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CALCULATION OF UNIVERSAL ELASTIC COMPENSATOR APPLIED TO THE PRESSING-EXTRUSION OPERATIONS

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Abstract: *The mathematical model and program for the calculation of universal elastic rotary compensator of slide direction errors is considered. There developed program is includes the definition of the pressing force during technological operation, calculation of the shape factor and overlap coefficient, determination of the compensator design parameters. The results of modeling of the working of universal elastic rotary compensator of slide direction errors for pressing-extrusion operations are given as developed program for determination of the optimal angle of turn of the compensator upper elastic plate for execution of necessary rigidity.*

***Keywords**: universal elastic rotary compensator, polyurethane plate, «press-and-die» system, slide direction, pressing and extrusion operations, program for calculation*

1. INTRODUCTION

The work of press equipment with open frame is accompanied by errors in "press-and-die" system. Dimensions of those errors are increasing with rising of the technological force of forgings pressing-extrusion. Arising slide direction errors lead to the development of parasitic loads [1]. The result is deterioration in the quality of products and their size mismatch original drawings, the increase in the level of industrial noise and vibration, increased wear and tear directing and working parts of the stamp accompanied by downtime and requiring costly repairs, and sometimes the restructuring of the process when installing new equipment. After analyze of mechanical equipment work at the paper [2] is affirms a necessary of understanding of the sources and the effects of geometry errors in machines and calibration procedures for perform an error mapping and a subsequent compensation of geometric errors. There kinematic errors, thermo-mechanical errors, loads, dynamic forces, motion control and control program are determinate among the main sources that have influence on parts accuracy.

The reasons of deformation of the press-frames during punching, pressing or extrusion are explain at the research article [3]. The necessary of reducing of uncoaxiality in the “press-and-die” system is grounded for a better quality of production and decreasing of bending forces on the punches. That is possible by the ways of increasing of press frame rigidity or ejection of the compensation devices in a “press-and-die” system.

2. ANALYSIS OF RECENT RESEARCHES AND LITERATURE

For the reducing of harmful elastic deformation in the press equipment applies the different designs of mechanical compensators of the “press-and-die” system [4-6]. But application of the mechanical compensators is justificative at the precision forming of thin sheet parts only. There main imperfections of these devices are complex kinematic, necessary in extra quality at its manufacturing, bulky devices makes near the 1/3 of die space.

A promising direction to reduce of distortions in the “press-and-die” system is the using of different constructions of compensators, which are based on elastic elements (plates). These devices are installing between the top (upper) die-plate and bottom of the press slide. The design of compensating devices is depend on batch of parts production, type of the press equipment, force modes of the technological operations and other factors. This complicates is consistent of calculation of the design parameters of elastic compensators.

The compensators designed for some technological forces have become widespread, for example a compensator of errors of press slide direction as polyurethane plate that repeated of the configuration of the upper plate of the die [7]. This design is characterized by a variable rigidity by the length due to the presence of openings disposed with different steps in each direction. The effective using of such polyurethane based compensators is defined by its properties, because the elastic plate with upsetting to 25% withstands up $7 \cdot 10^6$ of loading cycles of compression. The operating results gave a reason to modify the design, thus improving its rigidity.

However, all of these devices have a fixed general rigidity resulting in limited range of process loads. In case of change of power modes while changing process operations or pressing another work-part is necessary to replace compensator with a new one with other work characteristics (rigidity, upsetting etc.).

With a wide range of products it is advisable to use universal error compensators of the “press-and-die” system that based on composite pre-stressed elastic elements configured in the form of two rigid (circular for example) polyurethane plates with openings [8]. One of these plates is displaceable by turning relative to the other, which enables to change the supporting surface area and leads to a change of compensator rigidity by varying of the overlap relation. That is set to extending the range of technological operations.

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The mathematical model of the work of universal double-layers elastic rotary compensator was developed in article [9] applied to the drawing-forming operation of sheet metal. The application of investigated devices to other forging and forming operation is requiring of the calculation of their design parameters with account of adaptation to the power and work processes in another technologies and equipment. Such investigation is not certain applied to the pressing-extrusion operations.

However, the using of universal rotary compensators is requiring of pre-specification of their design parameters and work characteristics. In this case the used methods for selection and optimization of the design parameters of elastic compensators of “press-and-die” system errors are demand of algorithmization of calculation methods and the development of automated program product with using the visual media object-oriented programming.

3. PURPOSE OF THE STUDY

The aim of this study is an adaptation of the mathematical model for the calculation of universal rotary compensator for pressing-extrusion operation applied to the open frame crank-presses and development of the program for automate calculations of optimal geometric dimensions and angle of rotation of the movable compensator plate to achieve the desired rigidity characteristic.

To achieve this aim requires:

a. To compile a mathematical model for calculation of the power modes of pressing-extrusion process for work-piece with solid circular profile.
b. To consider the procedure for calculating of the shape factor and geometry characteristics for universal elastic rotary compensator affecting the rigidity of the device;
c. To develop and test a program for calculation of the power modes and design parameters of the elastic universal rotary compensator.

4. MATHEMATICAL MODEL

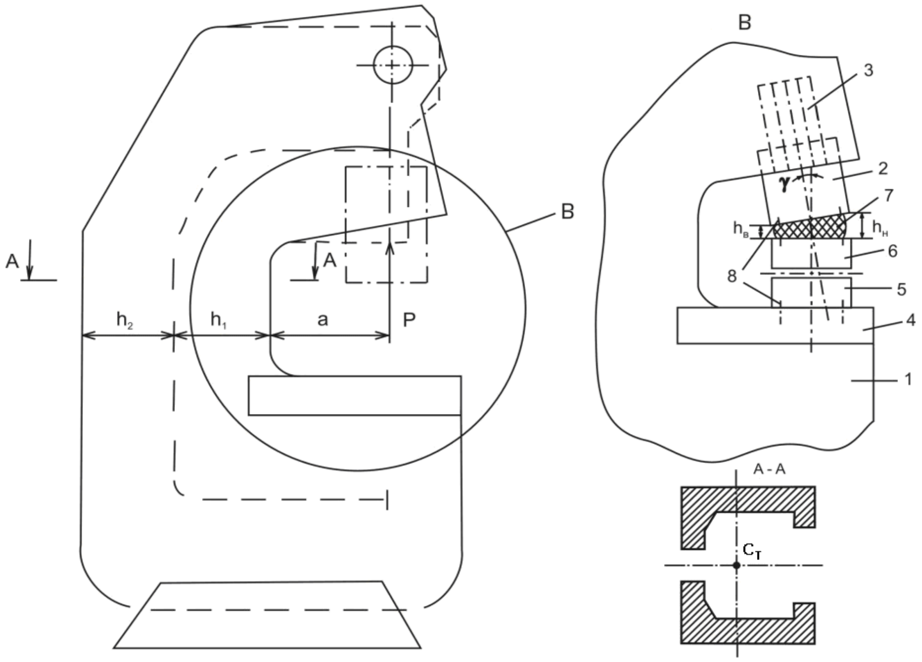
The compression pre-tests were carried with different shape samples from polyurethane brand SKU-PFL-100 (СКУ-ПФЛ-100, 100 Shore units of hardness). The elastic elements disposed axially between the supported surfaces of slide and top die plate of open frame crank press Fig.1.

During the processing of experimental data the value of compression pressure q was calculated as [9]:

$$q = P_d / F_{op},$$

where P_d – the deformation force;

F_{op} – the supporting area of the compensator.



1 - the press frame; 2 - the slide in the lowest position; 3 - guides; 5 and 6 - the upper and lower half of the die; 7 - elastic compensator; 8 - fixings;
γ - angular deformation of the frame opening;
h_b and h_H - height of upset elastic plate inside and the outside of the press; P - technological force generated by hydraulic loader;
a - the distance from the axis of load application to the press frame;
A-A - a cross-section of the press frame

Fig.1: Scheme and photo of the open frame crank-press with the elastic compensator

Dependence between the pressures during compression of polyurethane (q) and the degree of upsetting (ε) for $\varepsilon \in 0.2 \div 0.3$ is [9]:

$$q = 52 \cdot \varepsilon + 1.92 .$$

The shape factor (Φ) was determined by the formula [7, 9]:

$$\Phi = F_{bok} / F_{op} ,$$

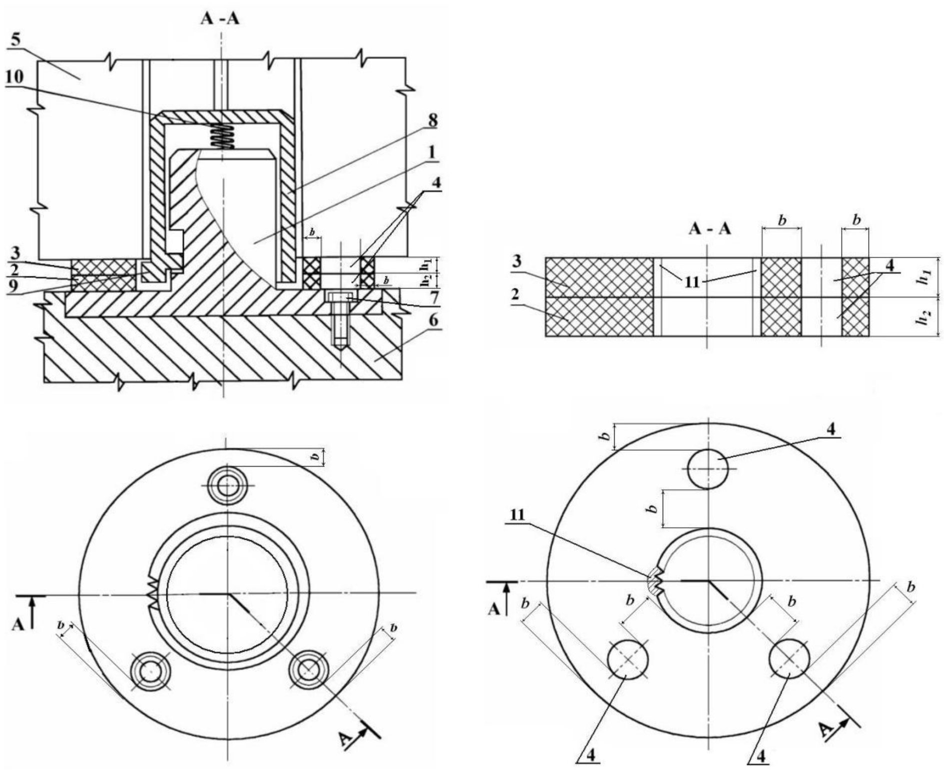
where F_{bok} – lateral surface area of the compensator.

The dependencies of the compression force from the shape factor and the degree of upsetting (compression) without lubrication (1) and lubricated (2) polyurethane plates were obtained on the basis of the developed mathematical model:

$$P = 0.001436 - 0.164223 \cdot \Phi - 0.289477 \cdot \varepsilon + 25.110390 \cdot \Phi \cdot \varepsilon, \quad (1)$$

$$P = 0.000434 - 0.00061 \cdot \Phi - 0.086781 \cdot \varepsilon + 0.128789 \cdot \Phi \cdot \varepsilon. \quad (2)$$

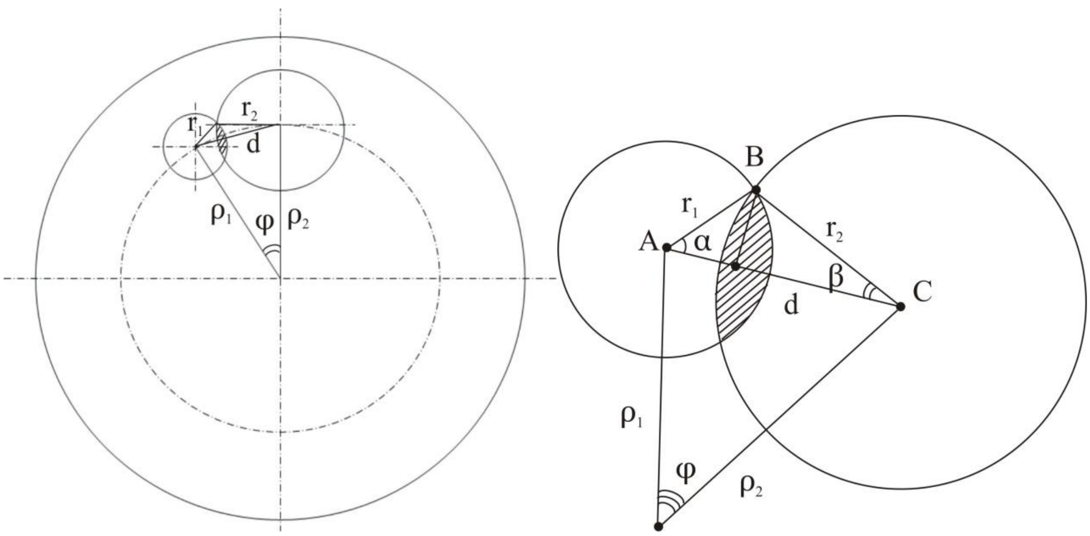
The design of universal compensator of the errors of slide direction was considered. There two elastic circular polyurethane plates with the radius (R) and openings with radiuses (r_1) and (r_2) where serves as the basis for composite pre-stressed elastic element (see Fig.2 and Fig.3).



1 – shank; 2 and 3 – the lower and the upper elastic circular plates; 4 – openings in the elastic plates; 5 – press slide; 6 – upper die-plate; 7 – screw connection the shank and the upper die-plate; 8 – cowl; 9 – the cog of the cowl for the fixing of the rotated elastic plate; 10 – spring; A-A - the designation of the cross-section; h_1 u h_2 – thickness of the upper and lower elastic circular plates; b – distances between openings, inner and outer backs of compensator

Fig.2: The universal compensator of the errors of the slide direction [1, 9]

The scheme for calculation of the overlapping of two openings of rotary elastic plates of universal compensator of the errors of slide directions is shown on Fig.3. This task is considered for the case when the radiuses of the openings (r_1) and (r_2) can not coincide.



d – distance between the centers of the openings of elastic compensator plates;
ρ_1 – distance from the axis of rotation to the center of the smaller opening;
ρ_2 – distance from the axis of rotation to the center of a larger opening;
φ – angle between the lines that beginning from the common origin to the centers of the openings in the plates

Fig.3: Scheme for determination of overlapping of openings of universal compensator of slide direction errors [9]

The distances (a) and (b) between adjacent openings in elastic plates and between the opening and the wall of the compensator according to [8, 9] are:

$$a \geq 0.63(h_1 + h_2); \qquad b \geq 0.63(h_1 + h_2), \tag{3}$$

where h_1 and h_2 – thicknesses of the upper and lower elastic plates, but the difference of thickness between the upper and lower elastic plate does not exceed 20 %.
Also the following limitations were introduced in the design of compensator: radiuses of the elastic plates of the compensator should be less than the length (A) and width (B) of the die-plate (correspondingly: $R < A$ and $R < B$), and the common height of the compensator should not exceed 1/3 of stamping space (H_{pr}):

$$(h_1 + h_2) \leq H_{pr}/3.$$

The supporting area of the compensator elastic plates is determined as: $F_{op.pl} = \pi \cdot R^2$.
The distortion of the slide axis on angle (φ_0) causes to irregularity of deformation of the elastic plates of universal rotary compensator in height. And the maximum of angle deviation should not exceed:

$$\varphi_0 \leq arctg((h_1 + h_2)/2R).$$

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Based on the imposed restrictions the radiuses of elastic plates of the compensator were determined as [9]:

$$R = 0.9 \cdot C_{AB}/2,$$

where is $C_{AB} = \min(A, B)$.

One of the elastic plates of the compensator is movable by rotating about the other. This is enables to change the supporting surface area and leads to a change in the rigidity of the compensator, which is calculated by the formula:

$$C = P/\Delta, \tag{4}$$

where P – compression force;
Δ – value of upsetting of the compensator.

The variation of the overlap coefficient (K_{per}) is carried out by shifting of the upper plate and is defined as:

$$K_{per} = F_{per.otv}/F_{otv}, \tag{5}$$

where $F_{per.otv}$ – overlapping area of the two plates openings;
F_{otv} – area of the plate openings.

According to the results of the work [9] the overlap coefficient in relation to the smaller opening of elastic plate of universal compensator of slide directions errors is calculated by the formula:

$$K_{per} = \frac{1}{\pi}\left(\arcsin A - A\sqrt{1-A^2} + \omega^2 \arcsin\frac{A}{\omega} - \frac{1}{2}\sqrt{\omega^2 - A^2}\right), \tag{6}$$

where are $A = \sin\alpha$, $\omega = r_2/r_1$ (according to the scheme on Fig. 2).
The shape factor is defined as:

$$\Phi = F_{bok}/F_{op}, \tag{7}$$

where $F_{bok} = 4 \cdot \pi \cdot R \cdot (h_1 + h_2)$ – lateral area of the compensator,
$F_{op} = (2 \cdot F_{op.pl} - F_{per.pl})$ – supporting surface of the compensator ($F_{per.pl}$ – overlapping area of the two elastic plates).

In this case, take into account the irregularity of deformation caused by distortion of the slide axis on an angle (φ) at the certain technological compression force (P). The angle of misalignment is defined as:

$$\varphi_0 = arctg(\Delta_{komp}/2 \cdot R),$$

where $\Delta_{komp} = 2.5 \cdot \Delta_{press}$ – maximum misalignment of the compensator, here $\Delta_{press} = P / C$ – maximum misalignment of the press, C – rigidity of the presses with open frame ($C = 300 \ldots 1200$).

The upsetting of compensator during compressing should not exceed 20...25 % of the common height of the elastic plates. Then the total height of the compensator plates can be determined as [9]:

$$(h_1 + h_2) = \Delta_{komp} / 0.2 .$$

For example was considered the using of the rotary elastic compensator during direct pressing (extrusion) of solid profile work piece with circular cross-section through a single channel matrix Fig.4.

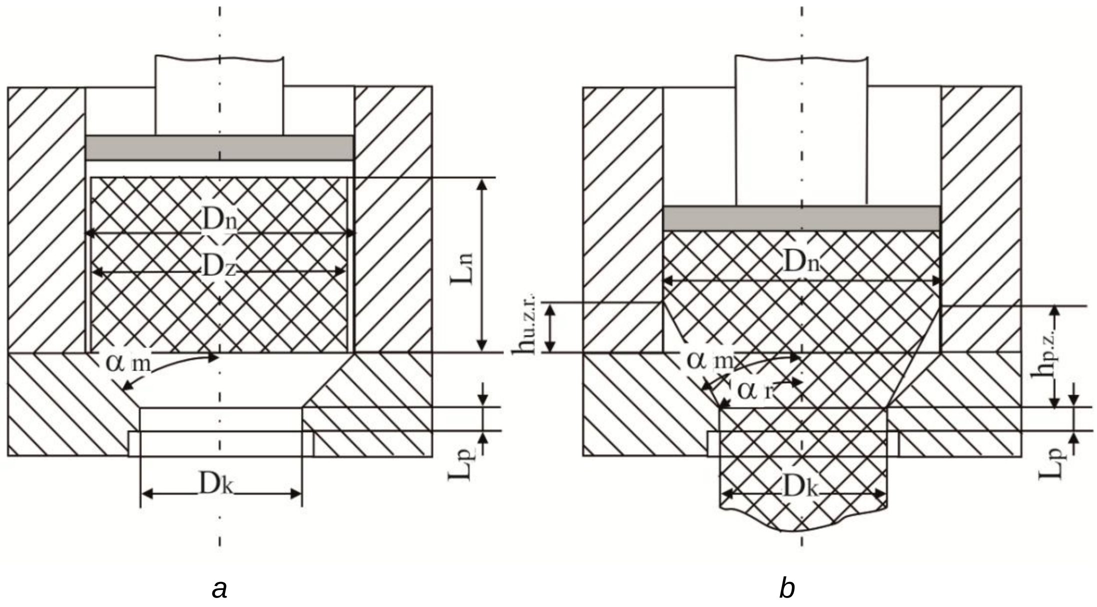


Fig.4: Pressing-extrusion with direct expiration of solid profile of the round billet through a single channel matrix on decompression step (a) and the beginning of the extrusion of the metal (b)

Depending on the actual angle of the channel (α_m), calculated angle (α_r) and the calculated height $h_{u.z.r.}$ are determined by the formulas [10]:

$$\begin{cases} if \quad \alpha_m \le 60^0, \quad then \quad \alpha_r = \alpha_m, \qquad h_{u.z.r.} = 0; \\ if \quad \alpha_m > 60^0, \quad mo \quad \alpha_r = 60^0, \quad h_{u.z.r.} = (D_n - D_k) \cdot (0.58 - ctg\alpha_m)/2, \end{cases} \tag{8}$$

where D_n and D_k – inner and outer diameters of the container.
The volume of the cylindrical work piece is calculated the following [10]:

$$V_z = L_n \cdot \pi D_z^2 / 4; \qquad D_z = 0.97 \cdot D_n , \tag{9}$$

where L_n – the height of the container;

D_z – diameter of the workpiece.
On the pressing-out step a current diameter of the workpiece was determined as:

$$D_h = 1.13 \cdot \sqrt{V_z / L_h}\,, \tag{10}$$

where L_h - the current height of the workpiece.
The final height of the workpiece at this stage was calculated as:

$$L_{OA} = V_z / \left(\pi D_n^2 / 4\right). \tag{11}$$

The pressing force in the first stage was calculated by the formula [10]:

$$P_{OA} = 20 \cdot \sigma_s \cdot \left[1 + 0.09 \cdot (D_h / L_h) \cdot \sqrt{\mu \cdot D_h / L_h}\right] \cdot \pi D_h^2 / 4\,, \tag{12}$$

where σ_s – deformation resistance considering hardening of the work piece material;
μ – friction coefficient ($\mu = 0.3$);
According to the source [10] the pressing-extrusion force on the efflux of the metal stage was determined the following way:

$$P = \left[f_{k.r.} K_{k.r.} \pi D_n \left(L_n - h_{u.z.r.}\right) + \frac{\pi D_n^2}{2} K_{m.b.} i \left(\frac{f_{k.m.}}{2\sin(\alpha_r)} + \frac{1}{\cos^2(\alpha_r / 2)} \right) + \right] 10^{-6} + \,, \tag{13}$$
$$+ \lambda \pi D_k f_{k.p.} K_{m.k.} L_p \cdot 10^{-6},$$

where $f_{k.r.} = f_{k.m.} = f_{k.p.} = f = 1$ – index of external friction;
$K_{k.r.} = K_{m.b.} = K_{m.k.} = K = \sigma_s / \left(3\sqrt{3}\right)$ – plastic shift strain;
$i = D_n^2 / D_k^2$ – deformation index;
$\lambda = 1 / \left[1 - \left(\left(D_n^2 - D_k^2\right) / D_n^2\right)\right]$ – drawing index.

Also the force of pressing was additionally calculated by M.V. Storozhev formula:

$$P = \sigma_s \left[\left(\frac{\mu + 0{,}5}{2\sin(\alpha_m)} + \frac{2}{1 + \cos(\alpha_m)} \right) \ln\left(\frac{D_n^2}{D_k^2} \right) + \frac{2L_h}{D_k} + \frac{4\mu L_p}{D_h} \right] \frac{\pi D_k^2}{4} \cdot 10^{-6}. \tag{14}$$

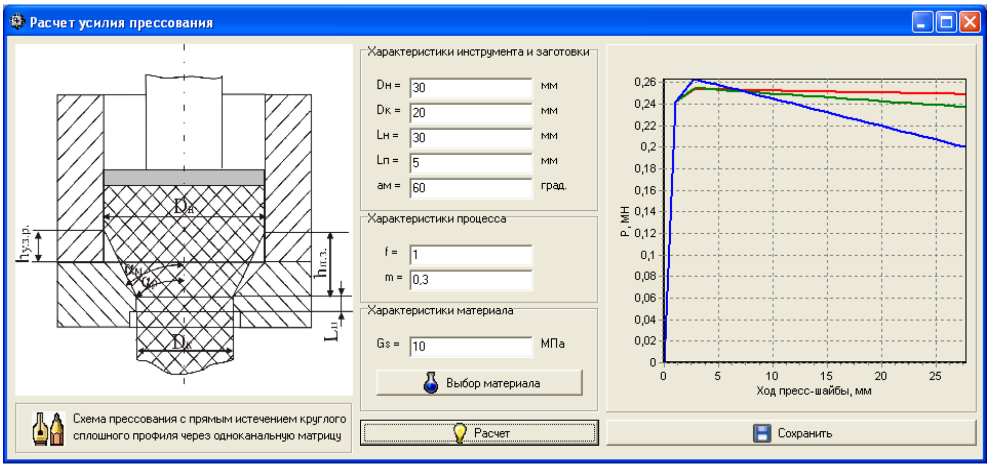
Also conducted the calculation with using the P.P. Edneral and I.G. Konstantinov formula:

$$P = \sigma_s \left(1 + f \cdot tg(\alpha_m)\right) \left[1 + tg(\alpha_m) + f + f \left(\frac{L_p}{D_k} + \frac{L_h}{D_n} \right) \right] \ln\left(\frac{D_n^2}{D_k^2} \right) \frac{\pi D_n^2}{4} \cdot 10^{-5}. \tag{15}$$

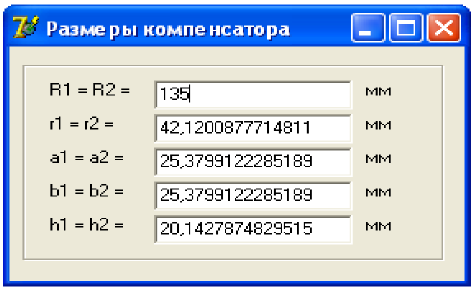
5. PROGRAM AND RESULT OF THE CALCULATION

The special program on the base of the developed mathematical model of pressing (extrusion) processes and work of universal elastic rotary compensator is written. The interface of this program is shown in Fig.5.

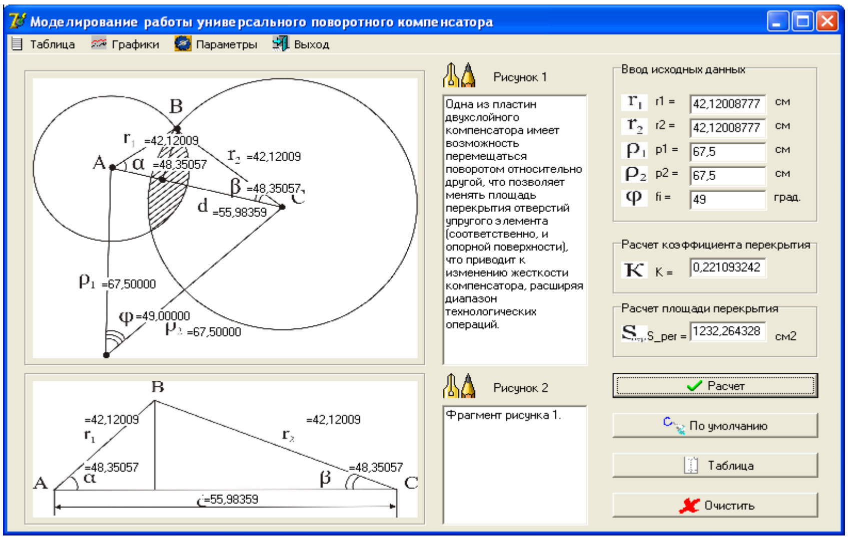
The process of pressing-extrusion is simulated with direct efflux of solid profile of the work piece with circular cross-section through a single-channel matrix (see Fig.5 a), where the inner diameter, outer diameter and the height of the container are respectively $D_n = 30$ mm, $D_k = 20$ mm and $L_n = 30$ mm; girdle height of the matrix $L_p = 5$ mm, channel angle of the matrix $\alpha_m = 60^o$. Index and the coefficient of friction are respectively $f = 1$ and $\mu = 0.3$.



(a)



(b)



(c)

Fig. 5: Interface of the program: calculation of pressing force (a), the determination of the initial dimensions of the universal rotary compensator (b) and modeling of universal rotary compensator (c)

It was assumed that the work piece material has 18.1 MPa value of resistance of deformation considering hardening (this corresponds to the lead as the material for laboratory tests and steel in condition of hot working by pressure). The graphs of dependence of pressing force from the course of press-washer are build (see Fig.5 a).

The initial dimensions the universal elastic rotary compensator were determined (see Fig.5 b): radii of elastic plate's $R_1 = R_2 = 135$ mm, the radius of the plate holes $r_1 = r_2 = 44.35$ mm, distance between adjacent holes in the elastic plates and between the hole and the wall of the compensator $a_1 = a_2 = b_1 = b_2 = 25.38$ mm, height of the plates $h_1 = h_2 = 20.14$ mm. It was established that for optimal rigidity the compensator must be turned the upper elastic plate at an angle $\varphi = 49^0$ (see Fig.5 b).

Thus, was adapted an automated method and developed the program for calculating structural characteristics of rotary elastic error compensators of the "press-and-die" system for the process of extrusion of profiles.

6. CONCLUSION

The using of the elastic compensators is a perspective and low-cost way to reducing of the distortions of slide direction at the “press-and-die” system of press equipment.

There was adapted a mathematical model and developed program for the calculation of the design characteristics of rotary elastic compensators of the "press-an-die" system errors for the process of extrusion or pressing of profiles.

This program allowed executing the following:

a. The calculation of the change of pressing force parameters and compression of the compensator depending on the power modes of pressing-extrusion operation for work piece with solid circular profile;
b. The calculation of the shape factor and the determination of the optimal geometric dimensions of the universal rotary elastic compensator; the calculation of the overlap factor and the optimum angle of rotation of the movable plate of the compensator to achieve the desired rigidity;
c. The program is worked out and tested for a calculation of the power modes and design parameters of the elastic universal rotary compensator.

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