Improving the durability of machine parts connections

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Abstract. Durability is the most significant component of reliability, which, depending on the purpose of the object and its operating condition, also includes indicators of failure-free, maintainability and preservation separately or a combination of them (both for the object and for its parts). Improving the strength and durability of machine parts connections, in particular, agricultural, at the present stage is achieved by various methods, the main of which is the method of statistical analysis of connections loading. This method is based on the theory of fatigue damage accumulation followed by strength calculation. The known calculation methods of durability of agricultural machines parts, based on this theory, often involve the use of data on the operation of the machine parts and units, its speed, the surface relief processed by the machine in real operating conditions, as well as duration of the impact of various loads (dynamic, peak or shock). Currently, agricultural producers have increased requirements for the machines used. They are not satisfied with agricultural machines that are often idle for a long time as a result of malfunctions and failures. In critical periods of operation this leads to a decrease in the quality of work, productivity and, ultimately, crop yields. In this regard, it is necessary to develop new agricultural equipment that will be characterized by high operational reliability.

1. Introduction
One of the main priorities of the country’s agricultural machinery industry is currently to improve the efficiency of agricultural machinery use [1-3].

A characteristic feature of agricultural machines is the short duration and frequency of their use when performing agricultural work, as well as the duration of storage. For example, the employment of combine harvesters in the year is 50 ... 60 days, machines for basic tillage, engaged in the spring plowing of the swell is 10 ... 15 days, in the fall on the plowing of the swell is 55 ... 60 days, grain drills for autumn and spring sowing is 25...30 days. More than half of the range of agricultural machines are used only 20 ... 40 days a year. At the same time, the technological maps for agricultural crops cultivation assume the performance of all agricultural work in accordance with the optimal agrotechnical deadlines. In this regard, of course, it is important that agricultural machines work flawlessly during the execution of technological operations. Violation of agrotechnical requirements leads to significant losses and reduction of crop yield.

Performance indicators of agricultural machinery use are significantly determined by the durability of threaded connections of operating parts and load-bearing structures. Therefore, the actual task is to study in detail the strength indicators of connections of operating parts and load-bearing structures of agricultural machines as the main criterion of durability [4, 5].
2. Research results

Improving the strength and durability of bearing connections of operating parts of agricultural machines that are operated under complex loading conditions is possible by using the proposed connection design [6-8]. The design scheme of the proposed connection is shown in figure 1 (a) and consists of a bushing 4, parts 1 and 2, a bolt 3, and a nut 5. In parts 1 and 2, a cylindrical hole is provided for the height of the connected parts, and the bolt is installed with a gap on the side of the head. On the side of the nut, a bushing is installed to fit the bolt 3 without a gap. During the connection assembly the bolt 3 passes freely through the hole of the part 1, since its diameter exceeds the bolt's landing diameter. In the hole of the bushing 4, the specified bolt enters with some tension. After that, the bolts are tightened with the specified force.

During the operation of a group bolt connection, as a result of the action of transverse loads, all bolts are deformed according to the type of three-support rods, with two edges pinched, and one support belt is located within the part 1.

The bolt is pinched in the section that coincides with the support surface of the head due to the fact that there is an elastic indentation of the bolt head 3 from the tightening. In part 2, the bolt is pinched due to both the fit without a gap in the bushing 4, and as a result of pressure from the nut on the washer pressed into this bushing. The last part as a result of tightening the bolt is elastic pressed into the part 2. The above circumstances help to increase the uniformity of loading of all bolts of the group bolt connection, which perceives the shear force as a result of the shift of two parts. In this case, the value of the uniform distribution of the transverse load is almost 1 and a significant increase in the real strength and durability of the connection is guaranteed. As a result, there is an increase in the margin of wear-resistant strength and durability of the connection. In addition, in this case, connection boring of the connected parts is practically excluded, which, as known, is rather labor-intensive technological operation.

The mathematical model of the proposed connection can be represented as a differential equation of the elastic line of the rod [9-11]:

\[
\frac{d^2}{dz^2} [EI(z) \frac{d^2y}{dz^2}] = q_y, \quad (1)
\]

The calculated connection scheme implementing equation (1) is presented as shown in figure 1 (b). To establish the forces and moments shown in this figure, we solved the equation (1) using the method of initial parameters [7, 8]:

\[
y(z) = EI_{\ell} \frac{d^2y}{dz^2} (0) + \frac{M_\ell z^2}{EI_{\ell} 2!} + \frac{R_\ell z^3}{EI_{\ell} 3!} + M_\ell (z-\ell)^2 + \frac{R_\ell (z-\ell)^3}{EI_{\ell} 3!}; \quad (2)
\]

\[
\frac{dy}{dz}(z) = EI_{\ell} \frac{d^2y}{dz^2} (0) + \frac{M_\ell z}{EI_{\ell} 2!} + \frac{R_\ell z^2}{EI_{\ell} 3!} + M_\ell (z-\ell)^2 + \frac{R_\ell (z-\ell)^3}{EI_{\ell} 3!}; \quad (3)
\]

when \( z = \ell \), we obtain \( y = -(\Delta + 2R_d) \). In this case according to (2) and (3) we obtain:

\[
-\frac{EI_{\ell} (\Delta + 2R_d)}{2} = \frac{M_\ell \ell^2}{2} + \frac{R_\ell \ell^3}{6}; \quad (4)
\]

\[
-\frac{EI_{\ell} \theta}{2} = M_\ell \ell + \frac{R_\ell \ell^2}{2}. \quad (5)
\]
Figure 1. Constructive (a) and design (b) schemes of the proposed connection.

when \( z = \ell_{ib} \), we obtain \( y = -\delta \). In this case according to (2) we obtain:

\[
- EI_z \delta = \frac{M_a \ell_{ib}^2}{2} + \frac{R_a \ell_{ib}^3}{6} + \frac{M_d (\ell_{ib} - \ell)^2}{2} + \frac{R_d (\ell_{ib} - \ell)^3}{6}.
\]

when \( z = \ell_{ib} \), we obtain \( \theta = \theta_0 = 0 \). In this case according to (3) we obtain:

\[
\begin{align*}
M_a \ell_{ib} + \frac{R_a \ell_{ib}^2}{2} + M_d (\ell_{ib} - \ell) + \frac{R_d (\ell_{ib} - \ell)^2}{2} &= 0. \\
M_a \ell_{ib} + \frac{R_a \ell_{ib}^2}{2} + M_d (\ell_{ib} - \ell) + \frac{R_d (\ell_{ib} - \ell)^2}{2} &= 0.
\end{align*}
\]

Analysis of equations (2) and (3) shows that they contain 6 unknown quantities: \( M_a, R_a, M_d, R_d, M_c, \) and \( R_c \). However only 4 equations were obtained for their calculation ((4) – (7)). In order to get the necessary number of equations, we attach a load-free console that is located behind the point \( B \), in the positive direction of the axis \( Z \), and consider the boundary conditions at the edge of this console at the point \( B_1 \). Since the points \( B \) and \( B_1 \) are theoretically quite close relative to each other at the edge of the console, where the moment and cutting force are absent, the boundary conditions are as follows: when \( z = \ell_p; y = -\delta; \theta = \theta_0 = 0 \). Thus, equations (2) and (3) can be written as follows:

\[
\begin{align*}
- EI_z \delta &= \frac{M_a \ell_p^2}{2} + \frac{R_a \ell_p^3}{6} + \frac{M_d (\ell_p - \ell)^2}{2} + \frac{R_d (\ell_p - \ell)^3}{6} + \\
&+ \frac{M_c (\ell_p - \ell_{ib})^2}{2} + \frac{R_c (\ell_p - \ell_{ib})^3}{6}; \\
M_a \ell_p + \frac{R_a \ell_p^2}{2} + \frac{R_d (\ell_p - \ell)^2}{2} + \frac{M_d (\ell_p - \ell)}{1} + \frac{R_d (\ell_p - \ell_{ib})^2}{2} + \\
&+ \frac{M_c (\ell_p - \ell_{ib})}{1} &= 0.
\end{align*}
\]
The resulting equations, which contain all unknown quantities, are the mathematical model of the problem under consideration.

If the bending rod is pinched in the support \( d \) with a gap \( \Delta \) and when selecting a gap at the edges of the belts, a moment appears. Figure 2 shows a diagram of the contact stresses that act on the rod if it is pinched at the edges of the belt, and the moment in the pinching \( M_d = M_z \), equivalent to the moment from the action of reactive forces.

![Figure 2. Load distribution diagram in case of pinching of the fastener rod](image)

In the case of zero clearance the value of the cross-section rotation angle is calculated using the expression:

\[
\theta_i = \arctg \frac{2\Delta}{\ell_n},
\]

that is, there is a contact, but there is no pinching:

\[
\beta = \ell \quad \text{if} \quad \theta = \frac{\theta}{2} \leq 0 \quad \text{then} \quad M_d = 0; \quad \text{if} \quad \theta > \frac{\theta}{2} \geq 0, \quad \text{then} \quad M_d = C \cdot \Delta \theta,
\]

where \( \ell_n \) is the belt length of part, in which at a certain gap \( \Delta \) between the hole and stud fastener there is a jamming of the rod in the belt with the advent of moment \( M_d = M_z \); \( C \) is the rigidity of the angular pinching of the rod in the belts; \( \theta \) is the angle of rotation of the rod cross section which coincides with the center of the belt parts, rad; \( \Delta \theta = \theta - \theta \) is the increment of the rotation angle due to deformation of the part with fixed edges, which move in opposite directions, rad.

We will divide the scheme of loading by transverse forces of the considered connection into three stages.

1. There is no gap \( \Delta \) choice in the rod support, therefore there is no bending moment. In this case, the values of reactions and moments at points \( A \) and \( C \) are equal.

2. The parts are displaced so much that a gap \( \Delta \) is chosen, the reaction \( R_d \) occurs, but there is no pinching. The second stage begins under the condition that follows from equations (4) – (9):

\[
\delta \geq \frac{I_{3,1} \cdot \ell_3 \cdot \ell_1^3}{I_2 \left(3\ell_1^5 - 2(\ell_1^3 - \ell_1^3)\right)}.
\]

3. Stage start at \( \Delta \theta > 0 \), so \( M_d \neq 0 \). In this case, there is a pinching of the fastener rod.

The proposed and standart design (figure 3) were tested for strength using an experimental installation equipped with special [7].
As a result of experimental research of the connection strength, it was found that the calculated values of bending stresses under the threaded part when the connected parts were shifted exceeded the experimental values for all variants of the initial tightening stress by 11.71...12.75% (table 1).

The reason for the difference is that the rigid sealing of the edges, adopted in the design scheme, is not fully implemented in the threaded part due to the fact that it is more pliable in comparison with the bolt rod. This circumstance is explained by the fact that the free section of the threaded part turns when displaced, while the nut remains stationary relative to the support part. In addition, the result is influenced by the measurement error during experimental studies (about 3 MPa, or an average of 1% of the measured values).

The results of comparative analysis of bending stresses obtained experimentally and calculated for the proposed design are summarized in table 2. The table shows that the experimental values of the bending stresses near the first turn of the bolt threaded part are less than the calculated values by 3.99...11.08%.

Small bending stresses near the bolt threaded part in the proposed design are explained by the fact that there are gaps between the bolt belts and the holes of the cylindrical bushings in the range of 0 ... 10 microns.
The results of experimental research indicate that with a small value of the initial bolt tightening stress (50 MPa) and the same transverse load equal to 2.16 kN, the connection rigidity of the proposed design is 20 times greater than the standard one, and the bending stress is 16.5 times less.

With an increase in the initial tightening stress to 200 MPa at the same transverse load of 6.33 kN, the effect is less pronounced. So, in this case, the connection rigidity of the proposed design for a shift in comparison with the standard one is 10.9 times greater, and the bending stress is 7.6 times less. The noted disproportionality in changes in the connection rigidity and bending stresses can be explained by the fact that with an increase in the initial tightening stress, the role of friction forces between the plates increases significantly.

In real conditions, agricultural machines operate under complex dynamic loads, when the friction forces in the parts connections are practically not manifested. With this in mind, the values of connection and bending stresses in the fasteners of the proposed design will correlate with the results obtained at low initial bolt tightening stresses.

3. Conclusion
In the case of installing the proposed design bolts in the part to which the nut is attached, with a slight tension, a rigid sealing of the bolt rod is guaranteed. As a result, bending stresses in sections that are directly located near the thread and in the thread itself are eliminated, which will significantly improve the durability of threaded connections of agricultural machines in real operating conditions.

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