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TAKING IMPACT OF OSCILLATION AMPLITUDE OF BEARING FRAME SECTIONS OF BOOM SPRAYERS INTO ACCOUNT ON ITS RESOURCE

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Summary. *The proposed calculation model for determining the resource of bearing frameworks of sections of barbed field sprayers. With the help of this model, the calculation of the rod's resource for the maximum amplitudes of the cyclic bending of its weakest elements was performed. It is shown that its resource decreases more than 2 times compared to the normative one.*

Key words*: sprayer, frame of high width boom, work resource, cyclic loading, oscillation amplitude, surface crack.*

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Problem statement. Chemical protection of plants is one of the effective and productive methods. However, it can also be very dangerous if the technology of application of the working drug is disrupted. Among the requirements for quality spraying is the timeliness of the spraying in accordance with the biological development of the plant, the pest, the progression of the disease etc. That is why chemical protection machines are given a short time frame for carrying out this technological operation. This means that, all other things being equal, a shift in the time of chemical protection may significantly reduce its effect or be unnecessary at all. Moving on to the chemical protection machines means they must be technically serviceable and ready to work.

If analysis of failures of this type of machines is carried out, a significant part of them are caused by failures related to loss of working capacity of metal structures, among which on the first place – damage of frames of sections of wide-grip booms. Complex operating conditions such as: field irregularities cause additional fluctuations of boom elements, causing significant dynamic overloads, as well as operation of such metal structures takes place in an aggressive environment, accelerates corrosive processes and contributes to reduction of service life of such elements.

As can be seen from Figure 1 that the damage of the elements of the structure of the sprayer can have different character – fatigue failure (Fig. 1, a, c) and due to improper operation, which caused collision with the obstacle, which caused critical deformation of the frame elements (Fig. 1, b).

a) *b)* *c)*

Figure 1. Examples of typicall frame elements damage of high boom width sprayers

Analysis of recent research and publications. Analyzing scientific publications concerning high boom width sprayers, they are predominantly aimed at mathematical or simulation modeling of oscillatory processes in them. In the end, the authors establish the stress-strain state of the booms or elements of the sprayer frame [1, 2]. It is clear that there is a large and integral part of research of chemical protection machines, but unfortunately it is not enough. The basis of the design of machines for chemical protection of plants is to determine the resource (residual life) of the elements of their metal structures, although information on methods of investigation of the damage of elements of booms wide sweep sprayers is almost absent. Professor T. I. Rybak made a significant contribution to the development of the theory for determining the residual life of the sprayer frame structures [3].

The goal of the work. Construct calculation models to determine the resource (residual life) of the elements of the bearing frames of the sections of the boom sprayers.

Presenting main material. As an experimental sample, a wide spreader boom of the OPSH-15 scanner (Fig. 2) was used, for which the initial data on checking the advanced theory of determining the residual life of the boom frames are known.

Figure 2. General view of sprayer «OPSH-15»

This sprayer in the classical view (Fig. 2) has a five-section boom with a grip width of 16.2 *m*. Of course, with this width of the boom grip, it is possible by increasing of forward speed, but this in turn leads to an increase in the dynamic forces acting on the structural elements.

Experimental studies have been established [3] (Fig. 3) that during the working process there are cyclic oscillations of such boom with frequency f within one hertz $f \approx 1$ Hz. As a result, some of its elements experience tensile, compressive, and bending cyclic loads. The task is to determine the resource of the boom (number of cycles of oscillation $N = N_*$ or real operating time $t = t_*$), upon which it will cease to perform its functional properties.

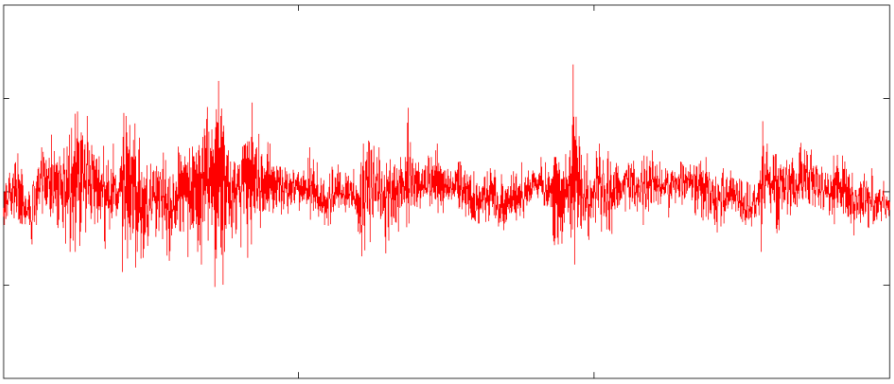


Figure 3. Fragment of oscillogram of experimental research of dynamic loading on boom metal construction

As is common in engineering practice [4], the resource of metal structure is determined by the lifetime of the weakest link, that is, one of its most loaded elements. According to the modern theory of fatigue (mechanics of delayed damage of structural elements at variable loads [4]), the resource of the structural element (number of oscillations N_* or time t_*) at variable loads is determined as follows:

$$N_* = N_{init} + N_{subcr}\,,\ t_* = t_{init} + t_{subcr}\,. \tag{1}$$

Here N_{init}, t_{init} are periods of origin of fatigue crack; N_{subcr}, t_{subcr} – periods of its critical growth. The calculation of the crack initiation period for such elements will be carried out on the basis of relations $\sigma \sim \lg N$ for the area of limited durability of the Weller diagram. On the basis of the results of work [4] it can be concluded that this dependence in semi-logarithmic coordinates will be approximately linear and analytically it can be represented as follows

$$\sigma\sigma_0^{-1} = \lg(N_0 N^{-1})\,,\ \sigma\sigma_0^{-1} = \lg(t_0 t^{-1})\,. \tag{2}$$

Here σ_0, N_0, t_0 are characteristics of fatigue damage of materials within the area of limited durability ($10^4 \le N \le 10^7$), determined on the basis of experimental studies. Then the period of fatigue crack occurrence N_{init} in the structural element, in which the external load causes the amplitude of stresses varying over time σ, will be determined from the results [4] as follows:

$$N_{init} = N_0 10^{-\sigma\sigma_0^{-1}}, t_{init} = t_0 10^{-\sigma\sigma_0^{-1}}, t_0 = 2{,}8 \cdot 10^{-4} N_0\ h. \tag{3}$$

Based on the results of work [5] determination of the period $N = N_{subcr}$ of precritical growth of corrosion-fatigue crack length l in the plate we reduce to the following equation

$$dl / dN = \alpha_0 (1-R)^4 (K_{\mathrm{Im}ax}^4 - K_{th}^4)[(1-\xi^2)(K_{fC}^2 - K_{\mathrm{Im}ax}^2)]^{-1}. \tag{4}$$

Here $\xi = p\sigma_t^{-1}$, p is the regular part of stresses at the tip of the crack; σ_t – sequence of stress in the zone of the front-end; $R = K_{\mathrm{Im}in} / K_{\mathrm{Im}ax}$; K_{fC} – Critical value of voltage intensity factor $K_{\mathrm{Im}ax}$ by cyclic load; The value K_{th} that would be the lower threshold value $K_{\mathrm{Im}ax}$ for the short crack, based on [4] is determined through K_{th}^* for the large as follows

$$K_{th} = K_{th}^* \sqrt{(1-\xi^2)}\,. \tag{5}$$

As follows from [5] the value K_{th} is not constant and depends on the load level ξ, that is, at all levels of crack propagation it will be different.

Equation (4) with corresponding start and end conditions

$$N = 0,\ l(0) = l_0;\ N = N_{subcr}, l(N_{subcr}) = l_* \tag{6}$$

and makes a design model for determining the period $N = N_{subcr}$ of subcritical growth of fatigue straight crack regardless of its size (excluding physically small cracks). At that, critical crack length $l = l_*$ is determined by KRT criterion [5]

$$\delta_t(l_*) = \delta_{fC}\,. \tag{7}$$

Here $\delta_t(l_*)$ is the maximum per cycle of crack opening at its top at average voltage σ_t , and δ_{fc} its critical value.

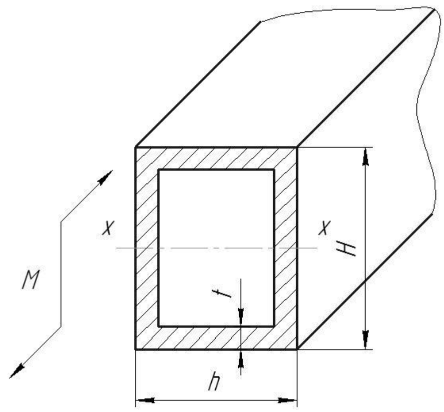


Figure 4. Scheme of rectangular pipe loading

Evaluation of life (residual life) of the most loaded bending element of the closed profile boom. Rectangular and circular pipes are used in most cases among the closed profile beam elements used in wide-spread sprayer booms. Consider the most loaded bending element of the boom, which is a rectangular pipe with a 3 section of 40x25x3 mm, which is subjected to a cyclic bend (Fig. 4).

As it is known [2] from in-situ studies of boom elements at constant dimensions H, h, t of pipe cross-section the maximum amplitude of stress change in it for this bend can be different:

$$140 \le \Delta\sigma \le 180\ MPa,\ R = 0{,}1,\ \ H = 40\ mm,\ h = 25\ mm,\quad t = 3\ mm\,. \tag{8}$$

The task is to determine by the relations (3)–(7) the resource $N = N_*$ (residual resource $N = N_*$) of the considered beam element of the closed profile.

In order to perform such calculations, you must define the characteristics of the material σ_0, N_0, t_0, K_{fC}, K_{th}, α_0. To this end, on the basis of the results of the works [4, 6], Weller's diagram and kinetic diagram of fatigue crack growth in Article 3 were identified and constructed (see Fig. 5).

Using the least squares method, equations (3), (4), and the data of these diagrams unknown constants $\sigma_0, N_0, K_{fC}, K_{th}, \alpha_0$ we will find as follows:

$$N_0 \approx 6{,}3\cdot 10^8\, cycles,\ \alpha_0 \approx 4{,}51\cdot 10^{-9}(cycle)^{-1}(MPa)^{-2},\ K_{fC} \approx 96\, MPa\sqrt{м},\ R = 0{,}1,$$
$$\sigma_0 \approx 88{,}23\, MPa,\ K_{th}^* \approx 12{,}81 MPa\sqrt{m},\ \sigma_t \approx 375\, MPa, \sigma_{-1} \approx 175\, MPa. \quad (9)$$

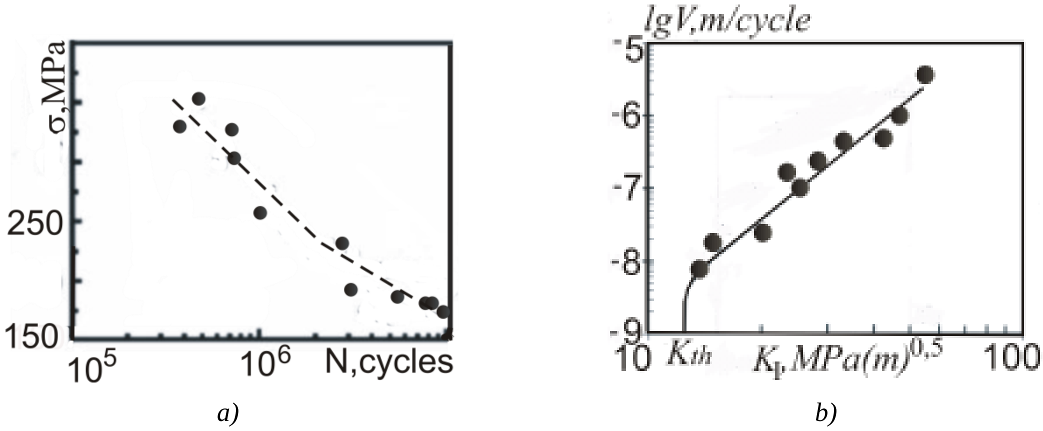


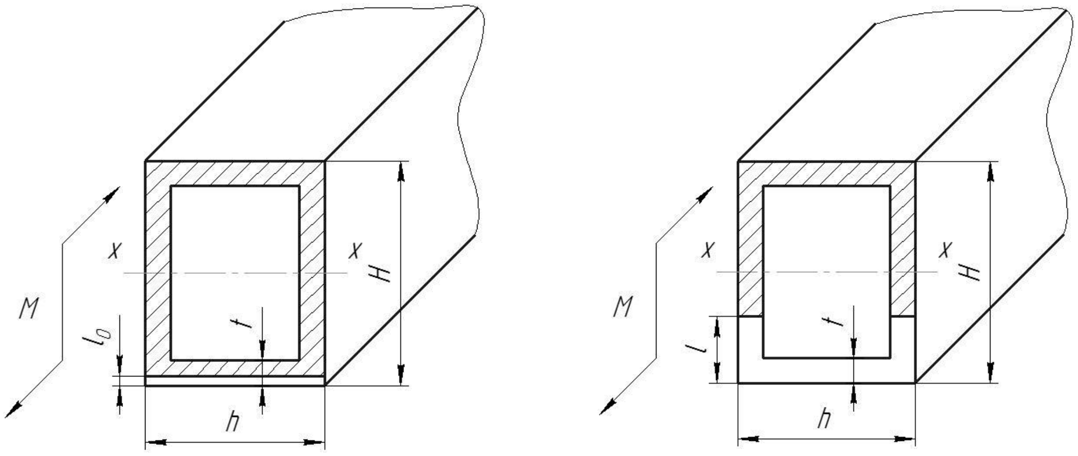
Figure 5. The diagram of Veller (*a*) and diagram of fatigue crack growth (*b*) for St. 3

In addition, a value l_0 is included in the ratios (6). This value is quite small in order of size of the structural parameter of the material. But for practical calculation, it is possible to select a value l_0 of at least one millimetre for the born crack to be considered macroscopic and to use the calculated model (4) and (6) lawfully. In such a case, the value N_* is calculated to be slightly understated, that is, the obtained error will go to the shelf life margin.

Since the amplitude $\Delta\sigma$ of stress variation in the beam element according to (8) is different, in order to establish the life of safe (non-destructive) operation of such a boom, we will calculate for the highest value $\Delta\sigma$, $\Delta\sigma \approx 180\, MPa$ or the same, cyclic bending of the analysed element by some moment $M \approx 6{,}4\cdot 10^{-4}\, MPa\cdot m^3$. For this purpose, on the basis of ratios (3) and (9), we find that the period N_3 of fatigue crack origin will be approximately equal to

$$N_3 = 6{,}3\cdot 10^6\ cycles \text{ or } t_3 = 1750\ h. \quad (10)$$

In the future, we believe that a straight surface crack with depth $l_0 = 0{,}001\, m$ was formed in the pipe. According to operational data, the most common defects in such beams are fatigue transverse cracks of the following two types: surface crack (Fig. 6, a) complex crack, occupies completely one of the walls and part of the other two (Fig. 6, b).



a) *b)*

Figure 6. Schemes of loading of rectangular pipe are with cracks:
a) the pipe with the superficial rectilinear crack; *b)* the pipe with the complex crack

During this period of resource exhaustion of the beam element of the rectangular profile of the frame structure can be represented as such. First, a surface crack occurs in the period N_{init} (Fig. 6, a), then such a crack propagates in the period $N_{subcr}^{(1)}$ before the complex configuration (Fig. 6, b), and finally, a crack of such a complex configuration will reach a critical size in the period $N_{subcr}^{(2)}$, causing complete damage of the beam element in question.

Thus, the life of a rectangular beam element can be determined by the following formula:

$$N_* = N_{init} + N_{subcr}^{(1)} + N_{subcr}^{(2)} . \qquad (11)$$

The value N_{init} is determined using formulas (9), and other components of formula (10) are determined using mathematical model (4)–(7) using results [5]. On the basis of this we will get

$$N_{subcr}^{(1)} = 60568 \; cycles \text{ або } t_{subcr}^{(1)} = 16{,}82\; h; \; N_{subcr}^{(2)} \approx 415 \; cycles., \text{ або } t_{subcr}^{(2)} \approx 7\, min. \qquad (12)$$

Setting (9), (11) to (10) we will find that the life of the boom t_* will be equal to

$$t_* \approx 1767 \; h. \qquad (13)$$

Conclusions. Using the basic provisions of modern theory of material fatigue, a calculation model has been built to determine the resource of load-bearing frames of sections of boom field sprayers, which are made of St. 3. Using this model, the boom life is calculated at maximum amplitude of cyclic bending of its weak elements, which is $t_* \approx 1767 \; h$. Compared to the known data [7], where the standard service life of the sprayers is 7 years, and their annual load is 550 *h*, and for the maximum amplitudes of oscillation of the boom elements accepted here, it will work about 3.2 seasons, does not correspond to the standard service life of the sprayers. This means that it is necessary to reduce (eliminate) vibrations of boom elements, as it is proposed in [8] or to reduce weight of boom section frames using lighter materials. One example of such a modification was the installation of OPSH-15 seven-section booms on the sprayers, which had extreme sections made of basaltoplastic composite [3], which significantly unloaded this structure, but without minimizing the fluctuations of the boom, especially extreme sections, it was not possible to achieve the necessary service life of its operation.

In addition, the result of the calculation shows that its life is mainly determined by the period of origin of the fatigue crack, as well as slightly in comparison with the standard almost 2 times.

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ВПЛИВ АМПЛІТУДИ КОЛИВАНЬ НА РЕСУРС НЕСУЧИХ КАРКАСІВ СЕКЦІЙ ШТАНГОВИХ ОБПРИСКУВАЧІВ

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Резюме. *Хімічний захист рослин – це один з найпоширеніших та дієвих засобів захисту рослин. Такий метод поряд зі своєю ефективністю може бути надто небезпечним для живих організмів та екології в цілому. Відносна безпечність даного методу полягає, перш за все, у чіткому дотриманні науковообґрунтованих вимог щодо здійснення такої технологічної операції. Серед вимог, крім дотримання норми та рівномірності внесення хімічного препарату, є терміни проведення обприскування. Тобто такі машини (оприскувачі) повинні бути технічно справними та готовими до роботи, щоб у стислі терміни провести хімічний захист.*

У роботі виконано аналіз основних технічних несправностей обприскувачів, де встановлено, що серед причин виходу з ладу обприскувачів вагоме місце займає руйнування каркасів широкозахватних штанг через виникнення втомних тріщин або дані елементи зазнають критичних перевантажень, наприклад при співударяння з нерухомою перешкодою внаслідок неправильної експлуатації. Що стосується аналітичного дослідження ресурсу роботи каркасів штанг широкої розгортки, то за аналізом літературних джерел така інформація практично відсутня. Тому в роботі побудовано розрахункові моделі для визначення ресурсу несучих каркасів секцій штангових польових обприскувачів, які виготовлені зі Ст. 3, на основі положень сучасної теорії втоми матеріалів. При застосуванні цієї моделі побудовано діаграму Веллера та діаграму росту втомної тріщини для Ст. 3 й проведено розрахунок ресурсу штанги за максимальних амплітуд циклічного згинання її найслабших елементів. Виконано порівняння з відомими нормативними даними та зроблено висновок про невідповідність існуючого ресурсу каркасів секції штанги до нормативного. Запропоновано шляхи вирішення даної проблеми.

Ключові слова: *обприскувач, каркас широкозахватної штанги, ресурс роботи, циклічне навантаження, амплітуда коливань, поверхнева тріщина.*

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