Stress-strain modelling state and fatigue of the working body of the tillage unit for weed removal

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Abstract. It is possible to increase the efficiency of crop cultivation technology through the use of multifunctional equipment for soil cultivation. The analysis of tillage equipment showed that the development of a multi-functional tillage unit capable of performing the main tillage with weed removal and a set of operations for pre-sowing tillage is relevant. To quickly adapt the soil cultivating unit to changing operating conditions, the method has been developed for regulating the angle of attack of the combing machine, the width of the capture unit and the angle of entry of the working body into the substrate [1]. A method for calculating the fatigue life of the working body of a tillage unit is proposed.

1. Introduction
Repeated application of loads on the working bodies of tillage machines, in particular on the working body of the combing unit, causes their destruction at stresses significantly lower than in the case of their single loading. With a large number of repeated loads, the fracture stress can be not only less than the tensile strength and yield strength, but even less than the elastic limit [1,2]. The destruction of parts as a result of the action of a large number of alternating stresses is a consequence of the fatigue of the metal of the working body of the combing unit. Since the working body is operated under load conditions of variable magnitude, taking into account the peculiarities of their action is necessary, since its destruction will lead to failure of the unit and disruption of the technological process of pre-sowing cultivation of the soil. The most appropriate technique in assessing fatigue life and conducting full-scale experiment is the use of computer technologies for engineering analysis of stress-strain state (SSS) and numerical simulation of structures, which include software packages MSC. Patran and MSC. Fatigue.

2. Materials and methods
To assess the SSS and fatigue of the working body, we consider the general case of its contact with an elastic-viscous substrate (soil). The load on the working body of the gun is determined as:

\[ F = -m\omega \frac{1}{\tan(2\alpha)} \]

where \( m \) is the mass of the working body, kg;
ω - acceleration of the working body, \( \frac{m}{s^2} \).
t\( \tan(2\alpha) \) is the angle of entry into the working body in the substrate, deg.
The kinematic parameters and device design for weed removal are shown in Figure 1.

By varying the basic structural parameters of the working body, it is possible to create the necessary SSS with a characteristic ratio of the main stresses in its design. In particular, it does not seem difficult to model the SSS of the working body in the design of the tillage unit for removing weeds in the area of its attachment to the disk. The estimated fatigue life of such a design, at present, has low accuracy, since the nature of the deformation of the working body and its attachment point from operational loads are not taken into account.

Modeling was carried out in two stages. At the first stage, a static assessment of the SSS of the working body in the MSC. Patran environment was carried out. The geometric model of the working body is built in the MSC. Patran preprocessor. The finite element mesh (CE) was generated by volumetric eight-node elements in the form of an isoperimetric hexahedron.

![Figure 1. Kinematic parameters and device design for weed removal](image)

The following boundary conditions are accepted:
- in the center of the disk with the working bodies - rotation around the Y axis is allowed;
- on the working body along the Y axis - the rated load of 365, 588 and 750 Newton is applied. The load is determined based on the formula 1 for the acceleration of rotation of the working body 5, 6 and 10 \( \frac{m}{s^2} \).

The material of the working body is Steel 65G according to GOST 1050-88 with the following mechanical characteristics:
- modulus of longitudinal elasticity \( E = 2 \times 10^5 \) MPa,
- Poisson's ratio \( \mu = 0.3 \),
- tensile strength \( \sigma_0 = 2200 \) MPa,
- yield strength \( \sigma_T = 790 \) MPa [4,5].

In the calculation model, the boundary conditions of fastening for the disk were taken as rigidly fixed along the supporting surface with the possibility of rotation around the Y axis. A force was applied to the working body along the entire length of the rod. The static solution of the computational model was carried out in the MSC.Patran postprocessor.

### 3. Results and discussion

The graphical interpretation of the results of the solution is shown in Figure 2. From the graphs of the stress isolines it follows that the maximum stresses are concentrated in the area of sealing the combing bar in the disk. So, the maximum stresses are observed during the rotation of the combing apparatus with an acceleration of 10 \( \frac{m}{s^2} \) and reach 2/3 of the yield strength of the material. The durability of the
working body in terms of fatigue strength with the identified type of SSS was estimated taking into account the linear accumulation of damage in the MSC. Fatigue software environment.

The load on the working body - 365 Newton (corresponds to the acceleration of rotation - $5 \frac{m}{s^2}$, the angle of entry into the working body in the substrate - $10^0$

The load on the working body - 588 Newton (corresponds to the acceleration of rotation - $6 \frac{m}{s^2}$, the angle of entry into the working body in the substrate - $10^0$

The load on the working body - 750 Newton (corresponds to the acceleration of rotation - $10 \frac{m}{s^2}$, the angle of entry into the working body in the substrate - $10^0$

Figure 2. Isolines of stresses of the working body of the apparatus for removing weeds

The fatigue life analysis procedure included the following steps:

- the history of loading of the working body was set, taking into account the static load, which varies in time;
- the curve of material fatigue and the probability of destruction of the working body were set.

The history of loading is given in the amplitudes of the stresses of the working body, based on its stress-strain state, in connection with this, the constructive resource (by fatigue strength) in the area of attachment to the disk is determined. Since the data on the structural fatigue curve are not known, the fatigue life was determined taking into account the existing Weler curve for Steel 65 G.

Consider an example with the conditions: the load on the working body is 365 N (corresponds to an acceleration of rotation of $5 \frac{m}{s^2}$, the angle of entry into the working body into the substrate is $10^0$. According to the calculated data (Figure 2), the applied force on the working body creates equivalent stresses in the seal corresponding to an amplitude of 299 MPa, while the cycle shape in calculating fatigue life with zero asymmetry is 0.5. Fatigue resistance characteristic Steel 65 G was determined based on the work [5, 6].

Since MSC.Fatigue accepts a linear model of the accumulation of Palmgren - Miner fatigue injuries, and also uses the zero cycle of the working body loading mode to determine durability, the effect of average stresses according to the Goodman hypothesis is taken into account in the numerical experiment. Taking into account the peculiarities of the MSC.Fatigue software package, the probability of sample destruction in the computational experiment is taken to be 50% [7].

As a result of calculating the fatigue life of the working body under complex stress conditions, taking into account the first hypothesis of strength, taking into account the maximum normal stresses in absolute value, failure will occur when:

\[ \sigma = 299 \text{ MPa after } 5.32 \cdot 10^7 \text{ loading cycles; } \]
\[ \sigma = 352 \text{ MPa after } 2.73 \cdot 10^5 \text{ loading cycles; } \]
\[ \sigma = 461 \text{ MPa after } 4.16 \cdot 10^4 \text{ loading cycles. } \]

4. Conclusion
The analysis of the calculated data allows us to state that the fracture zone will be located in the zone of maximum equivalent stresses located at the attachment point of the working body and the disk. The value of the SSS determined during the static tests does not significantly affect the location of the fracture site, but the value of the maximum equivalent stresses increases significantly from 299 to 461 MPa. At the maximum speed of movement of the working body, the calculated fatigue life does not exceed, in equivalent, the operating time of 1000 m traveled by the track unit. From what it becomes obvious, the need to improve the design of the working body and conduct fatigue tests of field samples.

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