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IMPROVEMENT OF THE PNEUMATIC DRIVE OF A FLAT DIE-CUTTING PRESS

For more efficient operation of the device, it is recommended to use cylinders with the lowest working pressure and a larger piston area to reduce the preparatory time. The pneumatic drive can significantly improve the cutting process due to the proposed design of the pressing plate with perforation grooves. The cutting method is more environmentally friendly due to the use of compressed air as the primary working force. The operational efficiency of the drive is ensured by the design features of the pressing plate, specifically its enhanced functionality through the introduction of a pneumatic chamber into its base. This new technical solution allows the creation of an air cushion analog during cardboard cutting, thus increasing the durability of the cutting tools.

Keywords: *cardboard blank; pressing plate; die-cutting press; pneumatic cylinder; compressed air; pneumatic chamber; reverse motion; piston, rod.*

Fig.: 3. References: 11.

Urgency of the research. A wide range of printing equipment is used for the production of cardboard packaging. Clamshell presses remain the primary type of equipment, as they are relatively simple to use. However, the main operating mechanism - the clamshell press - is highly loaded, energy-intensive, and requires significant modernization.

Target setting. Accelerated dulling of cutting elements is one of several issues encountered when using traditional die-cutting technology. The contact method of knife-against-anvil creates operational difficulties in the technological process of packaging production.

The task is to develop an energy-efficient drive for the pressing plate based on pneumatic cylinders.

Actual scientific researches and issues analysis. Pneumatic drives or systems include automatic control, stabilization, and regulation systems for working executive elements, which operate using compressed air [1-2]. Pneumatic cylinders with reciprocating motion of the output link have two fixed end positions of the piston. Due to the elasticity of the working medium, it is challenging to stop the piston at a specific position, even when the pneumatic supply and exhaust lines are closed. Thus, a rational choice of the pneumatic cylinder's structural dimensions and the use of regulating devices are necessary to improve their operational dynamics.

A geometric synthesis of the wedging mechanism of the pressing plate drive has been performed [3]. The work identified the effects of deformation in ejector cushions of the die-cutting form and the cutting and creasing of cardboard on the mechanism's component loads. The nature of the pressing plate drive's load during cardboard blank die-cutting was investigated.

Flat die-cutting presses are widely used in the printing and packaging industry. Significant technological loads occur in such presses. Therefore, the drives' designs are aimed at overcoming substantial technological forces in the pressing plate while applying minimal force to the input drive link [4, 5]. The goal is to ensure planar-parallel movement of the pressing plate over large contact areas.

The advantages and disadvantages of existing wedging mechanisms for flat die-cutting presses were analyzed in [6]. A wedging cam mechanism for driving the lower movable pressing plate was developed. The technological loads for die-cutting operations were determined, and the driving forces in the proposed wedging mechanism were analyzed.

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The use of a combined lever mechanism eliminates the drawback of non-parallel pressing plate movement [7]. Thus, a combined lever mechanism was used for the die-cutting press's movable plate drive. It consists of two pairs of crank-slider circuits: driving and executing. Kinematic calculations confirmed that the sliders move at identical speeds, maintaining the plate's parallelism during movement.

To reduce additional forces on the movable pressing plate mechanism, the use of driving cams and wedging levers was proposed [8]. By selecting a periodic motion law for the executive links, die-cutting forces are reduced.

The values of torques on the drive shaft were experimentally established [9]. The influence of drive speed parameters on the pressing plate's force loads was determined. A screw-nut drive mechanism for the movable pressing plate was proposed in [10], establishing the impact of cardboard blank thickness on the dynamic loads in the drive. Increasing the cardboard thickness leads to higher loads in the pressing plate drive.

Uninvestigated parts of general matters defining. A new method of cardboard blank die-cutting and a pressing plate mechanism were proposed [11]. The non-contact die-cutting technology eliminates support contact elements. A pneumatic drive was suggested for the die-cutting process. However, this method requires further experimental research.

The research objective. The aim of this study is to improve the pneumatic drive of a flat die-cutting press for higher-quality die-cutting of cardboard blanks. It proposes replacing articulated cyclic mechanisms with an enhanced pneumatic drive. The use of pneumatic cylinders for the pressing plate aims to refine the drive and improve overall performance.

The statement of basic materials. The technological process of die-cutting cardboard blanks in a flat die-cutting press involves a pneumatic chamber filled with compressed air. The generated pressure acts on the cardboard sheet during knife penetration. Simultaneously, an air cushion holds the sheet as blades penetrate it. Separation of the sheet occurs due to elastic materials and ejector cushions.

The proposed drive includes four pneumatic cylinders (1) comprising the pressing plate drive station. Each pneumatic cylinder is connected to the plate via fastenings (2). To create an air cushion interacting with the sheet, the fastenings (2) are equipped with internal channels for compressed air supply (not shown in the diagram). The pressing plate body (3) contains perforated slots (5) for passing die-cutting blades (7). The cavity (4) creates the necessary pressure, while ejector cushions (6) hold the sheet in the die-cutting plane. Guides (8) in the combined drive regulate the pressing plate's height above the die-cutting plane.

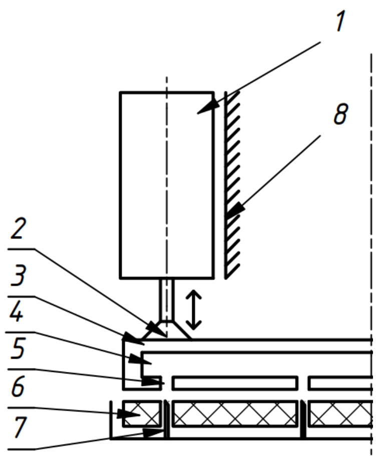

Fig. 1. Technological scheme of the pneumatic drive of a flat die-cutting press
Source: developed by the authors.

The dynamic operation of a standard double-acting piston pneumatic drive ensures the pressing plate's reverse movement according to the durations of strokes t_1, t_2, t_3, t_4. The sum of stroke durations constitutes the technological cycle period T_c of the pneumatic device, determining its productivity [11].

The primary goal of calculating a pneumatic drive is to select the effective piston area and the cross-sections of the supply f_n^e and exhaust f_v^e pipelines at a given piston speed, which will be considered constant and subject to a stable resistance force. In this case, it is necessary to account for the features of the improved pneumatic drive. Modeling the drive requires considering a specific combination of initial parameters, namely: the average piston speed υ_{avg}, its stroke S, the mass m_p of the moving parts, and initial conditions.

The first two assumptions are: 1) The pressure in the filling chamber equals atmospheric pressure, while the pressure in the exhaust chamber equals the mainline pressure. 2) The pressure in the common chambers is atmospheric.

Under the first class of conditions, the piston moves close to a stationary (uniform) state. Therefore, it is initially necessary to assess the feasibility of achieving such uniform (stationary) motion under the given initial parameters. The values of dimensionless parameters N, Ω, and χ determine the variation in operation speed. The dimensionless structural parameter N is calculated as follows: $N = 275.4 \cdot \mu_1 \cdot \frac{\mathrm{d}_1}{D_1^3} \cdot \sqrt{\frac{P}{p_м \cdot S}}$; where: μ_1 - the discharge coefficient of the exhaust line; d_1 - the exhaust line diameter (m); D – the piston diameter (m); P – the total resistance force on the rod (N); $p_м$ - the mainline pressure (MPa); S – the piston stroke (m). The dimensionless parameter Ω, which characterizes the throughput capacities of the supply and exhaust lines, is given by: $\Omega = \mu_n \cdot \frac{f_в}{\mu_в \cdot f_n} = \frac{f_v^e}{f_n^e}$; where: μ_n, $\mu_в$ – the discharge coefficients of the supply and exhaust lines, respectively, f_n^e, f_v^e are the effective areas of their cross-sections.

The dimensionless parameter χ, representing the load on the rod, is given by: $\chi = \frac{P}{p_м \cdot F}$; where $F = \pi \frac{D^2}{4}$ - the piston area on the side where the higher air pressure acts.

With sufficient accuracy for calculations, the condition for ensuring a piston motion mode close to steady-state can be expressed as: $\delta \le \delta_y$; where δ - inertia criterion of the pneumatic drive, numerically equal to:

$$\delta = \upsilon_{avg} \cdot \sqrt{\frac{m_p}{P \cdot S}} \tag{1}$$

where m_p - the mass of all moving parts of the pneumatic drive (piston, rod), and υ_{avg} - the average piston speed.

For the initial pressure conditions in the chambers, based on known experimental data, a critical value δ_y, has been established. Exceeding this value makes it impossible to achieve piston motion close to uniform. When using equation, it should be noted that the values of m_p and P are not yet known prior to selecting the diameter.

In the general case, the total resistance force P is equal to:

$$P = P_1 + P_{res} + P_g + p_a \cdot F_{ex}; \tag{2}$$

where P_1 - the frictional force in the piston and rod seals; P_{res} - the useful resistance force; P_g - the gravitational force of the piston and rod (considered for the vertical arrangement of pneumatic cylinders); F_{ex} - the opposing force due to pressure in the exhaust chamber. When the useful resistance force P_{res} dominates, the calculation of P_1 can be performed using empirical relationships: $P = 3.5\sqrt{P_{res}}$.

For selecting the parameters of the pneumatic drive based on a given piston speed υ_{avg} and resistance force *P* refined calculations are necessary. Additionally, for any arbitrary piston area *F* that must exceed the minimum value determined by the condition of providing sufficient driving force to overcome resistance, the drive can be tuned to the desired piston speed by selecting effective cross-sectional areas of the pneumatic lines at the inlet f_n^e and the outlet f_v^e . For each value of *F* multiple configurations are possible, characterized by different ratios of f_n^e i f_v^e .

The quantitative relationships between *F* f_n^e and f_v^e are described by dimensionless dependencies. The dimensionless effective cross-sectional area of the inlet line U_y is $U_y = f_n^e \cdot a_1$; where $a_1 = \dfrac{K \cdot p_m}{P \cdot \upsilon_{avg}}$ - proportionality coefficient.

As the parameter Ω, characterizing the throughput of the inlet and exhaust lines, increases while keeping other input data unchanged, the value of f_n^e decreases. Conversely, as Ω, decreases f_n^e increases. This is explained by the reduction in backpressure in the exhaust chamber, which impedes the piston's motion as the cross-sectional area of the exhaust channel increases relative to the inlet channel.

We determine the effective minimum cross-sectional area $f_{n.\min}^e$ of the pipeline: $f_{n.\min}^e = \dfrac{U_{y.\min}}{a_1}$, which delivers compressed air to the cylinder, and the optimal piston area F_{onm} : $F_{onm} = \dfrac{\frac{1}{\chi_{onm}}}{a^2}$, which corresponds to the cylinder diameter:

$$D = \sqrt{\frac{4 \cdot F_{onm}}{\pi}} . \tag{3}$$

We round the value of *D* to the nearest larger size from the standard range. The rod diameter d_{rod} is selected according to recommendations: $d_{rod} = (0.2...0.3) \cdot D$.

The structural dimensions of the pneumatic line elements, i.e., the areas of their cross-sections *f*, are calculated based on the adopted connection scheme of the elements and considering the condition $f_n^e \geq f_{n.calc}$, where $f_{n.\min} = f_{n.calc}$ the effective cross-sectional area of the inlet line determined from the dynamic calculation of the pneumatic drive. Let us consider the simplest case, where this line includes only elements with identical cross-sectional areas connected in series: $f_1 = f_2 = f_3 = ... = f_n$. The equivalent length of the pipelines, equivalent to the entire line, is always greater than the physical length of the pipes in the real system.

Having calculated using the formula $f^{e}_{n.\min} = \frac{U_{y.\min}}{a_1}$, we select the types of pneumatic devices and determine the equivalent (effective) length of the pipeline L_{eq}. Then, for $f^{e}_{n.\min}$ and L_{eq} e find the flow coefficient μ_n and the cross-sectional area of the pneumatic line *f* using the formula: $f = \frac{f^{e}_{n.\min}}{\mu_n}$, and also the diameter of the pneumatic line d_y using the formula: $d_y = \sqrt{\frac{4 \cdot f}{\pi}}$. The obtained diameter d_y is rounded to the nearest larger value according to the standards.

In a double-acting pneumatic drive, after the pneumatic distributor is triggered, air is supplied from the main line through the pipeline to the working cylinder chamber. As soon as the distributor port opens, pressurized air starts moving. For some time, the opening of the distributor and the propagation of the compressed air pressure wave to the working cylinder occur simultaneously but end at different times. The pressure wave arises after the distributor port is fully open. Since the distributor ports in most pneumatic drives open quickly relative to the operating cycle, this does not lead to significant delays in operation. The time intervals for these operations can also be individually determined according to this assumption: t_{valve} - the valve activating (opening) time, and $t_{\text{pressure wave}}$ - the time for the pressure wave to propagate from the valve to the pneumatic cylinder.

The pressure in the working chamber of the pneumatic cylinder increases as soon as the valve opens, and the piston begins to move. This time is referred to as the working chamber filling time, t_{fill}. The sum of the listed time intervals is called the preparatory period, t_{prep}:

$$t_{prep} = t_{valve} + t_{\text{pressure wave}} + t_{fill}. \tag{4}$$

The relationship between the design parameters of the device determines whether the pressure rises monotonically or fluctuates during the piston movement period. The pressure in the chamber connected to the main pipeline increases to the level set by the technological conditions as soon as the piston completes its stroke, *S*. In the chamber of the second (exhaust) cylinder, the pressure decreases to atmospheric pressure.

The time for the pressure wave to propagate from the valve to the working pneumatic cylinder is determined by the known formula: $t_{\text{pressure wave}} = \frac{L_t}{a}$; where L_t – the length of the pipeline from the valve to the cylinder, and a is the speed of the air flow propagation and *a* - the speed of the air flow propagation.

The pressure at the inlet is constant and equal to the main pressure during the filling of the initial volume of the working chamber with compressed air. The compressed air volume is equal to the volume of the pipeline, and the flow coefficient can account for calculation errors.

In the initial phase, there is a simultaneous increase in pressure in the working chamber and a decrease in pressure in the exhaust chamber. In this case, the driving force overcomes the resistance forces, and the piston moves when a pressure differential is created in both chambers. To establish the appropriate pressure differential, it is necessary to determine when the working chamber should be filled, and when the compressed air should exit the exhaust chamber. The filling time of the working chamber with compressed air is determined by the formula:

$$t_{fill} = \frac{3.62 \cdot 10^{-3} \cdot V_{01}}{f \cdot \mu_n(\psi_1(\sigma_2) - \psi_2(\sigma_1))}; \tag{5}$$

where V_{01} – the initial volume of the working chamber, μ_n – the air flow coefficient in the pneumatic line; *f* – the cross-sectional area of the supply pneumatic line, and σ_1 and σ_2 – the relative initial and final pressures in the working chamber of the cylinder. The values of the functions $\psi_1(\sigma)$ and $\psi_2(\sigma)$ can be found graphically according to recommendations [1, 2].

Using nomograms allows selecting the calculated pressure in both chambers of the pneumatic cylinder at the moment when the piston begins to move. Since the nomograms are built in dimensionless quantities, we first consider the load χ, which is the ratio of the maximum force F the resultant force P, acting on the piston.

To determine the preparatory period time for a typical double-acting pneumatic drive, we use the results from dynamic synthesis as initial data. Accordingly, we have: piston diameterя D = 0.08 m; rod diameter d_{rod} = 0.02 m; piston stroke S = 0.03 m; air flow coefficient of the supply line μ_n = 0.08; air flow coefficient of the exhaust line $\mu_в$ = 0.26.

Next, we find the initial volumes of the working and exhaust chambers, taking into account the volumes of the pipelines from the cylinder to the valve. The initial volume of the working chamber is calculated according to the recommendations [11] and the dimensionless load on the piston is determined by the formula: $\chi_1 = \dfrac{P}{p_m \cdot F_1}$.

We determine the values of relative pressures at the moment the piston begins to move by first calculating: $\upsilon = \dfrac{V_1}{V_\upsilon} \cdot \Omega = \dfrac{f_\upsilon \cdot V_{01} \cdot \mu_\upsilon}{\mu_n \cdot f \cdot V_\upsilon}$. The dimensionless area parameter is: $\Pi_{2,1} = \dfrac{F_2}{F_1} = \dfrac{D^2 - d_{rod}^2}{D^2}$.

According to the aforementioned recommendations, for $\chi = 0.37$ and $\upsilon = 0.57$ we find $\sigma_{pd}^n = 0.97$ and $\sigma_{vd}^n = 0.32$ taking into account that $\Pi_{2,1} = 0.938$. By comparing the filling and emptying times of the working cylinder chambers, we choose the larger value.

Figure 2 shows the dependence of the deformation strength S of sheet material on the destructive force of the knife F. It is assumed that the destructive force of sheet material during stretching should be experimentally determined as the arithmetic mean of ten measurement results.

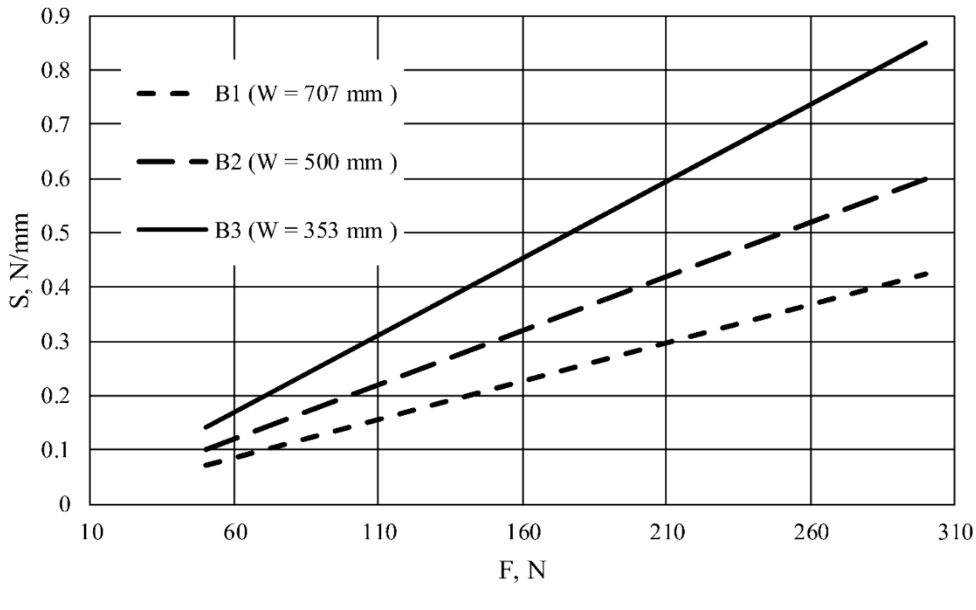

Fig. 2. Dependence of the ultimate deformation stretching strength S of sheet material on the destructive force F for format B1, B2 and B3

Source: developed by the authors.

For our theoretical calculations, we consider the deformation of the edge of the sheet when subjected to the cutting tool. In other words, the strength of the sheet material during stretching is calculated using the formula: $S=\frac{F}{W}$; where W – the width of the sheet material (mm), and F is the destructive force acting on the sheet (N). We also take into account the tensile strength limit of the sheet and its thickness. The rupture length is calculated using the following relation: $L=\frac{1}{9.8}\cdot\frac{F}{W\cdot g}\cdot 10^3$; where g – the mass of the paper per unit area (1 m^2).

The dependence of the deformation strength S (N/mm) on the destructive force F (N) under the action of the knife is examined for different sheet formats: $W_1=707mm$, $W_2=500mm$ and $W_3=353mm$. For the sheet format *B1* ($W_1=707mm$) the ultimate deformation strength *S* varies from 0.071 N/mm to 0.424 N/mm. For format *B2* ($W_2=500mm$) the value of *S* varies from 0.1 N/mm to 0.6 N/mm and for *B3* ($W_3=353mm$) the ultimate values of *S* range from 0.142 N/mm to 0.85 N/mm.

Thus, for the minimum ultimate edge rupture during punching of format *B3*, the value of *S* is 70.42% higher than when punching *B2*, and the maximum deformation value is 70.59% higher, respectively. Also, the minimum deformation rupture value when punching format *B2* is 71% higher than when punching *B1*, and the maximum value is 70.67% higher, respectively.

Figure 3 shows a graph the dependence of the pneumatic cylinder diameter *D* (mm) on the load indicator on the rod χ for the specified pressures: p_{m1}=0.1 MPa, p_{m2}=0.5 MPa and p_{m3}=1 MPa. The load on the rod χ is a dimensionless parameter. The indicator χ characterizes the force acting on the rod when a specified air pressure is simultaneously applied to the piston.

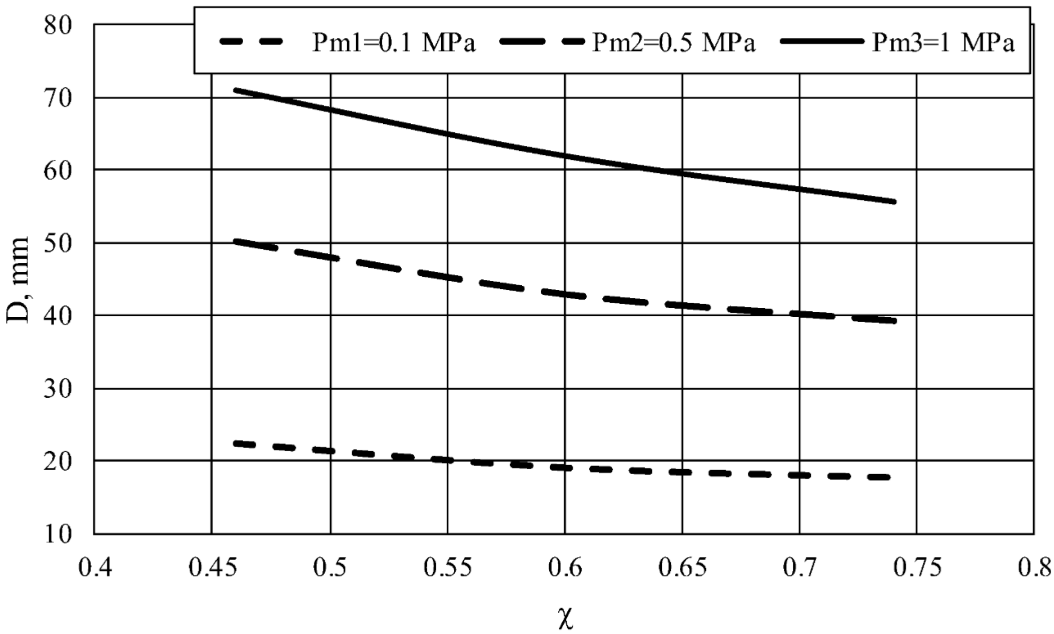

Fig. 3. Dependence of the pneumatic cylinder diameter D (mm) on the rod load indicator χ for the specified pressures: p_{m1}=0.1 MPa, p_{m2}=0.5 MPa and p_{m3}=1 MPa

Source: developed by the authors.

We consider the dependence of the pneumatic cylinder diameter *D* on the dimensionless rod load indicator for different pressures: p_{m1}=0.1 MPa, p_{m2}=0.5 MPa and p_{m3}=1 MPa. For a pressure of 1 MPa, the diameter *D* changes from 70.94 mm to 55.75 mm. For a pressure of 0.5

MPa the diameter D changes from 50.16 mm to 39.4 mm, and for pressure of 0.1 MPa the diameter D changes from 22.43 mm to 17.63 mm.

For the graph with a pressure of 0.1 MPa, we observe a difference of 44.7% compared to 0.5 MPa, and for 0.5 MPa, comparing with 1 MPa, the difference is 70.7%. The study shows a sharp change in the determined diameters depending on the pressure that will be supplied to the pneumatic cylinder chamber.

Analyzing the graphs, we note that as the cylinder diameters decrease, the excess load applied to the rod increases, which may lead to the eventual failure of the drive components.

Conclusions. The drive is a critical part of a cardboard blank manufacturing machine, directly impacting the quality of printed products made from cardboard.

The improvement involves using four supporting pneumatic cylinders that simultaneously lower the pressing plate. Synchronization of the pneumatic cylinders occurs by supplying air from a single system to all cylinders.

Preliminary analytical calculations provide an overview of the die-cutting technological process using a pneumatic drive. Key technical parameters include cylinder volumes, compressed air pressure, modeled pneumatic line diameters, and piston dimensions.

For more efficient operation, it is recommended to use cylinders with the lowest operating pressure and larger piston areas to reduce preparation time. The pneumatic drive significantly enhances the die-cutting process through the proposed pressing plate design with perforated slots. This method is more environmentally friendly due to the use of compressed air as the primary working force. The drive's functionality is ensured by the pressing plate's structural features, particularly its extended capabilities through the integration of a pneumatic chamber. This new technical solution enables the creation of an air cushion during cardboard die-cutting, thereby increasing the longevity of cutting tools.

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УДОСКОНАЛЕННЯ ПНЕВМАТИЧНОГО ПРИВОДА ПЛОСКОГО ШТАНЦЮВАЛЬНОГО ПРЕСА

Дослідження спрямоване на створення нового ефективного привода для натискної плити штанцювальної машини. Привод є ключовою частиною машини для виготовлення картонних розгорток і безпосередньо впливає на якість поліграфічної продукції виготовленої з картону.

Класичні механізми у приводах натискної плити мають недоліки. Вони громіздкі та споживають багато електроенергії. Тому метою роботи є розробка нового пневматичного привода пристрою та енергоощадного способу для виготовлення картонних розгорток.

Основне завдання полягає у детальному аналізі системи класичних приводів та їх удосконаленні. Удосконалення здійснюється за допомогою чотирьох опорних пневмоциліндрів, що одночасно опускають натискну плиту. Синхронізація пневматичних циліндрів буде відбуватися за рахунок одночасної подачі повітря з однієї системи в усі пневмоциліндри.

Попередні аналітичні розрахунки надають загальну характеристику технологічного процесу висікання з використанням пневматичного привода. Важливими технічними параметрами є об'єми циліндрів, тиск стисненого повітря, змодельовані діаметри пневмоліній та геометричні розміри поршня.

Для більш ефективної експлуатації пристрою рекомендується використовувати циліндри з найменшим робочим тиском і більшою площею поршня для скорочення підготовчого часу. Пневматичний привод може значно покращити технологічний процес висікання за рахунок запропонованої конструкції натискної плити з перфораційними пазами. Спосіб висікання є більш екологічним через використання як основної робочої сили стисненого повітря. Працездатність привода забезпечується завдяки конструкційним особливостям натискної плити, а саме розширеної її функціональності за рахунок введення в її основу пневматичної камери. Нове технічне рішення дозволяє створити аналог повітряної подушки при висікання картону і тим самим збільшити довговічність різальних інструментів.

Механіка технологічного процесу висікання картонних розгорток з використанням пневматичного привода пропонується вперше і потребує подальших наукових досліджень. Однак додаткове використання стисненого повітря у запропонованій технології для розділення картону буде позитивно відображатися на якості майбутньої поліграфічної і пакувальної продукції.

Ключові слова: *картонна розгортка; натискна плита; штанцювальний прес; пневмоциліндр; стиснене повітря; пневматична камера; реверсивний рух; поршень; шток.*

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