of the pressure plate by the wedging mechanisms of the left and right contours according to dependencies (2), (3), which are graphically depicted in Fig. 5.

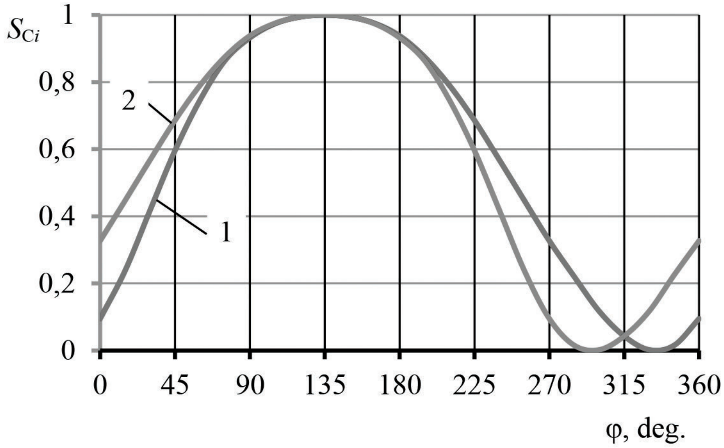


Fig. 5. Graphs of the dependence of the relative movement of the pressure plate by the wedging mechanisms of the left (1) and right (2) contours on the crank rotation angle

As it can be seen from the graphs, the minimum current value of $S_{c1i} = 0.095$ for the left circuit and the maximum $S_{c2i} = 0.327$ for the right circuit is recorded at the initial position of the crank, which causes the angular displacement of the pressure plate. An increase in φ leads to a smooth alignment of the values of the pressure plate's relative movement by the contours' wedging mechanisms. For φ = 135°, it occupies a strictly horizontal position. Further rotation of the crank causes an advance of the current values of the relative movement of the pressure plate along the left contour relative to the right one. However, for φ = 135° $S_{c1i} = S_{c2i} = 0.015$, which provides the pressure plate with a strictly horizontal position.

To find the angle of inclination of the pressure plate, we use the expression:

$$\alpha = arctg\frac{S_{C1i} - S_{C2i}}{2L_1}. \qquad (12)$$

The graph of the dependence of the angle of inclination of the pressure plate on the angle of rotation of the crank is shown in Fig. 6.

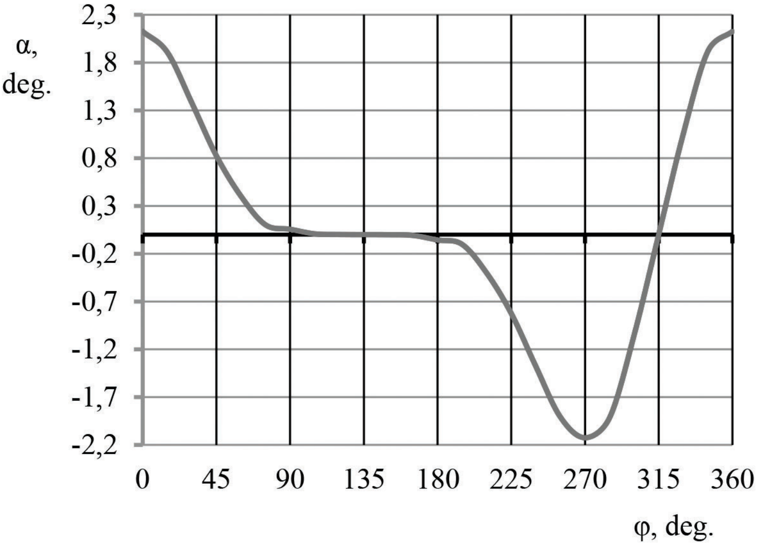


Fig. 6. Graph of the dependence of the angle of inclination of the pressure plate on the crank rotation angle

As it can be seen from the graph, the maximum positive angular displacement α = 2.125° is recorded in the initial position of the crank. An increase in φ causes the opposite change in α. So, for φ = 90° α = 0.056°, and in the position of the crank φ = 135° the pressure plate occupies a strictly horizontal position. Further, the angular movement of the crank is accompanied by a negative angular displacement of the pressure plate: for φ = 180° α = – 0.056°, and at the position of the crank φ = 270° the maximum angular displacement α = –2.125° is obtained. After the peak of the angular displacement, the pressure plate turning the crank is gradually leveled, and the position φ = 315° takes a strictly horizontal position.

Conclusions. Packaging experts predict that the paper and cardboard segment of the overall packaging market will grow the fastest due to its efficiency, innovation, adaptability, and sustainability. To meet the market's needs, container manufacturers are expanding the technological equipment park, an important one of which is die-cutting equipment. During the operation of the press during the cycle, the mechanisms of the drive's right and left wedging circuits ensure a strictly horizontal position of the pressure plate only at the moment of completion of the die-cutting process, and oscillations accompany its raising and lowering. The analysis of publications proves that the movement of the pressure plate in die-cutting presses has not been studied.

According to the results of the geometric synthesis of wedging mechanisms, the relative dimensions of the crank and the driving connecting rod are established. In the initial position of the crank, the minimum current value $S_{c1i} = 0.095$ of the relative movement of the pressure plate in the left circuit and the maximum $S_{c2i} = 0.327$ in the right circuit are found analytically. The pressure plate is strictly horizontal for the crank rotation angle $\varphi = 135°$.

The angular oscillation of the pressure plate is evaluated: in the initial position of the crank, the maximum positive angular displacement $\alpha = 2.125°$ is recorded; for $\varphi = 135°$, the pressure plate occupies a strictly horizontal position; in the position of the crank $\varphi = 270°$, the maximum negative angular displacement $\alpha = –2.125°$ is obtained.

The results obtained from the research will be used to improve the mechanisms of the pressure plate drive in die-cutting presses.

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ДОСЛІДЖЕННЯ ПЕРЕМІЩЕННЯ НАТИСКНОЇ ПЛИТИ ПРИВОДОМ З РОЗКЛИНЮВАЛЬНИМИ МЕХАНІЗМАМИ У ШТАНЦЮВАЛЬНОМУ ПРЕСІ

В. В. Влах, Ю. Ю. Михайлів

Українська академія друкарства,
вул. Під Голоском, 19, Львів, 79020, Україна
vlakh.v.v@gmail.com

Констатовано ріст обсягів виробництва паперу і картону у світі та найшвидше зростання сегмента паперової та картонної тари на загальному ринку засобів пакування. Наголошено, що для задоволення потреб ринку виробники тари розширюють парк технологічного обладнання. Важливою його складовою є штанцювальна техніка, яка призначена для виготовлення плоских розгорток з картонних заготовок. Зауважено, що у процесі експлуатації штанцювальних пресів натискна плита здійснює коливний рух. Він може негативно вливати на якість виготовлення продукції та експлуатаційні можливості обладнання. Аналізом наукової літератури не виявлено ґрунтовних досліджень характеру її руху протягом робочого і холостого ходів.

Запропоновано схему привода натискної плити з використання розклинювальних механізмів. Усі розміри ланок механізмів виражено в частках лінійного переміщення натискної плити. Для дослідження її переміщення розклинювальними механізмами лівого та правого контурів виконано їх геометричний синтез. За його допомогою обґрунтовано відносні розміри кривошипів і ведучих шатунів. Відносні розміри *коромисел і ведених шатунів отримано на основі припущення, що завершення штанцювання картонних заготовок реалізується за умови наявності гострого кута між коромислами і вертикальною віссю.*

Для дослідження характеру руху натискної плити протягом фаз робочого і холостого ходів виведені математичні залежності для кожного контуру. Розраховано поточні кути нахилу коромисел до міжбазових осей, що уможливило оцінку поточного відносного переміщення натискної плити розклинювальними механізмами лівого та правого контурів. Досліджено коливний рух натискної плити протягом повного циклу її переміщення. Отримано максимальні значення її кутового зміщення для визначених кутів повороту кривошипів. Окреслено перспективу використання отриманих результатів дослідження.

Ключові слова: *штанцювальний прес, натискна плита, розклинювальний механізм, кривошип, шатун, картонна заготовка, коливний рух.*

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