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Factors of Influence of Ore and Rocks on the Dump Truck Body Under Quarry Conditions

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Abstract

In this scientific work, structural, technological, and operational solutions aimed at increasing the service life of dump truck bodies operating under quarry conditions were systematically studied. It was established that the high abrasiveness of quarry rocks, the intensity of load impact, humidity, and corrosion factors cause rapid wear of the body metal, and the mechanisms of action of these processes were analyzed. In the study, structural weak zones were identified by optimizing the body geometry, increasing the strength of welds, reducing stress concentrations, and conducting stress-strain analysis in the ANSYS program.

Keywords: Quarry dump truck, body, life, wear, abrasive impact, hard rocks, hardwall, composite coating, lining, construction steel, body geometry, stress analysis, ANSYS, impact loads, torsion, corrosion.

Introduction

Trucks used in quarry conditions are one of the main vehicles of the mining industry, and their effective operation directly affects the overall production indicators of the enterprise. Heavy ores, hard and highly abrasive rocks, wet and frozen environments, as well as dynamic impacts arising during loading and unloading, are the most vulnerable and stressed parts of the dump truck body. Uneven stress distribution, concentrated load drop, high abrasive friction, and repeated impacts lead to rapid fatigue, deformation, and a significant reduction in the service life of the body metal. The physical and mechanical properties of quarry rocks - density, hardness, abrasiveness, granulometric composition, moisture content, and viscosity - determine the intensity of forces acting on the body during ore discharge. Particularly, the high-speed lowering of uneven, coarse fractional fragments formed as a result of blasting operations into the body by the excavator creates high impact loads on the metal surface. This process causes, on the one hand, local plastic deformations, and on the other hand, fatigue cracks over time.

Abrasive wear on the inner surfaces of the body, caused by the friction of ore and rocks, increases the main operating costs of the dump truck. Especially hard ore massifs (iron ores, quartz rocks, porphyries) cause significantly faster deterioration of the bottom, rear, and side walls of the body. In addition, high temperature differences (from -30°C to +50°C), extremely humid or dusty

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environments, and the effects of chemically aggressive substances increase the metal's susceptibility to corrosion.

Therefore, the service life of quarry dump truck bodies depends on many factors, among which the physical and mechanical properties of ores and rocks are the main influencing factor. The relevance of this research lies in the fact that, by studying the specific mechanisms of the load composition and loading process in real mining conditions, it is possible to develop new technical solutions to reduce the wear rate of the dump truck body, increase its strength indicators, and extend its service life.

In this work, the main factors influencing the body of a dump truck under quarry conditions, their mechanical properties, dynamic forces during unloading, abrasive friction processes, corrosive effects, as well as the analysis of stresses arising in the body as a result of these factors are scientifically studied. Based on the obtained results, technical strategies aimed at improving the design solutions of the body, selecting durable alloys and lining, determining the optimal geometric shape, and extending the service life are proposed.

METHODOLOGY

The forces of load and pressure on the surfaces of the Belaz-7555 car body can be theoretically analyzed in two states: static and dynamic. It is possible to determine the impact force on the sides of a standard dump truck body with a lower body thickness of 10 mm.

A mathematical model of the change in the load-bearing capacity and wear rate of the dump truck body after the introduction of changes has been developed.

$F_{IN=0}$In the static position, the movement of the load and the dump truck is considered in the null position. Here

$F_{IN\neq 0}$In a dynamic state, the movement of the truck and dump truck is considered in the presence of

Let's model the pressure distribution for the bottom section after the load is fully loaded. Since this load distribution is variable, we accept the following function change q .

$Q(x) = k \cdot x^2$ (1)

Since the above function is variable, let's enter the interval

$x \in [\mathrm{X}, \mathrm{L}]$, L-rock length

Assume that the load is inclined to the front and the pressure is distributed parabolically, with the maximum value at the front (X=0)

$$q(x) = q_{max} \cdot (1 - (\frac{X}{L})^2) \quad (2)$$

$q_{max}$where: q (x) - pressure at each point (Pa); - maximum pressure (Pa)

Determining the total pressure

$q(x) = q_{max} \cdot (X/L)^2$If there is a pressure distribution then force

$$F = \int_0^L q(x) \cdot B \cdot dx = Bq_{max} \cdot \int_0^L (1 - (\frac{X}{L})^2) \cdot dx => \int_0^L \left(1 - \left(\frac{X}{L}\right)^2\right) \cdot dx = = L\left(1 - \frac{2}{3}\right) = \frac{L}{3} \quad (3)$$

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$$F = B \cdot q_{max} \cdot \frac{L}{3}$$

If the total weight pressure is calculated in the range of forces

Here B is the width of the body, L is the length, h is the height.

Calculate areas for the bottom, side, and front

$$\begin{cases} S_1 = B \cdot h \\ S_2 = L \cdot h \ (4) \\ S_3 = L \cdot B \end{cases}$$

Load for the front part. Let's compile the pressure distribution change.

$q(y) = q_{max} \cdot (\frac{y}{h})^2$ (5)

$F = \int_0^h q(y) \cdot h \cdot dy = hq_{max} \cdot \int_0^h (\frac{y}{h})^2) \cdot dy => \int_0^h (\frac{y}{h})^2 \cdot dy = 0$ (6)

$y \in [0, h]$at the front, the pressure force is located in the lower zero direction.

This function remains a linearly decreasing function.

Theoretically (based on mechanical axioms), pressure decreases in the lower layers as a result of rising to the upper and higher layers.

1. $L/_3$For the bottom heater, the pressure increases from the front and is maximal at the point and then decreases in a parabolic manner and equals zero.
2. The pressure at the front is greatest at the lowest point and linearly decreases as it moves upward, reaching zero at 1.75 m.
3. $L/_3$The pressure on the side wall is maximal at the point and then parobolically decreases, eventually reaching zero.

The change in the distributed load force q (x) in a parabolic form, the determination of the total force by integrating it, and the graphical representations are fully analyzed. The distributed force function is defined as follows:

$q(x) = q_{max} \cdot (1 - (\frac{x}{L})^2)$ (7)

$q(x) - x$where: distributed force at the point (N/m2), this force represents the pressure or force density per unit length.

q_{max} −The maximum force density (N/m2) is usually in the central part of the profile.

x −horizontal coordinate (m), i.e., the position of the observed point along the section.

x −total length (m) - the length of the entire element.

• B - width or depth (m) of the element, which represents the depth measurement in the 3D element.

The following integral is used to convert the distributed force into the total force:

$F = \int_0^L q(x) \cdot B \cdot dx = B \cdot q_{max} \int_0^L \left(1 - \left(\frac{x}{L}\right)^2\right) dx = B \cdot q_{max} \cdot \frac{2L}{3}$ (8)

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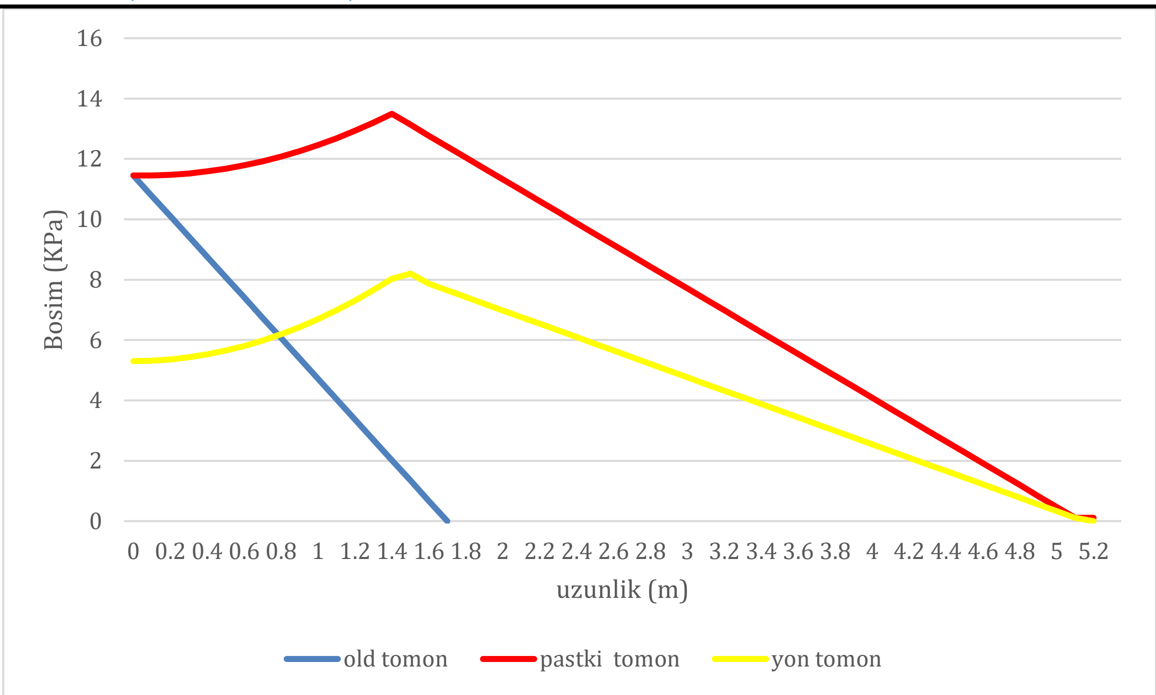

Figure 1. Graph of pressure distribution in the front, side, and lower parts of the BelAZ-7555 truck.

Front side (blue): The pressure decreases rapidly, after 1.75 meters it drops to zero due to the absence of load in the dynamic state this pressure can differ from these values by several times. The maximum pressure of 11.45 kPa is located at the bottom point of the front.
Bottom side (red): The highest pressure rises from the starting point x=0 to 11.45 kPa at a pressure of about 1.71 m (approximately 13.65 kPa), the highest at the center of mass, then linearly decreases to the rear of the body.
Side (yellow): Slowly rises, reaches maximum point, then falls (~8.3 kPa)
This confirms the balanced load on the rear and front wheels of the truck.
The pressure is maximal at one-third of the lower length - this indicates that the center of gravity is at the central point of the lower platform. Although this is a natural distribution, it is not linear - it can be a slight asymmetry or uneven load.
It can be observed that the highest pressure is observed near the maximum in one-third, i.e., at the center of mass, similar to the symmetrical distribution. However, the descent is not very steep - this indicates that the sides are compressing the load. These graphs are very useful for graphical evaluation of the mathematical formula of load distribution, identification of observation points, and determination of probable structural stress zones. -
These analyses are especially important in determining the risk of deformation of the BelAZ body, load, or overturning along all sides. In the static state, the dynamic data (during the movement of the BelAZ with a load) may differ several times, since additional dynamic values and load states may arise.

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CONSIDERATION

The stress distribution is similar to the 2D Gaussian normal distribution, where the stress reaches its maximum value near the center and gradually decreases.

In this case, its mathematical expression (surface force function) is:

$$\sigma(x,y) = \sigma_{max} * e^{-\left(\frac{(x-x_1)^2}{2\sigma_x^2}+\frac{(y-y_1)^2}{2\sigma_y^2}\right)} \qquad (9)$$

$\sigma(x,y)$ −where: surface tension of the body, MPa;

$\sigma_{max} = 0{,}053\ MPa$ −Maximum stress value.

$x, y \in [0{,}1]$ −Relative coordinates along the body length and width.

$x_0 \approx 0{,}45;\ y_0 \approx 0{,}25$ −Coordinates of the stress center.

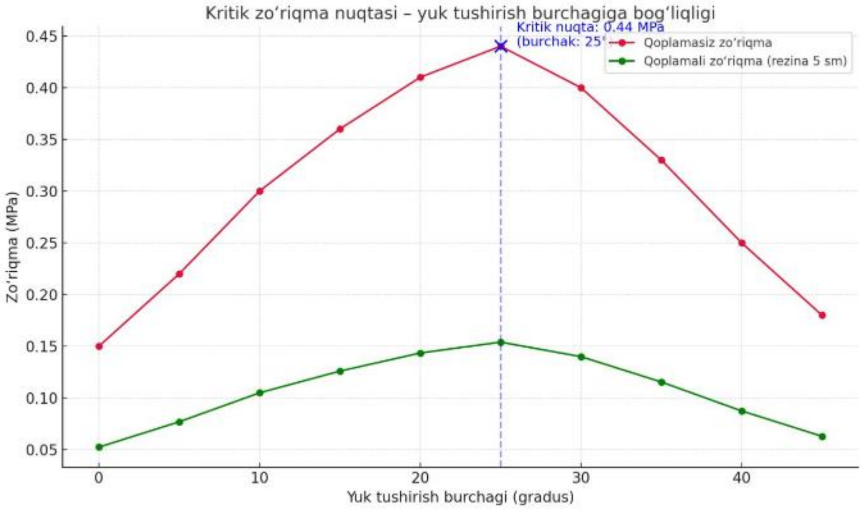

Figure 1. Graph of the dependence of the force and the angle of unloading during unloading from the dump truck body.

Studies have shown that during unloading with a body without a covering, the maximum stress value is 0.44 MPa and is considered the most critical zone in the range of 20-30°, and after the installation of a rubber covering, this critical zone remains precisely in the range of 20-30°, but the maximum stress value is 0.15 MPa. In this research work, the stress during unloading from the body was reduced by 66%.

With the help of this software, a virtual experimental test was conducted, and it can be concluded that during the process from the horizontal position of the load to its unloading, the stress zone formed in the middle part of the body, with the dumping of the load, creates friction between the rubber body covering, and on the basis of this friction, the rubber covering undergoes wear.

Now if the 55-ton load fully affects this body.

$$\mathrm{F} = 55000\mathrm{x}9{,}81 = 539{,}550\mathrm{N}$$

$$\sigma = \frac{\mathrm{F}}{\mathrm{A}} = \frac{539{,}550}{18{,}91} = 28{,}52\mathrm{kPa} = 0{,}0285\ \mathrm{Mpa}$$

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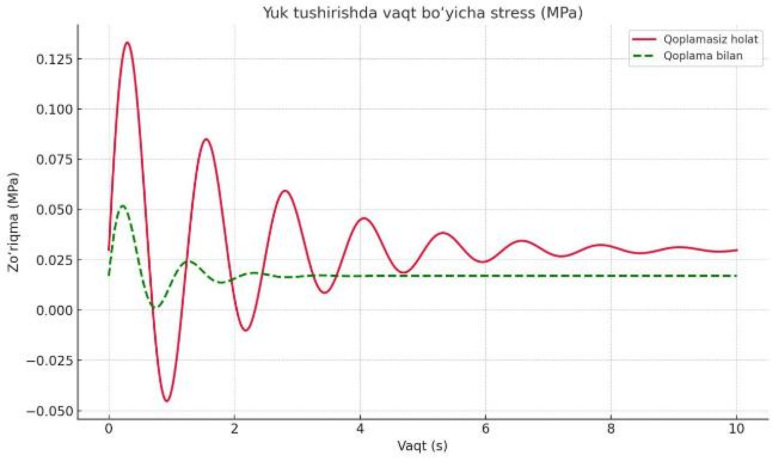

Figure 2. Graph of the dependence of the stress on time in the case of a dump truck body without a rubber coating and with a coating during unloading.

The graph of the dependence of the voltage on time when unloading a dump truck body without a rubber coating and with a coating is shown in Figure 4.14, which shows that the body without a coating extends to a maximum of 0.13 MPa and extinguishes within 7-8 seconds. After the rubber coating is installed, it reaches a maximum of 0.05 MPa, and internal damping occurs within 2 seconds. This leads to a decrease in the initial internal load by 0.08 MPa (61.5%), and a decrease in the time unit by 5 s (71.4%).

Effect of unloading angle:

When the angle of incidence changes, the components of the force also change. This is estimated by the following formula.

$F_n = F \cdot \cos\theta\,; \quad F_t = F \cdot \sin\theta$ (10)

$\theta - F_n - F_t$ –where: angle of incidence (degree); normal force (vertical pressure); tangential force

The stress at each angle is expressed as follows:

$\sigma(\theta) = \frac{F_n(\theta)}{A} + \Delta\sigma_t(\theta)$ (11)

Considering the role of the coating:

Because the rubber coating absorbs energy, the forces are not fully transferred to the rigid structure (the metal part of the body). Therefore:

Uncovered:

$\sigma_{metal}(\theta) = \frac{F\cos(\theta)}{A}$ (12)

Coated state

$\sigma_{rezina}(\theta) = \alpha \cdot \frac{F\cos(\theta)}{A}$ (13)

$$0 < \alpha < 1$$

α –where: depreciation coefficient (usually taken in the range of 0.3-0.5).

Based on the obtained results, we can see that the stress occurs at 25^0 as a critical angle, the result is 0.44 MPa in the case without coating and 0.15 MPa in the case with coating.

Hooke's Law (in the region of elasticity):

$\sigma = E \cdot \varepsilon$ (14)

$E - \varepsilon$ –where:young modulus; relative desertification.

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$M = F \cdot d$Conversion of forces into moments: the distribution of stress between elements is influenced by the modulus of the material and the angle.

The force graph is a curve near the sinusoid and paraboloid:

This corresponds to the general type formula below:

$$\sigma(\theta) \approx \alpha \cdot \sin(b \cdot \theta + c) \qquad (15)$$

or

$$\sigma(\theta) \approx -\alpha \cdot (\theta - \theta_0)^2 + \sigma_{max} \qquad (16)$$

The decay process is represented by a modified exponential or linear decay model.

a) for rubber residue (rapid wear):

When the rate of decay is constant:

$$h_r(t) = h_{r0} - v_r \cdot t \qquad (17)$$

$h_r(t)$ —where: thickness of the rubber at the moment of time (mm);

$h_{r0} = 50$ —initial thickness

$v_r = 0{,}03 \frac{mm}{sikl}$ —*decay rate*

t —*time (days)*

Archard's law - for abrasive wear

$$V = \frac{K \cdot F \cdot s}{H} \qquad (18)$$

$V - K - F - s$ —where:absorbed volume;age coefficient;surface force;sliding distance;H-hardness of the material

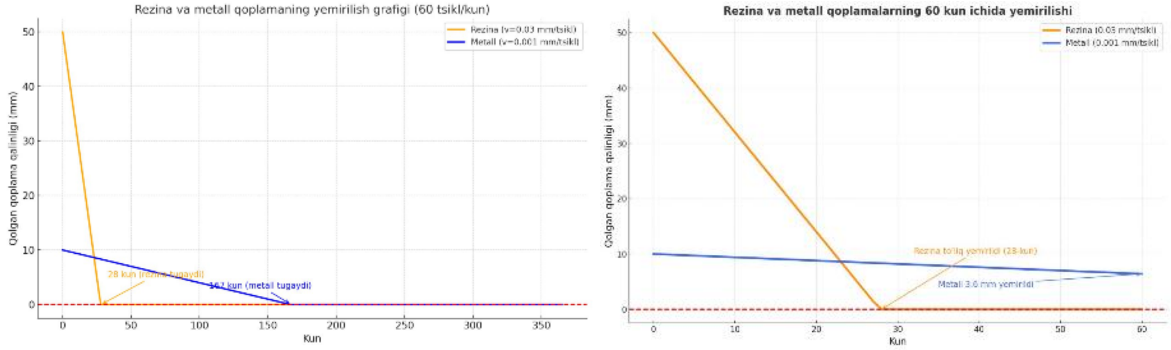
Distribution of forces along the lengtheron (for the new and old frame) according to the Euler-Bernoulli theory of linear bending

$$M(x) = EI \frac{d^2 y(x)}{dx^2} \qquad (19)$$

$M(x) - E - I - y(x) - x$ —where: bending moment; modulus of elasticity; moment of inertia; bending line; coordinate along the length of the lengtheron

CONCLUSION

Based on the obtained experimental and calculated results, **the** wear rate **of the bottom metal of the body** and **the rubber lining** was analyzed over time. Based on these data, a graph of the dependence " Depth of wear - time" was constructed (Fig. 3a).



a b

Figure 3. Graph of the dependence of the wear duration of the metal and rubber covering of the lower part of the body (a-167 days, b-60 days).

Graphical analysis shows that under intensive abrasive loads in quarry conditions, the total wear time **of steel body metal 10 mm thick is 167 days.** In this interval, the wear process is linear, and over time, the strength of the metal against friction and impact decreases sharply.

The dynamics of wear of the **rubber lining** applied to the inner surface of the body was also evaluated separately. As can be seen from the graph, due to the fact that the rubber coating is much softer than the metal, it completely wears out within **28 days** and reaches a condition requiring replacement. This confirms that although rubber lining mitigates high impact and abrasive loads, its service life is significantly shorter than that of metal.

As a result of production observations, it was established that the body without coating is in a state requiring repair within **60 days** of operation. During this period, **a 3.6 mm** of metal wear was observed in the lower part of the body, i.e., a loss of **36%** of the initial 10 mm thickness. This indicator confirms the correspondence between the actual operating loads under production conditions and the linear wear rate calculated on the graph.

The graphical results mean:

- The rubber coating lasts a short time, but significantly reduces the impact on the metal**,**
- unlined metal body wears out faster**,**
- regular rubber coating replacement increases the overall service life of the metal by 2-3 times**,**
- yeyilish tezligi yuk tarkibi, yuk tushirish balandligi va metallning sirt qattiqligiga to‘g‘ridan-to‘g‘ri bog‘liqdir.

Thus, based on the graphs, it can be concluded that the 28-day service life of the rubber coating protected the metal from 36% wear within 60 days**,** which indicates the technical and economic efficiency of the lining application.

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