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CONTENT

THE PROBLEMS OF ENSURING OF GEOMETRIC ACCURACY OF METAL PRODUCTS FROM COLD-ROLLED SHEET

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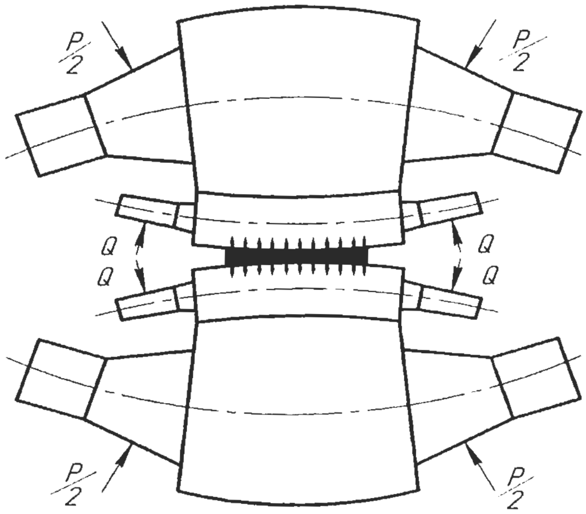
Introduction

High level of consumers' requirements for geometric precision of cold rolled strips stipulates the necessity of application for cold rolling the systems of automatic adjustment of shape and profile (hereinafter AASPS) of finished rolled stock at continuous and reverse mills [1, 2]. At that, improved efficiency of the aforesaid systems can be reached by refinement of mathematic models ensuring their operation, it allowing to increase their reliability and prediction of such indices, like, for instance, transverse polythickness, or the degree of flatness of cold rolled strips. Such indices, together with indices of precision, ensured as a result of shape-transforming operations of sheet stamping (like draw-forming, shaping and others) determine overall indices of precision of metal products, made of cold rolled sheets. To compensate errors of the "stand-tool-blank" system, eliminate unevenness of metal forming at manufacturing of sheet-metal stamping on pressing machines of open type and a consequence improvement of geometric precision of metal products it is advisable to use elastic compensators on the basis of polyurethanes. It would be necessary, at that, to improve the methods of design of elastic compensators of "press-stamp" system with the objective of obtaining high indices of metal products precision. With due regard to all aforesaid this work deals with the analysis of the problem of determination of optimal values of the force of counter-flexure of working rolls of continuous and reverse rolling mills, ensuring improvement of geometric precision of cold rolled strips, as well as the problems of designing appliances for compensating elastic deformation and increasing precision of metal production, obtained from cold rolled sheets, by means of sheet stamping.

Ensuring of geometric accuracy of cold-rolled sheet

For adjustment of the degree of flatness of steel sheets at cold rolling on continuous and reverse rolling mills the systems in which counter-flexure of working rolls are most widely used (see Fig. 1) [1, 2]. To ensure the proper operation of these systems it is advisable to set the target of specifying optimal values of the force of counter-flexure, ensuring constant constancy or minimal differences of overall elastic displacements of the unit of working and back up rolls, irrespective of fluctuations of the rolling force, emerging due to directed or stochastic changes in the original operational parameters of the cold rolling process.



Q – the force of counter-flexure of working rolls; P – the rolling force

Fig. 1. The scheme of counter-flexure of working rolls [2]

The analysis of the methods of evaluation of elastic deformation of rolls units of quarto-stands, described in [3–5] showed that such methods did not correspond fully to actual conditions of cold rolling of thin strips. Particularly, A.I. Tselikov's mathematical model is designated for evaluation elastic deformation of flexure of back up rolls of working stands of sheet rolling mills only, having been obtained at conditions of uniform distribution of inter-rolls linear load and could not take into account the influence of trimming and forces of counter-flexure of working rolls. V.P. Poloukhin's engineering method [4, 5] is devoid of the distinguished flaws, still due to alternations in rolling facilities and technology, implemented lately and wide application of AASPS [1, 2] this method requires some additional correction. The authors of [6, 7] suggested a modified method of evaluation of elastic deformation of rolls units of quarto-mills stands, that takes into account unevenness of distribution of inter-rolls linear load, as well as the rolls profile and allowing to ensuring to raise precision of evaluating optimal values of counter-flexure force of the working rolls. For these reasons development of the method of evaluating optimal values of the force of counter-flexure of working rolls, ensuring an increase in the degree of flatness of cold rolled strips, seems to be a problem of paramount scientific and engineering importance.

Determination of optimal values of the force of counter-flexure of working rolls was numerically formed on the basis of purposeful enumeration of possibilities, according to the following algorithmic diagram:

$$Q_{(t+1)} = Q_t + A_Q sign\left\{\Delta\delta_t - \Delta\delta^*\right\}, \tag{1}$$

where t – a number of the next in turn cycle of iteration solution procedure;

A_Q – a step of increment of the force of counter-flexure, set to be equal to $A_Q = 0.01P_n$;

$sign\left\{\Delta\delta_t - \Delta\delta^*\right\}$ – sign function, corresponding to:

$$sign\left\{\Delta\delta_t - \Delta\delta^*\right\} = \begin{cases} 1 \; if \; \Delta\delta_t > \Delta\delta^*; \\ -1 \; if \; \Delta\delta_t < \Delta\delta^*, \end{cases} \tag{2}$$

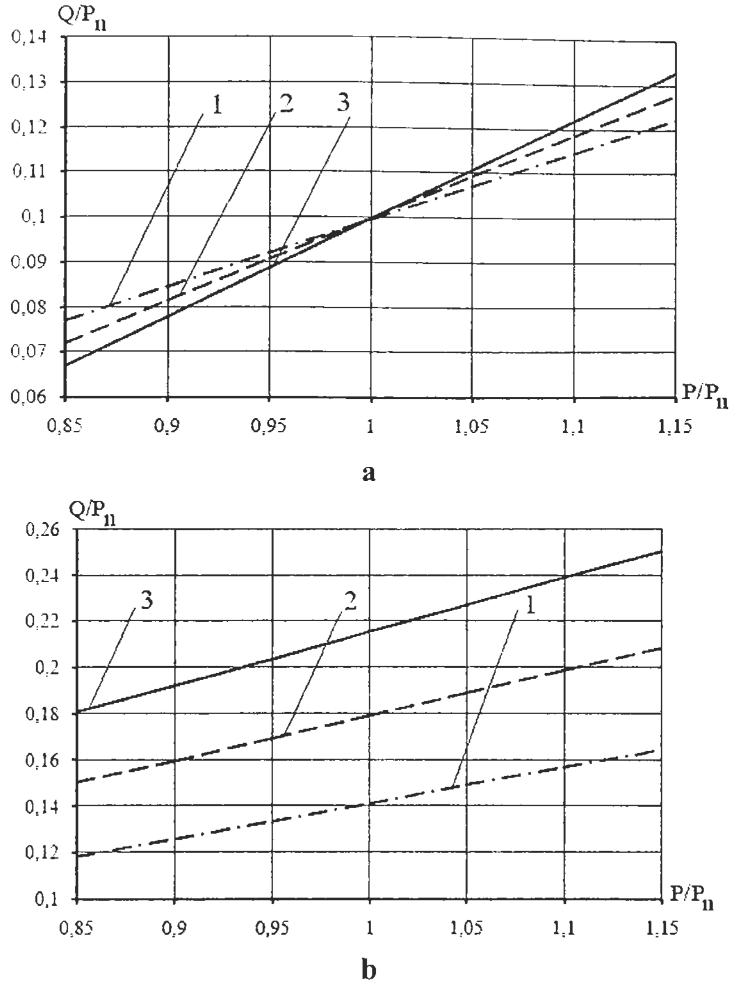
where $\Delta\delta_t$ – rated values of differences of overall elastic displacements in the middle and on the edges of bodies of working and back up rolls, determined within t_{th} cycle of iteration procedure solution procedure, depending upon the corresponding values of the rolling force P and the force of counter flexure Q_t;

$\Delta\delta^*$ – maximum permissible or nominal values of overall elastic displacements, determined at nominal values of the rolling force P_n and the counter flexure force, taken as equal to $Q_n = 0.1P_n$.

The values of differences in overall elastic displacements of the unit of working and back up rolls were calculated in strict accordance with recommendations prescribed by the authors of [6, 7], with due regard to uneven distribution of inter-rolls linear load and trimming of working rolls. The rated determination of the rolling force was carried out in accordance with the modified numerical one-dimensional mathematical model [8], that takes into account the actual character of distributions of mechanical properties of metal along the length of the area of deformation, as well as geometric parameters and coefficient of external contact friction. There, in the first cycle of iteration procedure of solution ($t=1$) the original value of the force of counter flexure $Q|_{t=1}$ was taken as equal to its nominal value, i.e. $Q|_{t=1}=Q_n$, then the value of $\Delta\delta^*$ was determined, after that according to expression (1) an increment of the force of counter flexure $Q_{(t+1)}$ was performed, and then we passed to the following *(t +1)* cycle of iteration procedure of the solution. This procedure was stopped in case of alternation of the sign of the original difference of $\Delta\delta_t - \Delta\delta^*$. The obtained values of the force of counter flexure should not exceed the values, admissible by conditions of necks strength, as well as bearings units of the working rolls.

As an example of numerical realization of algorithmic diagram (1)–(2) at Fig. 2a represents the rated values of distribution of Q/P_n values of the force of counter flexure, ensuring the constancy of the difference in overall elastic displacements in the middle and on the edges of the bodies of working and back up rolls, irrespective of the value of the rolling force, the data were obtained for one of the working stands of 1700 continuous 4-stand cold rolling mill of "Ilyich" Iron and Steel Works PrJSC (Mariupol). There, the range of alternations of this force was considered to be equal to $P=0.85P_n...1.15P_n\left(P/P_n=0.85...1.15\right)$, while nominal values of $\Delta\delta^*$ difference of elastic displacements of $\Delta\delta_t$ were determined in accordance with recommendations, specified by the authors of [6, 7], differentially, in accordance with width of the rolled strip b at nominal values of the rolling force P_n and the force of counter flexure $Q_n=0.1P_n$. Similar rated distributions (see Fig. 2b) were obtained for the condition of minimization of the difference of overall elastic displacements in the middle and on the edges of the bodies of the working and back up rolls, it ensuring obtaining of cold rolled strips with minimal transverse polythickness.

The analysis of the obtained result showed that the required values of the force of counter flexure of the working rolls depended largely upon the width of the rolled strips and they increased linearly with the growth of the rolling force (see Fig. 2b), it allowing approximation of these rated functions with linear functions of the type:



1 – P_n = 16.2 MN, b = 1350 mm; 2 – P_n = 15.0 MN, b = 1250 mm;
3 – P_n = 13.8 MN, b = 1150 mm

Fig. 2. Rated distributions of the corrected values of the force of counter flexure, ensuring constancy of the difference of overall elastic displacements in the middle and on the edges o the bodies of working and back up rolls (a) and minimal transverse polythickness of cold rolled strips (b), irrespective of alternations in the rolling force

$$Q = Q_n + k_{QP}(P - P_n), \quad (3)$$

where k_{QP} is the transmission coefficient, characterizing functional connection between increments of the force of counter flexure Q and the force of rolling P and determined on conditions of ensuring continuity of the difference of overall elastic displacements in the middle and on the edges of the bodies of the working and back up rolls or on condition of ensuring minimal transverse polythickness of cold rolled strips.

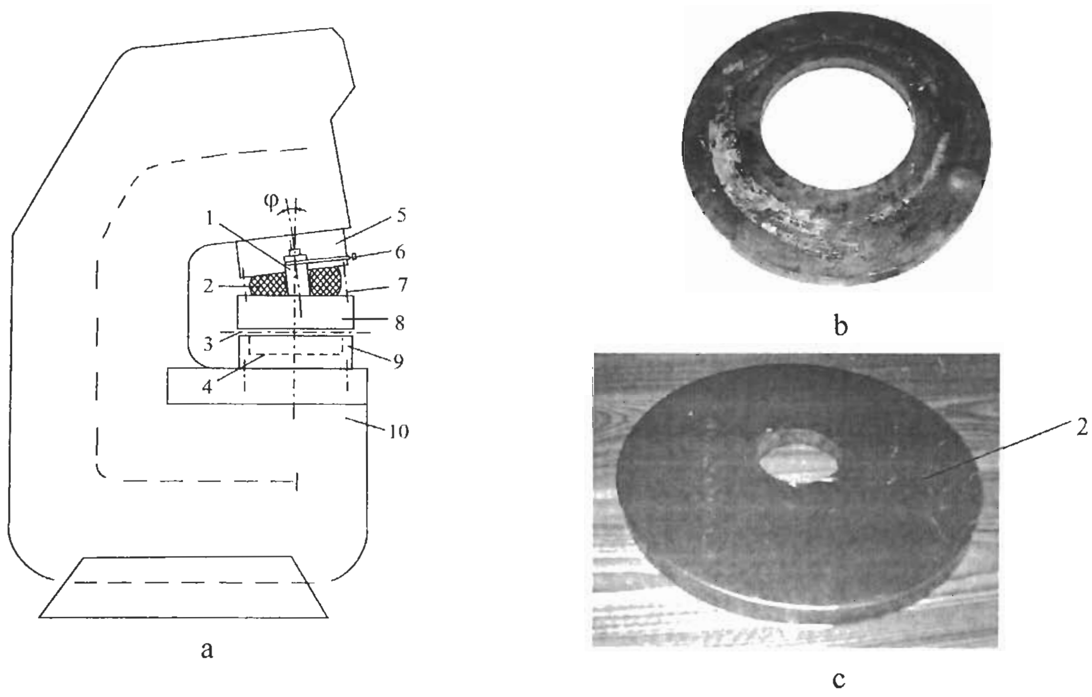
The advantage of linear functions (3) is in the possibility of their application with the objective of determining the values of the force of counter flexure of the working rolls in real time mode, corresponding to minimal polythickness of strips and also required for obtaining minimal longitudinal polythickness of rolled stock, alternations of inter-rolls gap, directly in the process of rolling. Particularly, for Fig. 1a it was obtained that for $k_{QP} = 0.051$ at $b = 1350$ mm, $k_{QP} = 0.088$ at $b = 1250$ mm and $k_{QP} = 0.123$ at $b = 1150$ mm, as for Fig. 1b – $k_{QP} = 0.151$ at $b = 1350$ mm, $k_{QP} = 0.186$ at $b = 1250$ mm and $k_{QP} = 0.221$ at $b = 1150$ mm.

It is worthwhile mentioning that beside AASPS, contemporary rolling mills are equipped with the systems of automatic regulation of strips widths (SARSW) [1, 2]. The main element of these systems is hydraulic screw-down device, ensuring fine regulation of the required value of preliminary rolls gap and its high precision adjustment in the process of rolling. There, in the process of adjustment of the rolls gap the range of alternations in the rolling force is raised, it eventually leading to an increase in alternations in the shape of active generating working rolls and, as a consequence, leading to an increase in transverse polythickness and in probability of defects formation in indices of flatness of the rolled strips. This call for changes in the force of counter flexure of working rolls to be applied without delay. Thus operation of AASPS and SARSW should not be considered separately, in other words, the values calculated by algorithmic scheme (1)–(2) of the value Q must be taken into account for designating of the process mode of SARSW of a particular rolling mill.

Ensuring of geometric accuracy of metal products from cold-rolled sheet

The accuracy of "press-and-die" systems are affected from misalignment of the slide axis to press table surface in the loaded state, from distortion in the gaps between press and slide guides, as well as cumulative errors of the punching unit [9–12]. Mechanical and elastic compensators are widely used to reduce errors in the "press-and-die" system of open-crank presses and ring compensators of elastomeric materials has not been studied enough. This makes it difficult to calculate their design parameters.

There are two basic approaches to eliminate the influence of deformation of the C-frame presses on the alignment of "press-and-die" system. The first approach is the closure of open C-frame press with using the ties [13], which greatly reduces its deformation. A second approach for solving the skew problem is to eliminate the misalignment between the upper and lower basing plates with various additional devices or elements (Fig. 3a). These devices or parts are called “compensators”. A further alternative approach to reducing of errors of the “press-and-die” system can be die-free [14] and impression-free [15, 16] methods of metal-forming.



1 – shank; 2 – compensator; 3 – die line connector; 4 – circuit of a formed product; 5 – slider; 6 – fixing pin; 7 – fasteners; 8 – upper plate; 9 –lower plate; 10 – frame; φ – skew angle of the slider

Fig. 3. The opening of the C-frame press (a), the design of the elastic ring compensator (b), and its upgrading design (c)

The main purpose of installing a skew compensator in forging equipment is to ensure the transfer of the operating force from basic die elements, which have lost alignment as a result of deformation of the frame, to coaxial (parallel) basic die surfaces (see Fig. 3a).

A promising direction to reduce of distortions in the “press-and-die” system is the using of different constructions of compensators, which are made, for example, of different brands of polyurethane. The choice of polyurethane is complicated by wide range of functional properties of the polyurethanes from various manufacturers. Recommendations for optimization of this choice are given in [17]. The method of calculating the optimum design parameters is discussed in [18]. Also, an automated method of calculation for the reconfigurable polyurethane compensators, taking into account the errors of "press-and-die" system, is proposed [19].

Polyurethane elements are widely used for compensation of power and geometric deviations in metal forming technologies [18–21].

The method of fastening the upper plate to the press slider through an elastic compensator (see Fig. 3a) is considered in [18, 19]. The example of the design of the ring elastic compensator is shown on Fig. 3b. However, the work of the ring compensator can be accompanied by cracking of the elastic plate because during the influence of deformation force the material can flow into non-technological openings, forming regions of increased concentration of stresses and unregulated deformations.

To solve this problem, the variant of the upgrading design of the compensator (see Fig. 3c) was developed, according to which the ring elastic compensator is placed between two ground metal plates with a central hole and connected to them using an adhesive bond.

The disadvantage of the above-described method is the lack of optimal bonding conditions (the glue is applied to the entire surface area of the contact surfaces of the protective plates and the elastic element), which should be determined on the basis of the structural and technological features of the ring compensator and the physical and mechanical characteristics of the glue.

The purpose of this work is to develop a mathematical model and study the deformations of the ring elastic compensator for modeling its shaping under the conditions of the "press-and-die" system work with slider skewing, as well as the calculation of tangential stresses in the adhesive bond with protective plates to determine the optimum gluing parameters, which will help to reduce adhesive consumption, tangential stresses, increase the reliability of fastening the compensator and improve the working conditions.

The compression pre-tests were carried with different shape samples from polyurethane brand SKU-PFL-100 (СКУ-ПФЛ-100, 100 Shore hardness). The approximating dependence between the pressures during compression of polyurethane q and the degree of upsetting $\varepsilon = (H_0 - h) / H_0$ (where H_0 is the initial height of the compensator, h is the height of the deformed compensator) for $\varepsilon \leq 0.3$ is received as the result of studying the influence of the shape of the compensator on its elastic properties [18, 19]:

$$q = 52\varepsilon + 1.92 . \qquad (4)$$

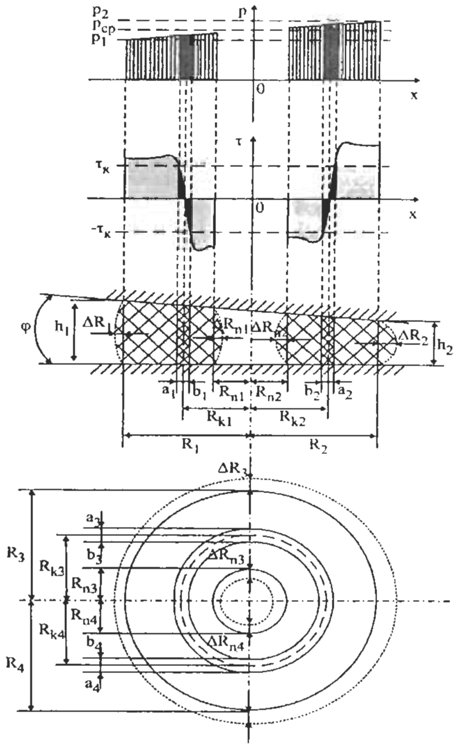


Fig. 4. Uneven deformation of the ring elastic compensator

A ring polyurethane element (see Fig. 4) with an outer radius R, an inner radius R_n and a height H_0 was considered. The support area of the compensator was determinated as $F = \pi (R^2 - R_n^2)/4$. Misalignment of the slide at an angle φ causes the uneven deformation along the height of the compensator (see Fig. 4). Moreover, the maximum of angle deviation should not exceed $\varphi \leq \mathrm{arctg}(H_0/2R)$.

The upsetting of the ring elastic compensator is accompanied by a simultaneous increase in the outer radius R and a decrease in the inner radius R_n. The changes in the radii ΔR and ΔR_n by analogy with the flow of a deformed ring metal preform at four characteristic points were considered. Without considering the shear strain, we had: $\Delta R_3 = \Delta R_4$ and $\Delta R_{n3} = \Delta R_{n4}$. Based on the accepted assumptions, only three characteristic points were considered.

The deformation of the ring elastic compensator is characterized by the presence of a neutral flow line of the material. The boundary of the material flow is the surface defined by the radius R_k. The contact tangential stress τ at points with radius R_k is equal to zero. At the same time, its average value was determined as:

$$\tau = \psi \tau_S\,;\ \tau_S = q/\sqrt{3}\,;\ \psi = \mu + (1/8)\cdot(R - R_n)/H_0 \cdot (1-\mu)\cdot\sqrt{\mu}\,, \qquad (5)$$

where μ is the coefficient of friction.

The radius of the critical surface R_k, as well as the change in the outer radius ΔR and the inner radius ΔR_n at the i-point not taking into account the barrel shape and taking into account the unevenness of the radial deformation along the height were determined by the methods [I.Ya. Tarnovskiy (1963), L.A. Shofman (1961)]. However, preliminary calculations showed a significant difference in the values of R_k

which showed the need to refine the dependence to determine the dimensions of the ring element with respect to barrel shape:

$$R_{k_i} = 63.25 \cdot \sqrt{\frac{R^2 \cdot R_n^2 \cdot \left(R^2 - R_n^2\right)}{8000 \cdot R^2 \cdot R_n^2 \cdot \ln\left(R/R_n\right) + 1599 \cdot h_i^2 \cdot \left(R^2 - R_n^2\right)}}. \tag{6}$$

The change in the outer radius ΔR and the outer radius ΔR_n at the i-point was calculated as:

$$\Delta R_i = 0.375 \cdot R \cdot \left(H_0 - h_i\right)/H_0 \cdot \left(1 - R_{k_i}^2 / R^2\right); \tag{7}$$

$$\Delta R_{n_i} = -0.375 \cdot R_n \cdot \left(H_0 - h_i\right)/H_0 \cdot \left(1 - R_{k_i}^2 / R_n^2\right). \tag{8}$$

The compression pressure p_i at the upsetting of the ring polyurethane compensator is distributed unevenly along its diameter, so the calculation was applied for each deformable part on the four sides of the ring compensator taking into account the unevenness of deformation of the elastic element.

In order to achieve optimum bonding characteristics, the application of glue (diagum FL, diagum P, diaflex) [22] should be carried out only on areas limited by the shear stress zone τ with values not exceeding the critical shear stresses τ_k (see Fig. 4). According to the proposed method, a number of diagrams of tangential stresses are constructed depending on the geometric parameters of the compensator and zones are defined for which $\tau < \tau_k$ (see Fig. 4). The tangential stressτ in the adhesive layer of the polyurethane ring compound with metal protective plates is calculated by the method of Lame, described, for example, in work [23].

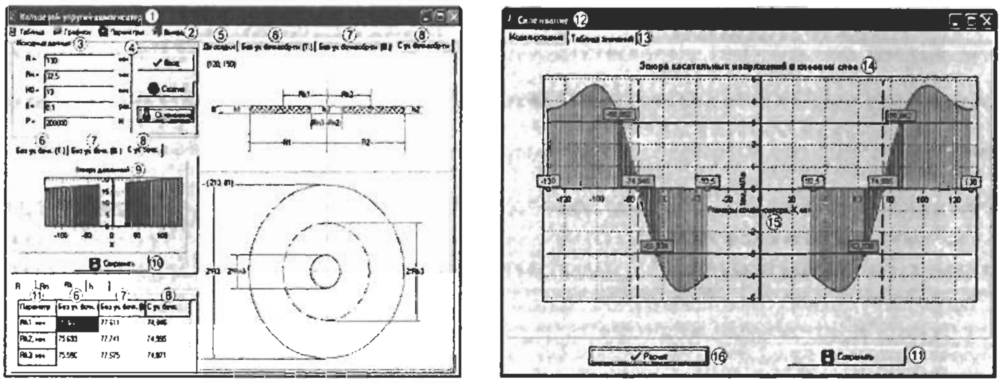
The dependence of the distances from the flow separation line R_k to the outer edge (a) and the inner edge (b) of the compensator from three factors (S/D, S/H_0, ε) was established using the method of experiment planning (2^3):

$$a = 10.283 + 0.034 \cdot S/D + 0.46 \cdot S/H_0 - 1.109 \cdot \varepsilon + 0.135 \cdot S/D \cdot S/H_0 - 0.073 \times$$
$$\times S/H_0 \cdot \varepsilon - 0.034 \cdot S/D \cdot \varepsilon - 0.015 \cdot S/D \cdot S/H_0 \cdot \varepsilon; \; b = 0.929a, \tag{9}$$

where S is the wall thickness of the compensator; D is outer diameter of the compensator.

The special software is written on the base of the developed calculation method (see Fig. 5). This software allows to calculate the dimensions of the ring elastic compensator after the upsetting with and without taking into account the barrel shape and also determine the bonding areas of the ring compensator with protective plates.

The research of the work of the elastic ring compensator with parameters R = 130 mm, R_n = 32.5 mm, H_0 = 13 mm (see Fig. 5a) was carried out. The angle φ = 0.1 rad. and the force P = 0.2 MN (see Fig. 5a). According to the results of simulation (see Fig. 5b), it is advisable to apply the adhesive to the areas of the compensator surface limited by distances a_1 = 14.01 mm and a_2 = 13.94 mm to the outer edge, b_1 = 13.01 mm and b_2 = 12.946 mm to the inner edge of the neutral line, R_{k1} = 74.95 mm and R_{k2} = 74.99 mm; the width of the adhesive layer from the maximum deformation side is 27.02 mm and from the minimal deformation side is 26.9 mm. The consumption of the adhesive compound was reduced by 72 %.

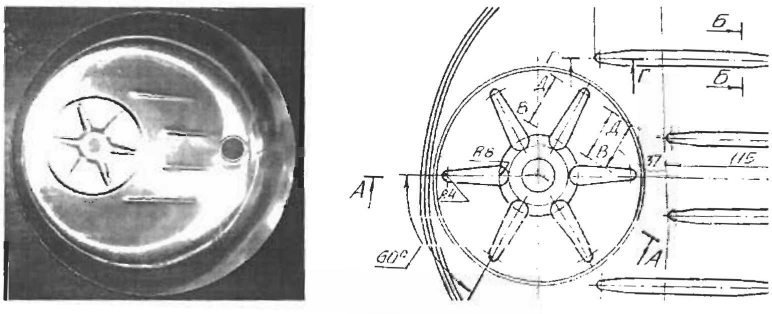


a b

1 – the ring elastic compensator; 2 – table, graphics, parametrs, exit; 3 – initial data; 4 – input, upsetting, gluing; 5 – before upsetting; 6 – without taking into account the barrel shape (by I.Ya. Tarnovsky); 7 – without taking into account the barrel shape (by L.A. Shofman); 8 – with taking into account the barrel shape;
9 – diagram of the compression pressure; 10 – save; 11 – parametrs; 12 – gluing; 13 – simulation, table of values; 14 – diagram of the tangential stresses in the adhesive layer; 15 – compensator sizes; 16 – calculation

Fig. 5. Software interface for modeling of the work of the ring elastic compensator taking into accountthe barrel shape (a) and plotting the shear stresses in the adhesive layer (b)

The upgrated design of the ring elastic compensator was tested during solving of the problem of increasing the accuracy of the dimensions of the stamped part "Bottom" of the washing machine "Donbas" (see Fig. 6) in the conditions of the enterprise "ElektroPobutPrylad" JSC (Mariupol). The compensator in the form of a flat ring made of polyurethane of SKU-PFL-100 grade had an outer diameter of 260 mm, an inner diameter of 65 mm and a height of 13 mm (see Fig. 3b); the calculated thickness of two ground metal plates (see Fig. 3c) was 0.8 mm (X18N9T steel grade). The operation was carried out on a single-crank press of a single action with a C-frame with a nominal force of 1.0 MN (KE 2130A model).



a b

Fig. 6. The bottom of the washing machine "Donbas" (a) and a fragment of its drawing (b)

The depth of molded ribs of activator jack (steel X18N9T) was measured from the front and rear of the press with depth gauge at six points on the extreme edges of the axis. Measurements were carried out using the compensator and without it. The upgraded design of the compensator made it possible to reduce the difference in depth of the ribs of the opening under the activator by 64–73 %.

Conclusions

The method of determination of the force of counter flexure of the working rolls was designed, ensuring improvement of the degree of flatness of cold rolled strips with due regard to the influence of unevenness of distribution of inter-rolls linear load, as well as trimming of the working rolls. On the basis of the analysis of numerical realization of this method an essential influence of the width of the rolled strips upon the optimal value of the force of counter flexure was found out. Also, it was suggested to approximate the link between increments of the force of counter flexure with linear equations, the application of which is to promote improvement in efficiency of AASPS operation of cold sheet rolling mills. For the cold rolling mill of "Ilyich" Iron and Steel Works of Mariupol the values of transmission coefficients of the aforementioned equations, depending on widths of rolled strips were determined.

Based on the performed analysis of the ring elastic compensator work under conditions of the "press-and-die" system work with slider skewing, its design, methodology and software for calculating its sizes after the upsetting were developed. This made it possible to evaluate the slider skew on the shaping of the elastic element, to determine the dimensions of the areas for bonding the elastic compensator to the protective plates, and to minimize shearing stresses in the adhesive layer by 72% and to improve the quality of the finished metal products from the sheet by 64–73%.

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