Analysis of the reliability of the plates of complex shapes for the cams of automatic half-hose machines by the criterion of fatigue strength

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Abstract. The purpose of the work is to develop the algorithm for calculation of the probability of failure-free operation of flexible plates of complex shapes for the cams, using the criterion of fatigue strength, to make decisions on minimization of their sizes with the subsequent evaluation of the implemented measures. The studies are performed on the example of a plate for the cam, the configuration of which provides both beam and double console deformations at the same time.
The comprehensive approach is presented, which includes dynamic analysis of the needle impact on the cam surface, computer simulation of geometric parameters of the plate by the finite element method (FEM) according to the condition of strength, and classical principles of mathematical statistics and Theory of Probability. The finite-element model with a volumetric 8-node element is used, which allows, with ease of calculations, to obtain the results with a high accuracy when determining the stress-deformed state of the study object. The computer simulation is performed using Code_Aster software with free access. Also, in the calculation of reliability, the random nature of the multifactorial influence on the value of the plate fatigue limit and the loading variations of the cam are considered. When determining the equivalent forces, taking into account the cyclical operation of the machine, the forces that affect the cams during the production of one product are taken as a loading. The numbers of cycles corresponding to these loadings are calculated by the formulas, which are made as a result of the analysis of trajectories of needles motion concerning the cams when knitting different sections of the product. When calculating equivalent stresses, only stresses that influence the accumulation of fatigue failures are used, that is, the values greater than the half of the detail fatigue limit.
The proposed computational approach can be used to generate the cams with flexible work plates of different configurations, including under the equal stress condition.

1. Introduction

In the automatic half-hose machines, technological trajectories of the needles are set by the movements of their heels after the cams, accompanied by impact loadings and accumulation of fatigue damages. It is obvious that the intensification of the speed modes of the automatic half-hose machines and the enhancement of technological capabilities lead to their further growth. A promising solution that can increase the reliability of the automatic half-hose machines is to reduce the impact force in a "needle-cam" pair, primarily by increasing the flexibility of the cam.

2. Literature review

The bibliography that concerns this way of improvement is presented in [1, 2, 3], where the design features of the cams with flexible work plates (FWPs) are mainly considered. Some works [4, 5] are devoted to the validation of FWPs by the criterion of static strength. However, given that FWPs are classified as the elements that are cyclically loaded and limited in size, we consider it appropriate to analyze them for fatigue strength in probabilistic figuration.

3. Research methodology

The purpose of the work is to propose the algorithm for calculation of the probability of failure-free operation of flexible plates of complex shapes for the cams, using the criterion of fatigue strength, to make decisions on minimization of their sizes with the subsequent evaluation of the implemented measures. The studies are performed on the example of a plate for the cam, the configuration of which provides both beam and double console deformations at the same time and the dimensions of which are obtained in [4] as the result of the computer simulation.

Classically, according to the criterion of fatigue strength, the probability of failure-free operation of the part is determined depending on the quantile [6]:

$$u_p = -\frac{\bar{n}-1}{\sqrt{\bar{n}^2 \nu_{-1D}^2 + \nu_a^2}}, \tag{1}$$

where $\bar{n} = \bar{\sigma}_{-1D} / \bar{\sigma}_a$ – the coefficient of safety on the average values of the calculated stress $\bar{\sigma}_a$ and the detail fatigue limit $\bar{\sigma}_{-1D}$; accordingly, ν_{-1D} and ν_a – the coefficients of their variation.

In the probabilistic aspect, in the calculations for the plates of complex shapes for the cams, σ_{-1D} and σ_a are considered as random values, for which the mean values $\bar{\sigma}_{-1D}$ and $\bar{\sigma}_a$ have been determined earlier.

The transition from the material $\bar{\sigma}_{-1}$ to the detail $\bar{\sigma}_{-1D}$ according to the medial values of fatigue is an important step in the calculation of fatigue. The traditional provisions, given in [7], are used:

$$\bar{\sigma}_{-1D} = \frac{\bar{\sigma}_{-1}}{K}, \tag{2}$$

where K – the total coefficient of multifactorial impact, which is determined by the formula:

$$K = \left(\frac{k_\sigma}{k_{d\sigma}} + \frac{1}{k_{F\sigma}} - 1 \right) \frac{1}{k_V \cdot k_A}. \tag{3}$$

The coefficients k_σ, $k_{d\sigma}$, $k_{F\sigma}$, k_A and k_v, provided in (3), respectively characterize the influence of concentration of normal stresses, absolute dimensions (scale factor), quality of processed surface, the anisotropy of material and surface strengthening on $\bar{\sigma}_{-1D}$. The values of coefficients are determined according to the tables and graphs, which, in turn, are made according to the results of the experiments, or calculated in the absence thereof.

The dependence [8] is used in the calculation of k_σ:

$$k_\sigma = 1 + q_\sigma (\alpha_\sigma - 1), \tag{4}$$

where q_σ – the coefficient of metal sensitivity to the concentration of normal stresses; α_σ – theoretical coefficient of concentration.

Since for the structural bearing steel SHKH 15, from which the FWP of the cam is made, the limit strength is $\sigma_B = 2410$ MPa [9], $q_\sigma = 0.025$ is chosen, which is reasonable at $\sigma_B > 1300$ MPa.

According to (4), it is possible to take $k_\sigma = 1$. Considering the dimensions of the dangerous section of the FWP console, it is also obvious that there is no influence of the scale factor, i.e. $k_{d\sigma}=1$.

In terms of quantity, the influence of the quality of the detail surface on its fatigue limit is determined by the formulas:

$$k_{F_\sigma} = 1 - 0.22 \lg R_Z \left(\lg \frac{\sigma_B}{20} - 1 \right) \text{ at } R_Z > 1 \text{ mcm; } (a)$$

$$k_{F_\sigma} = 1 \text{ at } R_Z \le 1 \text{ mcm } (b) \tag{5}$$

where R_Z – the roughness of the surface of the plate.

For the FWP of the cam, the average height of fine irregularities is $R_Z \approx 3.2$ mcm [10]. Taking (5, *a*), we have:

$$k_{F_\sigma} = 1 - 0,22 \lg 3.2 \left(\lg \frac{2410}{20} - 1 \right) = 0.88 .$$

The value of the anisotropy coefficient is determined according to the table [7] at $\sigma_B > 1200$ MPa: $k_A = 0.8$.

Considering the instructions, provided in [7, 8], the following values of the coefficient k_v are recommended: 1.0 – without surface strain hardening; 1.15 – with surface strain hardening. In accordance with the technology of FWPs manufacturing, the value $k_v = 1.15$ should be selected.

Applying the calculated and accepted values of coefficients to the (3), we have $K = 1.24$. As a rule, in the calculations, we have $K = 1.5…3$. The deviation occurs, first of all, because of the small dimensions of the dangerous section of the plate compared to most real parts. The reduction K is also influenced by the technology of the FWP production. Then, according to (2), we have $\overline{\sigma}_{-1D} = 770/1.24 = 621$ MPa.

To determine the statistical characteristics of FWP loading, the range of its loadings (forces F_i and corresponding number of cycles of loading z_i) resulted from the needles actions in different speed modes is presented separately for the needle-lifting cam and the stitch cam, which brings the needles down. The numbers of modes are listed in Table 1.

The forces are calculated by the formula from [1], the reliability of which is verified experimentally:

$$F_{\max} = V_x \cdot tg\alpha \sqrt{\frac{m_{red} \cdot C_{red}}{1+K_c}} + \frac{1}{1+K_c}(F_r + 2h \cdot V_x \cdot tg\alpha \cdot m_{red}) , \tag{6}$$

where V_x – the horizontal component of the needle heel speed, which is equal to the circular speed of the points on the needle cylinder surface; α – the angle of inclination of the working surface of the cam to the horizontal line; m_{red}, C_{red} – the reduced mass and rigidity in the “needle-surface cam” pair; K_C – coefficient that takes into account the deformation of the bend of the needle rod at the moment of impact; F_r – the resistance force of the needle movement in the race, which is created specifically to eliminate the willful lowering of the needles; h – the damping coefficient.

In the calculations, the reduced mass m_{red} is considered equivalent to the mass of the needle and C_{red} is calculated as in case of the series connection of rigidity of the needle during the interaction with a non-deformable cam (which is determined in the manner similar to h, considering the frequency characteristics of the oscillograph chart [1] and the rigidity of FWP of the cam, calculated in [4]).

The results of the calculations of needle loadings on FWPs of the needle-lifting and stitch cams by the formula (6) at different speed modes are presented in Table 1. Analytical determination of stresses in the FWP of the cam as a statically indeterminate spatial construction of complex shapes with two rigid structures is voluminous and uninformative in the end result. Therefore, the computer simulation is used, which also makes it possible to eliminate most of the assumptions made in the analytical approach [11]. A finite-element model of the FWP is developed (Figure 1), where a volumetric 8-node finite element is used, which is simple for calculations and allows obtaining results with a high accuracy when determining the stress-deformed state of the study object. The simulation is performed using Code_Aster software with free access [12,13,14]. The maximum stresses at the FWPs points for all six loading modes are presented in Table 1, and sample screenshots – in Figure 2.

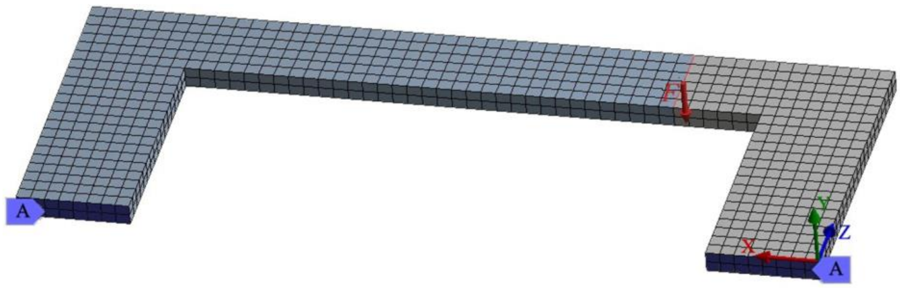

Figure 1. The model of the flexible plate for the cam according to the finite element method: F – force; A – areas of the hardcover of the plate.

a)

b)

Figure 2. The distribution of the fields of stresses σ according to Mises in the flexible plate for the cam at the following loadings: *a)* mode 1 and *b)* mode 6, Table 1.

Since the fatigue strength calculations take into account the stresses at $\sigma_i > 0.5\sigma_{-1D}$ (stresses $\sigma_i < 0.5\sigma_{-1D} = 0.5 \cdot 621 = 310.5$ MPa do not affect the accumulation of fatigue failure [8]), then in the subsequent actions the loadings in the modes 1 and 4 in Table 1 are neglected. It should be noted that since $\sigma_a < \sigma_{-1D}$, the FWP for the cam has the loading on the gigacycle area of the curve of fatigue.

Considering the cyclicity of the automatic half-hose machines work, it is advisable to take forces acting on the cams during the production of one product (sock) as the loading. The number of cycles of loading z_i is set for typical socks with jacquard weaving on the ankle and foot areas, with classic heel and toe, which are knitted during the direct and reverse rotations of the needle cylinder and with additional technological rows. The selection of needles by the cam with an incomplete closing of the third system during the knitting of spandex of the product, by the ornamented gates of three systems during the formation of the pattern on the ankle areas (the average number of included gates is 2,23 gates [15]), by the malfunction of gates on the foot area in the second and third systems, the devices for turning on and off the needles when knitting heel and toe tabs, as well as the number of needles and knitting systems, are considered. The calculation of z_i takes into account the results of the analysis of trajectories of the movement of the needles in relation to the cams when knitting different sections of the products. The formulas for determination of z_i in accordance with the modes of loading and the corresponding calculated numbers are presented in Table 2.

Table 1. Summary information about loading of the cams.

Loading modes			F_i,	σ_i,	z_i,	$p_i = \frac{z_i}{\Sigma z_i}$
i	Type of cam	V_x, m/s (areas of the product)	N	MPa	cycles	
1	Needle-lifting $\alpha = 38.0^o$	V_x = 0.6 m/s (heel and toe)	10.54	254.7	24824	-
2		V_x =1.1 m/s (spandex)	13.14	317.53	14112	0.148
3		V_x =1.3 m/s (ankle, foot and technological rows)	14.18	342.66	81480	0.852
4	Stitch $\alpha = 47.5^o$	V_x = 0.6 m/s (heel and toe)	11.74	283.7	15836	-
5		V_x =1.1 m/s (spandex)	15.42	372.63	9366	0.274
6		V_x =1.3 m/s (ankle, foot and technological rows)	16.87	407.67	24790	0.726

The real loading with an obvious regularity of alternation of different levels during the cycle of knitting of one typical product is replaced by that one, which is equivalent in the degree of accumulation of the fatigue failure, by the formula [16]:

$$\overline{F} = \sum_i p_i \cdot F_i , \tag{7}$$

where $p_i = z_i \big/ \sum z_i$ – the relative occurrence of the loading F_i (the calculated values are given in Table 1).

Table 2. Formulas for calculating the numbers of cycles loading of the cams in accordance with the technological modes.

Cam	Loading mode		Number of cycles of loading z_i	
	i	V_x, m/s (areas of the product)	Formula for calculation	Number
Needle-lifting	1	V_x = 0.6 m/s (heel and toe)	$z_1 = (n_{hl} + n_{toe})(2k^{0-1305} + 4k^{0-1306}) / m$	24824
	2	V_x =1.1 m/s (spandex)	$z_2 = 2n_{sp}(k_2^{0-1305} + k_2^{0-1306} + k_2^{0-1308})$	14112
	3	V_x =1.3 m/s (ankle, foot and technological rows)	$z_3 = \frac{3}{r}(n_{ank} + n_{ft} + 2n_{tr})(\kappa_3^{0-1305} + \kappa_3^{0-1306} + \kappa_3^{0-1308})$	81480
Stitch	4	V_x = 0.6 m/s (heel and toe)	$z_4 = 2(n_{hl} + n_{toe})(\kappa_4^{0-1305} + \kappa_4^{0-1306}) / m$	15836
	5	V_x =1.1 m/s (spandex)	$z_5 = 1.33n_{sp}(k_5^{0-1305} + k_5^{0-1306} + k_5^{0-1308})$	9366
	6	V_x =1.3 m/s (ankle, foot and technological rows)	$z_6 = (0.91n_{ank} + 1.12n_{ft} + 2n_{tr})(\kappa_6^{0-1305} + \kappa_6^{0-1306)} + +(0.91n_{ank} + 0.68n_{ft} + 2n_{tr})\kappa_6^{0-1308} / r$	24790

Note. In Table 2, the following notations are used:
1. $m = 2$ – the number of simultaneously selected needles for one reverse movement of the cylinder; r – the number of cams, which make the looped rows at one rotation of the cylinder;
2. n_{hl}, n_{toe}, n_{sp}, n_{ank}, n_{ft}, n_{tr} – the number of looped rows on the following areas of the product: heel and toe, spandex, calf, foot, and technological rows;
3. κ_1^{0-1305}, κ_1^{0-1306}, κ_4^{0-1305}, κ_4^{0-1306} – the number of included needles of the positions 0-1305 and 0-1306 when knitting the sock heel and toe in the modes 1 and 4; κ_2^{0-1305}, κ_2^{0-1306}, κ_2^{0-1308}, κ_5^{0-1305}, κ_5^{0-1306}, κ_5^{0-1308} – the number of included needles of the positions 0-1305, 0-1306 and 0-1308 when knitting the spandex in the modes 2 and 5; κ_3^{0-1305}, κ_3^{0-1306}, κ_3^{0-1308}, κ_6^{0-1305}, κ_6^{0-1306}, κ_6^{0-1308} – the number of needles of the positions 0-1305, 0-1306 and 0-1308 when knitting the ankle, foot and technological rows in the modes 3 and 6 accordingly.

Applying the values, presented in Table 1, to the (7), we have $\overline{F}_{lif}$ =14.03 N and $\overline{F}_{st}$ =16.47 N for the needle-lifting cam and the stitch cam respectively. Using the computer simulation, the corresponding equivalent stresses $\overline{\sigma}_{lif}$ = 339.04 MPa (Figure 3, *a*) and $\overline{\sigma}_{st}$ = 398.0 MPa (Figure 3, *b*) are determined according to the median values of equivalent stresses.

The analysis of fatigue limits of materials and details indicates their variations due to the structural heterogeneity of the material of one melting, changes in mechanical characteristics in different melts, variations of dimensions, random changes in the modes of mechanical, thermal or chemical treatment, etc., which are taken into account in the general coefficient of variation. According to the rule of quadratic addition of separate coefficients of variation, the coefficient of variation of the detail fatigue limit is determined by the formula [6]:

$$\nu_{\sigma_{-1D}} = \sqrt{\nu_{\sigma_{\max}}^2 + \nu_{\bar{\sigma}_{-1}}^2 + \nu_{\alpha_\sigma}^2}\ , \tag{8}$$

where $\nu_{\sigma_{\max}}$, $\nu_{\bar{\sigma}_{-1}}$ and ν_{α_σ} – separate coefficients of variation of the maximum stresses in the zone of concentration of average (in one melting) fatigue limits of smooth laboratory specimens, theoretical coefficient of concentration of stresses α_σ.

a)

b)

Figure 3. The distribution of the fields of stresses σ according to Mises in the flexible plate for cam at the following loadings: *a*) $\overline{F}_{lif}$ =14.03 N and *b*) $\overline{F}_{st}$ =16.47 N.

According to [6], in the first approximation $\nu_{\sigma_{\max}} \leq 0.1$ we chose $\nu_{\sigma_{\max}}$ = 0.1. Considering the linear dependence between the fatigue limits and the limit strengths of the materials [8], it is acceptable to assume that $\nu_{\overline{\sigma}_{-1}} = \nu_{\sigma_B}$, where ν_{σ_B} – the coefficient of variation of the metal strength limit considering the number of all melts. Since for the alloy steel the indicator ν_{σ_B} is within the range 0.1…0.16, we used $\nu_{\overline{\sigma}_{-1}}$ = 0.13 in the calculations.

Considering the small sizes of the dangerous section of the plate, the effect of ν_{α_σ} on $\nu_{\sigma_{-1D}}$ is neglected. Therefore, according to (8), we finally have $\nu_{\sigma_{-1D}}$ = 0.164.

4. Results

According to (1), standard deviations of the operating loads are $S_{lif} = 1.375$ N for the needle-lifting cam and $S_{st} = 1.416$ N for the stitch cam. Accordingly, the coefficients of variation are $\nu_{alif} = S_{lif} / \overline{F}_{lif} = 0.098$ and $\nu_{ast} = S_{st} / \overline{F}_{st} = 0.086$. The calculated values of coefficients of variation fall under the assertion provided in [6] that $\nu_{\overline{\sigma}_{-1}} > \nu_a$.

Therefore, at (1) for the needle-lifting cam at the assurance coefficient at the average stresses $n_{lif} = \overline{\sigma}_{-1D} / \overline{\sigma}_{lif} = 621/339.04 = 1.832$ and at $\nu_{\sigma_{-1D}}$ = 0.164 and ν_a^{lif} =0.098, we have $u_{p_{lif}}$ = -4.95, which allows determining the probability of failure-free operation $p(t) \to 1$ using the tables [6].

Accordingly, for the stitch cam $n_{st} = \overline{\sigma}_{-1D} / \overline{\sigma}_{st} = 621/398.0 = 1.560$, $\nu_{\sigma_{-1D}}$ = 0.164, ν_a^{st} =0.086, $u_{p_{st}} = -2.077$ and $p(t)$ = 0.981, which is sufficient for the technical systems.

If we set a goal that both needle-lifting cam and stitch cam have the same probability of failure-free operation, for example, $p(t)$ = 0.981, then, concerning the needle-lifting cam, we should solve the inverse task on determination of geometric parameters of its FWP, which would provide the corresponding equivalent stress $\overline{\sigma}_{lif}$ = 398.0 MPa. The adjustment is made by the width of the console beam B, where we have the dangerous section. The screenshot at such a stress at B = 3.17 mm is presented in Figure 4.

Figure 4. The distribution of the fields of stresses σ according to Mises in the flexible plate for cam at the loading $\overline{F}_{lif}$ =14.03 N and the width of the console beam of the plate B = 3.17 mm.

Using the results of similar calculations, the graphical dependence of $p(t)$ on B (Figure 5) is constructed, which is convenient to use when designing a plate for the needle-lifting cam at any given probability of failure-free operation.

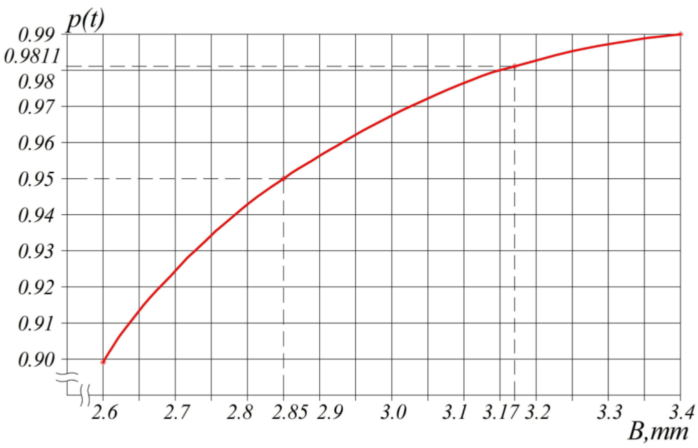

Figure 5. The dependence of probability of failure-free operation $p(t)$ on the width of the console beam of the plate *B*.

5. Conclusions

The comprehensive approach to estimation of the probability of failure-free operation of plate of complex shape for the cam of automatic half-hose machine, using the criterion of fatigue strength, is proposed, which includes dynamic analysis, computer simulation by the finite element method (FEM), and principles of mathematical statistics and Theory of Probability. On the example of the plate of the cam as the detail, the sizes of which are limited, the feasibility of using the strength analysis in probabilistic figuration is shown, which in comparison with the traditional calculations (that consider the normalized coefficients of safety) provides a given level of reliability together with minimization of sizes. To describe the stress state of the plate as a statically indeterminate spatial construction of complex shape with two rigid structures, the finite element method is used in Code_Aster software. The proposed computational approach can be used to generate the cams with flexible work plates of different configurations, including under the equal stress condition.

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