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## DESIGNING THE PRESSING PLATE MECHANISM OF THE FLAT DIE-CUTTING PRESS

The article proposes a pneumatic mechanism of the pressure plate of a flat die-cutting press.

**Keywords:** cardboard blanks; pressure plate; die-cutting press; pneumatic cylinder; compressed air; rod; piston; reverse movement; pneumatic chamber.

### Introduction

Taking into account the intensification of packaging processes, the performance of various technological operations for the production of cardboard containers is combined in one machine. The packaging machine is perceived as a technical system that performs basic, auxiliary and additional operations.

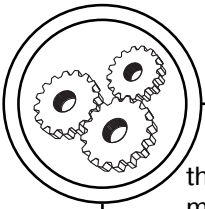
The technological operation of die-cutting cardboard blanks corresponds to the quality characteristics of future products. This is especially the application of the packaging industry — cardboard products. Therefore, a comprehensive approach to the selection of the drive of the pressure plate — the main executive mechanism that implements the process of making cardboard blanks — is an urgent issue.

Modern examples of packaging machine drives are complex technical systems. Drives designed according to the aggregate-module principle are often encountered. The development trend of packag-

ing machine drives provides that the latest models will be created on the basis of mechatronic functional modules. Each of them is a functionally and constructively independent product with a large number of interconnected characteristics and parameters.

In most cases, the main executive elements of the drive of the pressure plate of the flat die-cutting press are classic hinged lever mechanisms. They have a number of disadvantages: they create excessive vibration loads, the contact elements fail quickly, and they have bulky designs in general. They have a number of disadvantages: they create excessive vibration loads, the contact elements fail quickly, and they have bulky designs in general.

Since drive mechanisms are an integral and important component of printing and packaging machines, the quality of the process of cutting, perforating or creasing a cardboard blanks, will depend on their technical characteristics. Therefore,



the task is to design a new pneumatic type drive for a pressure plate with improved technical characteristics.

To solve the problem, it is necessary to make a comprehensive analysis of the known drive mechanisms of the pressure plate and, based on them, to improve the design through the use of pneumatic cylinders.

### Methods

A large range of packaging machines, which implements packaging of the product both in consumer condition and in transport cardboard containers, requires a different use according to the design of the pressure plate mechanisms [1]. In most cases, hinged cyclic lever mechanisms are required to move the pressure plate.

Kinematic studies of two-slider, wedging, eccentric-slider and eccentric mechanisms have been carried out [2]. In the wedging mechanism, the force of inertia of the mass of the pressure plate is maximally used in the process of die-cutting the cardboard blanks to overcome the technological resistance. The two-slider mechanism provides an almost constant relative acceleration of the pressure plate during the long rotation angle of the main shaft, which determines the die-cutting process.

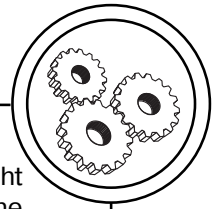
Schemes of the mechanisms of flat die-cutting presses are considered in the work [3]. The analysis is based on the principles of overcoming a significant technological effort on the pressing plate under the condition of applying a minimum force on the input drive link, ensuring strict plane-parallel movement of the pressing plate

with a significant area of contacting surfaces, relative simplicity of construction and low material consumption. Accordingly, on the example of a flat die-cutting press (crank type) with a leading eccentric link, positive and negative phenomena occurring in the drive were revealed. The use of a wedge press in the press mechanism provides better conditions for the transmission of forces than in the crank or eccentric mechanisms.

The leveraged wedge mechanism, which is used in die-cutting presses to drive the pressure plate, creates an oscillating movement during working and idle strokes. To reduce the negative consequences regarding the stability of the drive, a mechanism using a screw-nut transmission is proposed [4]. The mechanism converts the rotary movement of the screw into the translational movement of the pressure plate.

An analysis of the existing wedging mechanisms of flat die-cutting presses was carried out, highlighting their advantages and disadvantages [5]. To perform the punching operation, a wedging cam mechanism for driving the lower movable pressure plate has been developed. Theoretical calculations were carried out regarding the technological load of the die-cutting process and the magnitude of the driving force that arise in the proposed wedging mechanism.

Techniques for analytical assessment of costs of components of the kinetic power of the drive for overcoming technological loads have been developed [6]. The following parameters are considered in the work: technological resistance of stamping of cardboard blanks;



inertial loads caused by moving masses of drive elements and rolling friction in 'screw-nut' gears. When designing a pressure plate drive based on the use of a 'screw-nut' transmission, auxiliary mechanisms that perform reciprocating, reversible-rotating, reversible-rocking, plane-parallel, and rotary movements are considered. The work evaluates the kinematic parameters of the articulated four-link.

The comparative force characteristic of the drive mechanism of a movable pressure plate containing drive cams, rollers, springs and wedging levers is carried out in work [7]. The main criterion in the described force analysis is the reduction of all additional efforts that do not perform the technological operation of die-cutting. For the mechanism of the movable pressure plate, which contains the drive cam, spring, rollers and wedge levers, it is recommended to adopt the sinusoidal law of periodic motion, which is characterized by smaller indicators of additional efforts.

A number of experimental studies were conducted to determine the torques on the drive shaft of the combined drive mechanism of pressure plates [8]. Research was conducted when cutting along and across the fibers for different ranges of cardboard thickness. Research has experimentally confirmed the possibility of using the drive mechanism of the lower pressure plate in flat die-cutting presses.

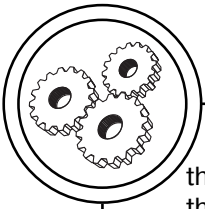
Application of 'screw-nut' transmission in die-cutting presses is studied in work [9]. Study of the components of the kinetic power of the pressure plate drive, which is spent on overcoming the technological resistance of cutting card-

board blanks, overcoming the weight of the pressure plate during the working stroke, and overcoming the inertial loads that occur when braking moving masses.

To eliminate the disadvantage of non-parallel movement of the pressure plate, it is proposed to use a combined lever mechanism [10]. A combined lever mechanism consisting of two pairs of crank-slider circuits is used to drive the movable plate of the die-cutting press: leading and executive. Kinematic calculations allow us to state that the sliders move at the same speeds, maintaining the parallelism of the pressure plate to the plane of the stationary support.

A method of theoretical calculation of the operating resource of the drive mechanism of the pressure plate of a flat die-cutting press has been developed [11]. The mechanism consists of a drive cam and rollers that are in constant contact with the help of a spring. The permissible operation of the working surfaces of the cam and rollers is established. The possibility of occurrence and violation of the geometric dimensions and correctness of the shape of the executive mechanisms was analyzed, which in the future may lead to a decrease in the productivity of the press and the appearance of additional dynamic loads.

Optimization of the geometric dimensions of the mechanism, in which the strokes of the right and left parts of the pressure plate will be identical, was carried out in work [12]. The influence of the mechanism parameters on the amount of travel of the pressure plate is analyzed in detail. The lever mechanism of the stamping press with



the same angular movements of the rocker arms has been improved. The proposed modernization of the mechanism will have a positive effect on the movement of executive parts.

An experimental evaluation of the force load of the eccentric mechanism of the movable pressure plate of the die-cutting press was carried out [13]. The proposed mechanism allows you to smoothly change the frequency of rotation on the drive shaft. Due to this, eccentric mechanisms avoid excessive loads arising in the drive of the movable pressure plate of the die-cutting press.

The influence of the speed parameters of the drive mechanism of the pressure plate of a flat die-cutting press on its power loads is determined [14]. The values of the torques on the drive shaft of the mechanism were experimentally established. It was established that with an increase in the frequency of rotation of the crank, a decrease in the torques on the drive shaft of the mechanism is observed.

Cardboard cutting based on the 'screw-nut' drive mechanism for the movable pressure plate has been researched [15]. The influence of the thickness of the cardboard blank on the value of the torque was determined experimentally. The dependence of the values of the torques on the drive shaft, which occur during cutting, on the thickness of the cardboard blanks has been established. It was established that with an increase in the thickness of the cardboard blank, the torque on the drive shaft of the pressure plate drive mechanism increases.

A new way of cutting contours in cardboard blanks is proposed

[16, 17]. The technology of a non-contact method of cutting cardboard blanks involves the removal of supporting contact elements such as: counter plate, marzan, counter knife, etc. To perform the technological process of cutting, a pneumatic module is offered for cutting contours in cardboard blanks.

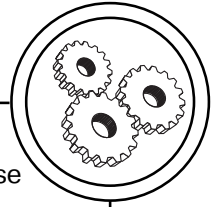
### Results

The aim of this study is to propose the use of pneumatic cylinders located at four support points with the possibility of reversible movement as an improved pneumatic drive of a pressure plate.

### Discussion

The technological scheme of the pressure plate of pneumatic drive is shown on figure 1. The pneumatic drive consists of four pneumatic cylinders *1*, which are connected at four support points to the pressure plate *10* due to hinge connections *9*. Compressed air (CA) is supplied to each pneumatic cylinder alternately through main pipelines *2* and *3*. The force created through the piston *4* and the rod *5* due to the pneumatic chambers *6* and *7* is transmitted to the cutting lines *13* mounted in the pressure plate. Guides *8* are intended for adjustment and fastening of pneumatic cylinders, and *11* are in charge of the accuracy of positioning of the executive elements of the pressure plate *10*.

This die-cutting method is implemented due to the contact of the cardboard sweep *12* with the die-cutting lines *13* and the ejector cushions *14*. In the lower position of the pressure plate, the die-cutting lines enter the grooves of the overhead table *15*, the pneu-



matic chamber of which (for uniform distribution of CA) is divided by the perforation wall 16.

The reciprocating movement of the pressure plate occurs due to the prepared compressed air through the pneumatic line. In this way, the CA is moved through the working zone of the two-way pneumatic cylinder through the rod cavity.

In turn, the created pressure operates on the piston and moves the rod. To determine the power characteristics, we will use the mass of moving parts  $m_m$ ; with the average speed of the piston  $v_{avr}$  and the distance  $S$ , which characterizes the height from the plane of the cardboard sweep to the die-cutting tools. Using the stroke of the piston with the force of useful resistance  $P_{use}$  and the pressure of compressed air  $p_m$ , we determine the friction force in the piston compaction  $P_1$ .

Accordingly:  $P_1 = 3,5 \cdot \sqrt{P_{use}}$ . In our case, due to the dominance of the useful force, we can use the transformed formula. Then the total resistance force:  $P = P_1 + P_{use}$ . To determine the inertia criterion of

the pneumatic drive, we will use the expression:

$$\delta = v_{avr} \cdot \sqrt{\frac{m_p \cdot p_m}{P \cdot S}} \quad (1)$$

To determine the maximum value of the piston speed at which the condition of equality would be fulfilled  $v_{avr} \approx v_y$ , where  $v_y$  the piston speed is uniform. Choosing a uniform piston speed and substituting into the formula, we get the maximum value of the piston speed:

$$v_{max} = \frac{\delta_y}{\frac{m_p \cdot p_m}{P \cdot S}} \quad (2)$$

Having chosen a random value of the area of the piston, we must comply with the condition that the area sought must be greater than the minimum value, which is determined from the condition of obtaining sufficient driving force to overcome the resistance forces. We simulate the drive of the pressure plate at a given speed by selecting effective cross-sections of the pneumatic pipelines at the inlet and outlet. For each area, we will have several

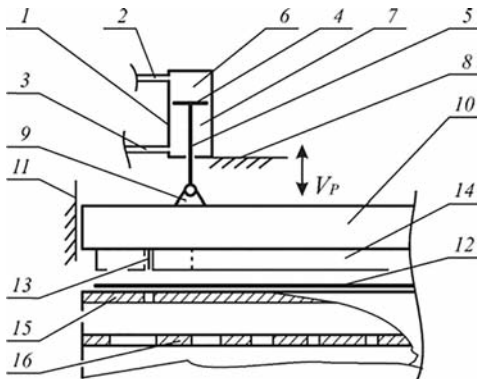
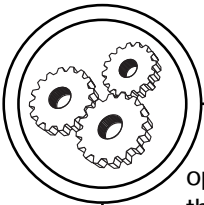


Fig. 1. The technological scheme of the pressure plate of pneumatic drive



options that we will explore through the ratio of dimensionless quantities.

According to the calculation method, we will use the speed of sound in a gas environment:

$$K = \sqrt{\frac{2 \cdot g \cdot k \cdot R \cdot T_{\text{gas}}}{k - 1}}, \quad (3)$$

where  $k$  — the adiabatic coefficient;  $R$  — universal gas constant;  $T_{\text{gas}}$  — the gas temperature and  $g$  — the gravitational acceleration.

To determine the cross-sectional area of the pipeline, we will use the first proportionality factor:

$$a_1 = \frac{K \cdot p_m}{P \cdot v_{\text{cep}}}. \quad (4)$$

According to the recommendations, for further calculations, we will choose the parametric area of the passage section of the supply line  $U_y$  and a dimensionless value characterizing the load on the rod, which we express through a dimensionless parameter  $\chi$  at the pressure on the piston.

To determine the optimal piston area, we calculate the second proportionality factor:  $a_2 = \frac{P_m}{P}$ . We

also determine the parametric effective cross-sectional area of the output line  $\Omega$ , taking into account  $f_n$  the effective minimum cross-sectional area of the pipeline that supplies the compressed air to the cylinder. Accordingly  $f_n = \frac{U_y}{a_1}$ .

We find the optimal area of the piston:  $F = \frac{1}{\chi \cdot a_2}$ . From this depend

ence, we will calculate the diameter of the cylinder:  $D = \sqrt{\frac{4 \cdot F}{\pi}}$ . Dia-

eters are standardized values, so we choose it from a number of standard sizes, rounding to the nearest tabular value.

We choose the diameter of the rod  $d_{\text{rod}}$  according to the recommendation  $(0.2-0.3) \cdot D$ . Accordingly:  $d_{\text{rod}} = 0.25 \cdot D$  and we round to the nearest standardized value.

The dimensions of the areas of the cross-sections are chosen according to the location of the elements one by one. We will consider the classic case where all elements in the line have the same cross-section. In parametric calculations, the length of the pipelines may, in some cases, be greater than the actual physical dimensions.

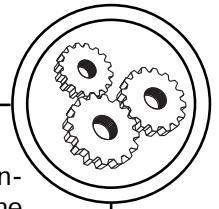
To calculate the equivalent length of the pipeline  $L_{\text{eq}}$ , we choose the length of the pneumatic line  $L_{\text{pl}}$  from the estimated standardized tabular values, the filter  $L_{\text{fil}}$ , the oil atomizer  $L_{\text{at}}$ , the air distributor  $L_{\text{dis}}$  and the throttle with the non-return valve  $L_t$ :

$$L_{\text{eq}} = L_{\text{pl}} + L_{\text{fil}} + L_{\text{at}} + L_{\text{dis}} + L_t. \quad (5)$$

Determine the diameter of the intake pipeline:  $d_v = \sqrt{4 \cdot \frac{f}{\pi}}$ , where

$$f = \frac{f_n}{\mu_n} \text{ the area of the output pneu-}$$

matic line,  $\mu_n$  — the cost factor of the supply line. The coefficient of air consumption in the main line is the main indicator responsible for the performance of the pressure plate and, accordingly, the die-cutting equipment. In each individual case, it will differ depending on the



design of pneumatic cylinders and the operating modes of the equipment as a whole.

Calculation of the propagation time of the pressure wave from the distributor to the working pneumatic cylinder:

$t_w = \frac{L_m}{a}$ , where  $a$  is the speed of sound spread in the air,  $L_m$  is the length of the pipeline from the cylinder to the distributor.

Calculation of the initial volume of the working cavity of the cylinder  $V_{wc}$ :

$$V_{wc} = \frac{\pi \cdot d_m^2}{4 \cdot L_m} + \frac{\pi \cdot D^2}{4 \cdot l}, \quad (6)$$

where  $l$  — the distance from the end of the piston to the cover of the pneumatic cylinder,  $d_m$  — the diameter of the pipeline from the cylinder to the air distributor.

Calculation of the initial volume of the output cavity  $V_{oc}$ :

$$V_{oc} = \frac{\pi \cdot d_m^2}{4 \cdot L_m} + \frac{\pi \cdot (D^2 - d_{rod}^2)}{4 \cdot (S - l)}. \quad (7)$$

Taking into account the previous calculation data, we will determine the optimal area of the piston  $F_1$ . For our case, we will substitute

the value  $F_1 = \frac{\pi \cdot D^2}{4}$  in the calculation formula for determining the dimensionless parameter that characterizes the force on the piston

$$\chi_1 = \frac{P}{p_m \cdot F_1}.$$

Calculation of the dimensionless parameter of the volume of the rod cavity  $\nu$ :

$$\nu = \frac{V_{pn}}{V_{oc}} \cdot \frac{\mu_0 \cdot f_d}{\mu_n \cdot f}, \quad (8)$$

where  $\mu_0$  — coefficient of the output line consumption;  $f_d$  — the area of the effective minimum cross-section of the line that diverts the compressed air from the cylinder.

Having calculated the value of the parametric area of the piston

$$P = \frac{D^2 - d_{rod}^2}{D^2}, \text{ it is possible to find}$$

the value of the relative pressures at the moment of the start of its movement. Let's calculate the time it takes to fill the working cavity with compressed air before the piston starts to move:

$$t_f = \frac{3,62 \cdot 10^{-3} \cdot V_{wc}}{f \cdot \mu_n \cdot (\psi_1 - \psi_2)}, \quad (9)$$

where  $\psi_1$  is a variable characterizing the relative initial pressure in the pipeline;  $\psi_2$  — a change characterizing the relative final pressure in the pipeline.

Knowing the time of spread of the pressure wave  $t_w$  from the distributor to the working pneumatic cylinder and the time of filling the working cavity  $t_f$  before the start of the piston movement, we determine the time of the preparatory period:  $t_{pr} = t_w + t_f$ .

Figure 2 shows the dependence of the optimal piston area on its load at a given compressed air pressure. The upper graph shows a sharp change in area indicators from 0.012 m<sup>2</sup> to 0.007 m<sup>2</sup>. The second graph is characterized by a relatively mod-

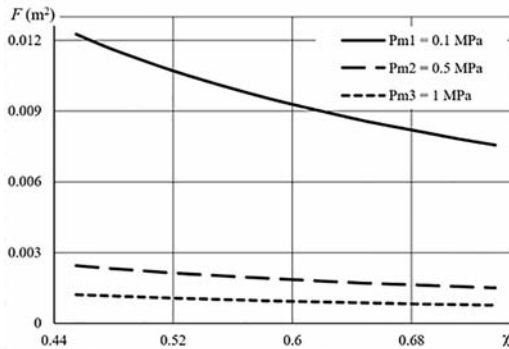
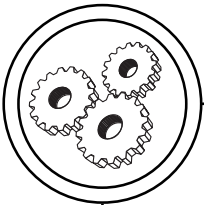


Fig. 2. Dependence of the optimal piston area  $F$  on its parametric pressure  $\chi$  for  $p_{m1} = 0.1$  MPa,  $p_{m2} = 0.5$  MPa, and  $p_{m3} = 1$  MPa

erate change in area from 0.002 to 0.001 m<sup>2</sup>. The same downward capacity is shown in the third graph (change in area from 0.001 m<sup>2</sup> to 0.0007 m<sup>2</sup>). All three curves are considered in the interval of the dimensionless parameter of pressure on the piston  $\chi$ , which increases from 0.45 to 0.73.

Having analysed the data of the graphs, we can come to a logical conclusion that it is more appropriate to use the values of the first upper graph in the pressure plate drive. Since the working diameters of the cylinder pistons will have the

greatest values. Accordingly, the larger the area, the larger the diameter will be, and the more technologically it is possible to explain the production of component parts. Namely, such technical phenomena as tightness, density and strength can be considered in a simplified form. For example, at the pressure in the main line  $p_{m1} = 0.1$  MPa and the diameter of the piston  $D = 124$  mm, the most favourable conditions will be observed in terms of minimal technological loads on the executive nodes of the pressure plate during its reverse movement. The

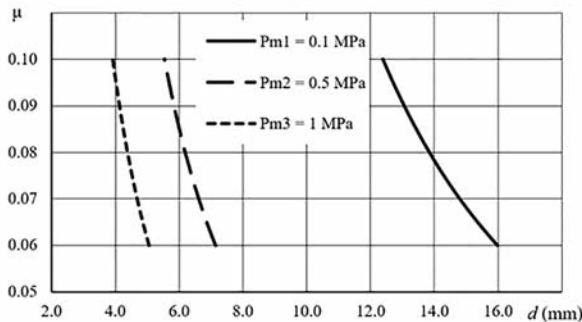
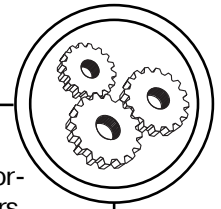


Fig. 3. Dependence of the compressed air consumption coefficient  $\mu$  on a number of pneumatic line diameters for:  $p_{m1} = 0.1$  MPa;  $p_{m2} = 0.5$  MPa and  $p_{m3} = 1$  MPa





structure will withstand greater loads and, as a result, accumulate greater pressure.

Figure 3 shows the dependence of the compressed air consumption coefficient  $\mu$  on the diameters of the pneumatic lines used in our drive at the required pressure set in the pipeline. Looking at the graphs, we can conclude that it is most appropriate to use the value of a separate graph with pressure  $p_{m1} = 0.1$  MPa in the pipeline. Since the proposed parametric values differ significantly from the previous two for  $p_{m2} = 0.5$  MPa and for  $p_{m3} = 1$  MPa due to the increased diameter of the pipeline. Therefore, the piston will accumulate more power and, accordingly, have greater technological strength and operating speed of the cylinder.

In the graphs, we can observe a sharp change in the compressed air consumption ratio from 0.1 to 0.06. For  $p_{m3} = 1$  MPa when the diameter changes from 3.91 to 5.05; for  $p_{m2} = 0.5$  MPa from 5.53 to 7.14 and at  $p_{m1} = 0.1$  MPa from 12.37 to 15.97. At the selected pressures in the pipeline, we observe a decrease in the consumption coef-

ficients of compressed air in proportion to the increase in their diameters.

Figure 4 shows the dependence of the preparatory time period  $t$  (s) on the variable pressure difference  $\psi$ . So, with a variable pressure difference  $\psi$  from 0.71 to 1.2, we see a smooth decline in the graphs. On the graph for  $p_{m1} = 0.1$  MPa, the preparation period decreases from 0.5 s to 0.2 s, on the second graph for  $p_{m2} = 0.5$  MPa, the preparation period decreases from 2.45 s to 0.95 s, and on the third graph for  $p_{m3} = 1$  MPa, the preparation period decreases from 4.89 s to 1.9 s.

After analysing the data of the graphs, we can come to the conclusion that it is more appropriate to use the  $p_{m1}$  value, since the preparatory period is characterized by the most optimal time intervals for the reversible movement of the pressure plate. Accordingly, the greater the difference between the initial and final pressure, the more time it takes to balance it.

Figure 5 shows the detailed dependence of the preparatory time period  $t$  (s) on the variable  $\psi$  in which, when the pressure difference changes from 0.71 to 1.2, it will be observed

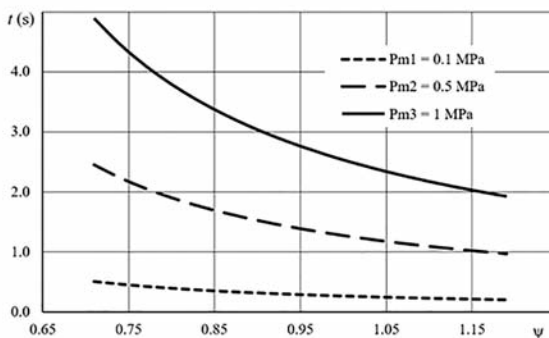


Fig. 4. Dependence of the preparatory time period  $t$  on the variable pressure difference  $\psi$  for  $p_{m1} = 0.1$  MPa,  $p_{m2} = 0.5$  MPa and  $p_{m3} = 1$  MPa

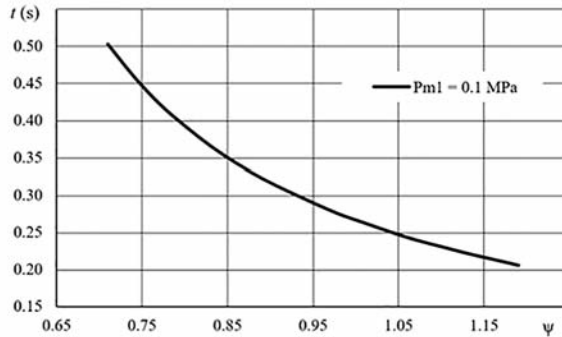
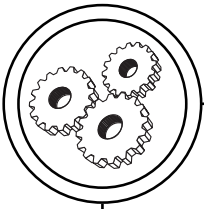


Fig. 5. Dependence of the preparatory time period  $t$  (s) on the variable pressure difference  $\psi$  for  $p_{m1} = 0.1$  MPa

a decrease in the graph. The arch-shaped form of the graph indicates that the preparatory period is the best suited for our case in comparison with the ones given above (fig. 4). Thus, the preparatory period is  $t = 0.2$  s for a variable pressure difference  $\psi = 1.2$ .

### Conclusions

The performed analytical calculations provide a general description of the technological process of cutting cardboard products by the proposed method. The selected power indicators of pneumatic cylinders for the pressure plate drive allow us to assert the expediency of their use. The main technical indicators of the improved technology are: working volumes of pneumatic cylinders, diameters of pipelines, calculated areas of the piston and rod of the pneumatic cylinder. The check of the performance of

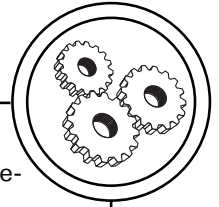
the pneumatic drive of the pressure plate is created due to the criteria of inertia and speed characteristics of the piston.

Recommended to use pneumatic cylinders with their relative lowest working pressures. In addition, each individual pneumatic cylinder must use the largest calculated area of the piston, as the selected geometric data will have a positive effect on the preparatory time period.

From the conducted research, it can be concluded that the replacement of cyclic actuators with pneumatic actuators of the pressure plate will significantly reduce the resulting vibration loads and simplify the overall design. As a result, create high-performance equipment and at the same time significantly reduce the cost of the construction of the pressure plate drive of machines specializing in the production of cardboard packages.

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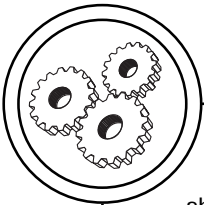
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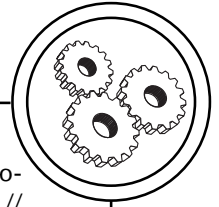
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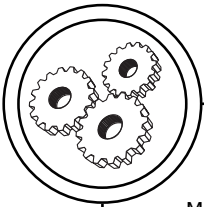
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**Завданням наукової роботи є детальний аналіз відомих класичних циклових приводів натискної плити та проектування на їх основі принципово нового пневматичного механізму для висікання картонних паковань.**

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

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